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

W. L. Greenstreet
S. E. Moore
J. F. Callahan

OAK RIDGE NATIONAL LABORATORY

operated by

UNION CARBIDE CORPORATION

for the

U.S. ATOMIC ENERGY COMMISSION

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Applied Mechanics Section
REACTOR DIVISION

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(NUCLEAR SAFETY)**

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for Piping, Pumps, and Valves

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and Analytical Investigations of Nozzles

JULY 1971

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee
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Second Annual Progress Report on Studies in Applied Solid Mechanics (NUCLEAR SAFETY)

ABSTRACT

Accomplishments during calendar year 1970 under Oak Ridge National Laboratory programs for developing structural design technology for piping components and for attachments of nozzles to pressure vessels are summarized in this report. Both theoretical and experimental efforts are under way to develop improved design methods for assuring, on a practical level, the structural integrity of pressure vessels and primary coolant systems of nuclear power plants. Applications of the technology developed are not limited, however, to nuclear reactor systems. Through these programs, experimental data on structural behavior are being obtained and experimentally proven methods of design analysis established. Design rules and analytical procedures for incorporation into current industry-developed standards and USAEC Division of Reactor Development and Technology (RDT) supplemental standards are also derived. The piping and nozzle programs are discussed in Parts I and II of the report, respectively.

INTRODUCTION AND SUMMARY

This report describes progress during calendar year 1970 on programs for developing structural design technology for piping components and for attachments of nozzles to pressure vessels. These programs comprise the studies in applied solid mechanics being carried out as a part of the Oak Ridge National Laboratory's nuclear safety program. Both the nozzle and piping programs are being conducted in cooperation with

industry through the Pressure Vessel Research Committee (PVRC) of the Welding Research Council. In the case of the program on piping components, work sponsored by Oak Ridge National Laboratory is discussed; while in the case of the nozzle program, descriptions of work sponsored by PVRC are also included. Progress reports on these two programs are also included in the Oak Ridge National Laboratory's nuclear safety program bimonthly and annual reports.

The nozzle and piping programs are directed toward producing experimental data on structural behavior and establishing experimentally proven methods of design and analysis. As a result, greater assurances of structural integrity of pressure vessels and primary coolant systems of nuclear power plants are obtained, and current industry-developed codes and standards for nonnuclear, as well as nuclear components, can be expanded and improved.

Another important benefit from this work is associated with recent rapid advances in analytical methods development, in general. These advances have intensified the need for accurate, definitive experimental data for analysis assessment and verification. The significant amounts of precise experimental data being obtained under the nozzle and piping programs will be very useful in meeting the needs of developers and analysts.

Under the program on design technology for piping components, a survey and engineering evaluation report covering available information pertaining to the structural design and stress analysis of piping system components that existed prior to the initiation of the ORNL program was published. Experimental and analytical studies are in progress on normally intersecting

thin-walled cylindrical shells (branch connections), commercial piping tees, and commercial elbows. Scheduled tests on some of these items were completed. In addition, analytical studies to provide stress indices for several other configurations, such as straight pipe with out-of-round cross sections, pipe with integral attachments (lugs), etc., were completed, and the associated design rules are being developed for inclusion in design codes.

The work under the program on attachments of nozzles to pressure vessels includes studies of single and closely spaced nozzles, which may be either radially or nonradially attached. In the area of elastic behavior, analysis methods were developed and are being used to investigate reinforcement configurations for single radial nozzles attached to spherical shells. In addition, the development of a method for analyses of single, unreinforced, nonradial nozzles attached to spherical

vessels was completed, and experimental work for this configuration is being continued. Developments related to the first phase, that is, the consideration of ideal geometries, are in progress in the case of cylindrical shells. Experimental investigations of the effects of closely spaced holes in flat plates were completed, as was the test of a spherical vessel with closely spaced holes. Experiments and theory development are in progress, with some portions having been completed.

Parametric studies of reinforced attachments of nozzles to spherical shells were made with available computer codes for design-code use. In addition, design rules are being prepared on the basis of studies to date.

Studies being carried out under PVRC sponsorship for determining plastic collapse loads for cylindrical and spherical vessels with single nozzles are in progress. Portions of these studies have been completed.

Part I. Design Criteria for Piping, Pumps, and Valves

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Program Description

The ORNL piping program is the AEC portion of a joint AEC-industry program for the development, refinement, and extension of methods for structural design of nuclear-service piping, pumps, and valves. The joint program is one of several AEC-industry programs for the development of improved codes and standards to assure the structural integrity of nuclear reactor pressure vessels and primary coolant systems. The objectives of the piping program are to develop basic information on the structural behavior of piping system components and to prepare this information in forms suitable for use in both AEC-RDT standards and industry codes and standards.

The primary tasks to be carried out are accurate and thorough experimental and analytical stress analyses of piping components. At present, the experimental work is confined to tees, branch connections, and elbows, while analytical studies are planned for all standard pipe fittings. From the analyses and companion studies, data correlations and evaluations will be made; design charts, graphs, and tables will be prepared; and code rules will be drafted for use by the various code bodies. The data will be presented in terms of stress indices and flexibility factors for direct use. Overall interpretive reports will also be written on work sponsored under this program and on that done by others.

The ORNL piping program is an outgrowth of a research proposal¹ that outlined 12 tasks for the development of stress indices and methods of analysis.

The proposal was developed by an ad hoc committee of the Pressure Vessel Research Committee (PVRC) of the Welding Research Council. The AEC (ORNL)-sponsored portion consists of work on piping components, while industry is charged with tasks on pumps and valves and the development of low-cycle fatigue design criteria. Coordination of the overall research, including the work sponsored by industry, is accomplished through the PVRC Subcommittee to Develop Stress Indices for Piping, Pumps, and Valves.

The Oak Ridge National Laboratory was charged with the responsibility for planning, managing, and directing the AEC portion of the cooperative program, with specific management functions to include the following:

1. detailed technical planning of the program and development of work priorities, within budgetary limitations;
2. procurement of consultant services, materials, and placing of contracts for research and development;
3. coordination of work with cooperating organizations;
4. periodic reporting of progress of work;

1. *Program and Request for Proposals for Development of Stress Indices and Methods of Analysis for Piping Components, Pumps, and Valves*, unnumbered PVRC report, dated July 1, 1967.

5. preparation of summary reports;
6. development of recommended requirements for specifications, codes, and standards needed to provide assurance of the continued safety of nuclear plant piping, pumps, and valves.

The program has been conducted according to the guidelines established by a PVRC ad hoc committee, but since ref. 1 was prepared, changes have been made in the program and information regarding the associated work tasks has been developed in much greater detail. In order to facilitate management of the AEC portion of the program by ORNL, a more comprehensive program plan² was recently developed and distributed to the AEC and members of the PVRC Subcommittee to Develop Stress Indices for Piping, Pumps, and Valves. The program presently consists of seven major tasks and various subtasks, which include all the work on piping components requested by the PVRC in ref. 1 plus additional work on limit-load studies requested by the AEC.

Summary descriptions of work completed during the calendar year 1970 are presented in this report, which describes new work, as well as the continuation of the efforts reported for calendar year 1969 (refs. 3 and 4). In addition to these annual reports, periodic progress reports, both written and oral, have been given at meetings of the Pressure Vessel Research Committee, Design Division, and Subcommittees. Annual information meetings sponsored by the Oak Ridge National Laboratory are also used to report progress and the current status of the work. During 1970 the status of the ORNL piping program was reported at two such meetings: the Nuclear Safety Program Annual Information Meeting, February 9-11, 1970, and the Second Annual Information Meeting - Studies in Applied Solid Mechanics, April 2-3, 1970. Annual information meetings will also be held early in 1971 to report on progress during calendar year 1970.

2. S. E. Moore and W. L. Greenstreet, *Program Plan for the ORNL Piping Program -- Design Criteria for Piping, Pumps, and Valves* (unpublished).

3. W. L. Greenstreet, S. E. Moore, and R. C. Gwaltney, *Progress Report on Studies in Applied Solid Mechanics*, USAEC Report ORNL-4576, Oak Ridge National Laboratory, August 1970.

4. W. L. Greenstreet et al., "Design Criteria for Piping, Pumps, and Valves," pp. 124-148, *ORNL Nuclear Safety Program Ann. Progr. Rept. Dec. 31, 1969*, USAEC Report ORNL-4511, Oak Ridge National Laboratory.

Survey Reports

Because of the relatively long history associated with the design of piping systems, an abundance of information has been developed, much of which can be used in the development of design methods for nuclear piping systems. Thus in order to form a basis for further analytical and experimental studies, one task consisted of collecting and critically evaluating available information pertinent to the stress analysis and design of piping components. While such a survey was implied to some extent by the PVRC proposal,¹ no specific task was assigned. Both the AEC and the PVRC, however, agreed that a literature survey was a necessary first step. The first major objective of the program was therefore to identify and evaluate existing test data and analytical methods with regard to their applicability in the development of stress indices and structural design methods.

The draft of the survey report was completed in 1969, and over 100 copies were distributed throughout the United States for review and comment. Additional material and comments from reviewers were then used in preparing the final report.⁵ To the extent possible, the report provides a comprehensive list of references and summary discussions of the research work completed prior to 1969 on the stress analysis and structural design of piping systems and components. Nearly 800 papers, documents, and reports were examined and referenced. Thus, the report gives an invaluable summary of work on piping components and represents the state of the art prior to the initiation of the ORNL piping program.

Straight Pipe, In-Line Configurations, and Attachments

A number of analytical parameter studies originally requested by the PVRC were for components that could be classed under the general heading of straight pipe. Included in this category were studies on the effects of lug attachments, out-of-roundness, and circumferential butt welds in straight pipe and the development of stress indices for socket welded and threaded fittings, concentric reducers, transition joints, and flanged joints. Studies of all these were to be based on existing theoretical solutions and correlated with

5. E. C. Rodabaugh and A. G. Pickett, *Survey Report on Structural Design of Piping Systems and Components*, USAEC Report TID-25553, December 1970.

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, in some cases it is now apparent that additional experimental data are needed. Analytical parameter studies for several of the components were completed during the past year, and the results are summarized below.

Stresses in Out-of-Round Pipe

A nonlinear strength-of-materials solution developed by Haigh⁶ in 1936 was used as the basis for studying the effects of circumferential bending stresses caused by internal pressure in a straight pipe with an out-of-round cross section.⁷ Haigh's formulation was based on assumptions that thin-walled shell theory is valid and that deviations from a circular cross-sectional shape are small in comparison with the average radius of the pipe and can be adequately represented by a Fourier sine-cosine series:

$$r_o = R + \sum_{n=1}^k a_n \cos n\phi + \sum_{n=1}^k c_n \sin n\phi, \quad (1)$$

where

r_o = initial mean radius of pipe,

R = average mean radius,

ϕ = circumferential angle,

a_n, c_n = constants.

The circumferential stress is given by

$$\sigma_c = \frac{pR}{t} \left(1 \pm \frac{6}{t} \sum_{n=1}^k \frac{a_n \cos n\phi + c_n \sin n\phi}{1 + J_n} \right), \quad (2)$$

where

$$J_n = \frac{pR}{t} \frac{12(1 - \nu^2)R^2}{Et^2(n^2 - 1)},$$

p = internal pressure,

t = uniform wall thickness,

E = modulus of elasticity,

ν = Poisson's ratio.

The effects of out-of-roundness on the stresses in straight pipe were then investigated by studying three specific types that may be considered typical: (1) essentially elliptical cross section, (2) flat spots that are long relative to the pipe diameter, and (3) peaks that could occur along a seam weld. By proper choice of coefficients, the stresses in all three can be calculated from Eqs. (1) and (2).

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 and USAS B31.7-1969 (refs. 8 and 9). This conclusion has special significance for designers of nuclear power piping systems who must satisfy the requirements of the nuclear section of the piping code USAS B31.7. A strict interpretation of the rules in the 1969 edition would require that the cross sections of straight pipe, curved pipe, or welding-end elbow be measured and used to calculate the stresses and the peak stress index K_1 for evaluating the fatigue life under cyclic pressure loading.

Proposed revisions to the code rules were developed that provide the designer with more complete guidance and, in most cases, obviate the need for complete dimensional inspection. It is necessary only to assure that the pipe meets the dimensional requirements specified in the purchasing agreement. These rule revisions are currently being considered by the code committee for inclusion in the next issue of the nuclear piping code.⁹

Integral Attachments to Straight Pipe

Structural attachments such as lugs welded directly to the pipe are often used to connect a piping system with its supporting elements. Since significant local stresses may be induced in the pipe by the force and moment loadings transmitted, these stresses must be considered in checking for compliance with the design stress criteria of the codes (see, for example, Paragraph 1-704.3.6 of USAS B31.7-1969). The code provides,

6. B. P. Haigh, "An Estimate of the Bending Stresses Induced by Pressure in a Tube That Is Not Initially Quite Circular," *Proc. Inst. Mech. Engrs.*, 133: 96-98 (1936).

7. E. C. Rodabaugh, *Stresses in Out-of-Round Pipe Due to Internal Pressure*, USAEC Report ORNL-TM-3244, Oak Ridge National Laboratory, January 1971.

8. *General Requirements for Specialized Carbon and Alloy Steel Pipe*, ASTM A530-66, published by the American Society for Testing and Materials, Philadelphia, Pa.

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

however, little guidance for calculating the stresses induced by integral attachments.

The most widely used analytical methods for calculating stresses in pipe and cylindrical pressure vessels due to integral attachments are based on the work of Bijlaard¹⁰⁻¹⁴ during the 1950's, which was summarized in Welding Research Council Bulletin No. 107 by Wichman, Hopper, and Mershon¹⁵ in 1965. A small amount of experimental data is also available. Thus sufficient background information exists for developing stress indices for direct use in the piping codes and standards.

A series of parameter studies was conducted under the ORNL piping program in which a slightly modified form of Bijlaard's equations was used to develop stress indices for the simplified design procedures of the nuclear piping codes and standards. Secondary stress indices have now been developed for both round and rectangular attachments welded to the pipe, in-plane and out-of-plane bending moments, and radial thrust loads.¹⁶ The stress indices are given as simple but conservative equations of best fit that can be included directly in the codes. Currently both primary and peak stress indices for lugs are being investigated, and a proposed set of design rules is being drafted for transmittal to the code committees.

Concentric and Eccentric Reducers

Stress indices and flexibility factors were developed¹⁷ for butt-welding reducers that meet the requirements of

standards ASA B16.9 or MSS-SP48 (refs. 18 and 19). Since neither of these standards specifies the actual shape of piping components, but rather only overall length requirements and dimensional tolerances at the welding ends, several manufacturers were asked to provide design dimensions for their reducers. Dimensional data were received from seven manufacturers, and these were used as a basis for outlining the analytical parameter studies.

The dimensional data received showed that although there are wide variations in the shapes of reducers, conservative and realistic stress indices for concentric reducers could be developed by using the three different analytical models sketched in Fig. 1. The first two models are composed of five basic shell segments: the two end cylinders representing the pipe, the conical reducing section, and two toroidal transition segments, each of which has a radius of curvature equal to one-tenth the outside diameter of the large end of the reducer. The first model has a constant-wall-thickness cone, whereas the second model has a variable-wall-thickness cone that is most representative of reducers that are hot-formed from a segment of constant-wall-thickness pipe. The third model is similar to the first, except for the absence of transition segments, and represents a limiting design condition. The end cylinders have the same wall thicknesses, t_1 and t_2 , as pipe designed for the same internal pressure; thus

$$t_2 = (R_2/R_1)t_1,$$

where R_1 and R_2 are the outside radii of the large and small ends, respectively.

Variables of interest in the parameter studies were the cone angle α , which ranged from 15 to 60°; the outside diameter ratio D_2/D_1 , which ranged from 0.25 to 0.9; and the diameter-to-thickness ratio D_1/t_1 , which ranged from 10 to 160. Parameter studies for internal pressure and external moment loadings were carried out at Battelle Memorial Institute by using a proprietary computer code called MQLSA developed by Kalins.²⁰

10. P. P. Bijlaard, "Stresses from Local Loadings in Cylindrical Pressure Vessels," *Trans. ASME*, 77(6): 805-816 (1955).

11. P. P. Bijlaard, "Stresses from Radial Loads in Cylindrical Pressure Vessels," *The Welding Journal Research Supplement*, 33(12): 615-S-623-S (1954).

12. P. P. Bijlaard, "Stresses from Radial Loads and External Moments in Cylindrical Pressure Vessels," *The Welding Journal Research Supplement*, 34(12): 608-S-617-S (1955).

13. P. P. Bijlaard, *Additional Data on Stresses in Cylindrical Shells Under Local Loadings*, Welding Research Council Bulletin No. 50, pp. 10-50, New York, 1959.

14. P. P. Bijlaard and E. T. Cranch, *Interpretive Commentary on the Application of Theory to Experimental Results for Stresses and Deflections Due to Local Loads on Cylindrical Shells*, Welding Research Council Bulletin No. 60, pp. 1-2, New York, 1960.

15. K. R. Wichman, A. G. Hopper, and J. L. Mershon, *Local Stresses in Spherical and Cylindrical Shells Due to External Loading*, Welding Research Council Bulletin No. 107, New York, 1965.

16. W. G. Dodge, *Secondary Stress Indices for Integral Structural Attachments to Straight Pipe* (to be published).

17. E. C. Rodabaugh, *Stress Indices and Flexibility Factors for Reducers* (to be published).

18. ASA Standard, *Wrought Steel Butt Welding Fittings (ISO R-285)*, ASA B16.9-1964, American Standards Institute, New York.

19. MSS SP48-1969, *Steel Butt-Welding Fittings (26-in. and Larger)*, Manufacturers Standardization Society of the Valve and Fittings Industry, Arlington, Virginia.

20. A. Kalins, "Analysis of Shells of Revolution Subjected to Symmetrical and Nonsymmetrical Loads," *ASME J. Appl. Mech.*, September 1964.

Fig. 1. Schematic Diagrams of Three Analytical Models for Parameter Studies of Concentric Reducers Loaded with Internal Pressure or External Moments.

The results of the parameter study showed that maximum stress intensities always occurred at either location A or location B indicated in Fig. 1. At location A the maximum stress was in the circumferential direction, and at B the maximum stress was in the axial direction. As might be expected from the relatively large number of variables investigated, no simple relationship was found for representing the maximum stress intensities. Simple conservative equations, however, were obtained for approximate correlations of the stress indices. In some cases these equations are more conservative than necessary to assure structural integrity, but in the interest of simplicity, they will be proposed to the code committees. If, in certain design applications, the proposed stress indices are too restrictive, the designer can perform a detailed stress analysis by using one of the many available computer codes.

The more difficult problem of analyzing eccentric reducers was also investigated. In this case the large variations in shapes used and the limits on available analytical tools precluded the possibility of doing realistic parameter studies. In addition, it was not possible to find sufficient experimental data to use as a basis for developing stress indices empirically. It thus appears that if eccentric reducers are to be used to any extent in nuclear piping systems, greater standardization in manufacture and analytical and experimental development work are needed before stress indices can be developed for code use.

Thin-Walled Tees and Branch Connections

Studies of thin-walled tees and branch connections were combined because the configurations are similar. Both tees and branch connections consist of a straight section, or run, of pipe and a branch pipe that intersects the run. 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 (idealized cylinder-to-cylinder intersections). The studies are being conducted primarily for obtaining basic information for developing and verifying analytical methods. A considerable amount of information has already been developed for intersecting cylindrical shells because of their use in pressure-vessel nozzle design where the primary loading of interest is internal pressure. However, relatively little information is available for applied force and moment loadings, especially when such loads are applied on the run. The branch connection studies refer to parameter studies being carried out in response to the PVRC request (see ref. 1, Subcommittee Task 3) to obtain additional information and stress indices for branch connections designed and fabricated in accordance with the rules of the nuclear piping code USAS B31.7. Studies on both types of components are being conducted under the ORNL piping program in cooperation with other AEC-supported programs at ORNL, which include investigations on this type of structure.

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

Both experimental and analytical studies are being conducted on the series of four carbon-steel thin-walled cylinder-to-cylinder models listed in Table 1. The models are thin-shell structures consisting of two circular cylindrical shells intersecting at right angles. There are no transition regions, reinforcements, or fillets at the junction; and the models are as geometrically perfect as could be obtained. Each of the models in this series is to be tested and analyzed for 13 different loading conditions, including internal pressure and each of three mutually perpendicular bending moment loads applied to both the branch end and the run end of the model, and three mutually perpendicular force loads applied to the branch and run ends. Experimental studies of the first model and comparisons with finite-element analytical results for the internal pressure loading were completed and reported last year (see ref. 4, pp. 127-136).

During the past year the remaining studies for model 1 were completed, and a report containing all the results was written.²¹ In addition, models 2 and 3 were fabricated. Model 3 was instrumented with 165 three-gage strain rosettes, each having an individual gage length of 0.030 in., and tested with 10 of the 13 scheduled loading conditions. The three remaining tests on model 3 will be completed early in 1971, and the

results from all the loadings will be compared with the finite-element analytical study results. The tests remaining are those corresponding to the application of out-of-plane force loadings on the run and branch and in-plane force loadings on the run.

In the case of model 1, two diametrically opposite quadrants of the model were instrumented on both the inner and outer surfaces as shown in Fig. 2. Although there were small variations that had noticeable influences on the stress distributions, the experimental results for this model were very good and constitute an especially significant set of data for use in the development of analytical methods for treating intersecting-cylinder structural problems. These data were sent to several other researchers for use in comparative studies.

The finite-element analytical studies of model 1 conducted with the use of the computer program JOINT²²⁻²⁴ showed reasonably good overall agreement with the experimental results, and the method used is considered satisfactory for most engineering purposes. In all but one loading case the maximum stresses occurred on the outer surface of the nozzle at the junction of the nozzle and cylinder. The exception was for the out-of-plane force loading on the nozzle, where the maximum stresses were on the inner surface rather

21. J. M. Corum, S. E. Bolt, W. L. Greenstreet, and E. C. Gwaltney, *Theoretical and Experimental Stress Analysis of ORNL Thin-Shell Cylinder-to-Cylinder Model No. 1*, USAEC Report ORNL-4553, Oak Ridge National Laboratory (to be published).

22. C. P. Johnson and P. G. Smith, *A Computer Program for the Analysis of Thin Shells*, Report No. 69-5, Department of Civil Engineering, University of California, Berkeley, August 1967.

23. Ojars Geste, *A Computer Program for the Analysis of Tubular K Joints*, Report No. 69-19, University of California, Berkeley, November 1969.

24. Ojars Geste, *Finite-Element Analysis of Tubular K Joints*, Report No. UCSESM 70-11, University of California, Berkeley, June 1970.

Table 1. 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 will be made by boring out the nozzle of model 3.

Fig. 2. Strain-Gage Layout for Thin-Shell Cylinder-to-Cylinder Model 1 (Inner and Outer Surfaces).

than the outer surface. However, the results for the two surfaces differed only by a small percentage. As would be expected, the in-plane and out-of-plane transverse force loadings on both the branch and run gave results that were quite similar to the results obtained for the corresponding moment loadings. The axial force loadings gave rather high maximum stresses when the load was applied on the branch; the agreement between the experimental and finite-element results was excellent in this case. When the axial load was applied to the run, however, the maximum stresses were low, and agreement between experimental and analytical results was rather poor.

A concise summary of the results from the experimental and analytical studies for model 1 is given in Table 2. The angular locations of the maximum stresses correspond with the gage lines shown in Fig. 2. The stress ratios shown are the maximum principal stress values normalized to a nominal stress, where the nominal stress was taken as pr/t for internal pressure loading, M/Z for bending and transverse force loadings, and F/A for the axial force loadings, where p is the internal pressure, r/t is the radius-to-thickness ratio, M is the nominal bending moment at the intersection, F is the applied axial force, and A is the cross-sectional area of the branch or run.

Table 2. Summary of Maximum Principal Stress Ratios and Locations

Loading Case	Experimentally Determined Maximum Stress ^a		Theoretical Maximum Stress		Overall Agreement Between Theory and Experiment
	Stress Ratio ^b	Location ^c	Stress Ratio ^b	Location ^c	
Internal pressure	13.3	Nozzle outer surface, 180°	10.2	Nozzle outer surface, 0°	Good qualitative, bad quantitative
M_{XN} , out-of-plane moment on nozzle	35.3	Nozzle outer surface, 270°	37.2	Nozzle outer surface, 270°	Excellent
M_{YN} , torsional moment on nozzle	12.5	Nozzle outer surface, 120°	5.1	Nozzle outer surface, 300°	Fair
M_{ZN} , in-plane moment on nozzle	10.0	Nozzle outer surface, 300°	10.9	Nozzle outer surface 300°	Excellent
F_{XN} , in-plane force on nozzle	10.5	Nozzle outer surface, 120°	11.4	Nozzle outer surface, 330°	Excellent
F_{YN} , axial force on nozzle	36.3	Nozzle outer surface, 270°	36.0	Nozzle outer surface, 270°	Excellent
F_{ZN} , out-of-plane force on nozzle	36.0	Nozzle inner surface, 270°	32.6	Nozzle inner surface, 270°	Good
M_{XC} , torsional moment on cylinder	10.0	Nozzle outer surface, 120°	13.0	Nozzle outer surface, 120°	Good
M_{YC} , out-of-plane moment on cylinder	2.3	Nozzle outer surface, 90°	2.7	Nozzle outer surface, 0°	Fair
M_{ZC} , in-plane moment on cylinder	3.8	Nozzle outer surface, 90°	5.7	Nozzle outer surface 270°	Fair
F_{XC} , axial force on cylinder	4.7	Nozzle outer surface, 270°	5.6	Nozzle outer surface 270°	Good qualitative, bad quantitative
F_{YC} , in-plane force on cylinder	6.0	Nozzle outer surface, 270°	5.7	Nozzle outer surface, 270°	Excellent
F_{ZC} , out-of-plane force on cylinder	3.1	Nozzle outer surface, 300°	3.5	Nozzle outer surface, 300°	Good

^aBased on extrapolation of experimental distributions to maximum at junction.

^bRatio of maximum absolute principal stress value to nominal stress value.

^cMaximums all occurred at the junction.

The comparative ratings given in the last column of the table are an appraisal of overall qualitative agreement and quantitative maximum stress agreement. Generally the agreement was best for those cases involving loadings on the nozzle, with the exception of the torsional loading. This disagreement is apparently due to a restriction in the finite-element computer program concerning the number of degrees of freedom for the element deformation model. A more complete discussion is contained in ref. 21.

Small Branch Connections

In addition to the studies of intersecting cylindrical shells discussed above, the program plan calls for analytical studies of a series of branch-connection models with diameter ratios d/D less than one-half. These studies are to provide information needed to supplement the existing experimental data used as a basis for the stress indices incorporated into the 1969 issue of USAS standard B31.7 (see ref. 25).

The studies are to include detailed finite-element stress analyses of four models without reinforcement at the junction and the same models with fillet reinforcement. Pertinent dimensionless parameters are listed in Table 3. The particular set of model dimensions given was chosen to correspond with photoelastic model studies conducted by Leven for the case of internal pressure loading.²⁶⁻²⁸ Each of the eight models in the series is to be analyzed for internal pressure loading and 12 externally applied force and moment loadings: three moments and forces on the branch and run ends of the model with one of the run ends held fixed.

The first four models listed, those without reinforcement, were analyzed during the past year by using the finite-element computer code JOINT at Auburn University and at Oak Ridge National Laboratory. In order

Table 3. Branch-Connection Models for Analytical Parameter Study

Model No.	Dimensionless Parameters ^a				
	D/T	d/D	t/T	s/S^b	r/T^c
1	59.4	0.115	0.235	0.49	0
2	20.0	0.02	0.02	1.00	0
3	20.0	0.08	0.473	0.169	0
4	20.0	0.32	1.00	0.322	0
1F	59.4	0.115	0.235	0.49	0.60
2F	20.0	0.02	0.02	1.00	1.00
3F	20.0	0.08	0.08	1.00	1.46
4F	20.0	0.32	0.32	1.00	3.34

^a D is the outside diameter of the run pipe; d is the outside diameter of the branch pipe; T is the wall thickness of the run pipe; and t is the wall thickness of the branch pipe.

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

^cThe dimensionless parameter r/T is the ratio of the reinforcement fillet radius to the wall thickness of the run pipe.

to establish practical guidance for setting up the calculations and to establish a measure of confidence in the results, comparative studies were made between finite-element calculations for different sized elements and experimental photoelastic results for model 1 (Westinghouse model No. WC-2AY, ref. 28) loaded with internal pressure.²⁹ Results obtained from two of the finite-element calculations are shown in Fig. 3, along with the photoelastic data for two sections of the model. The normalized stress ratios σ_n/S and σ_θ/S shown in Fig. 3(a) and (b) correspond to the normal and longitudinal directions for the 0° or longitudinal plane of the model; similar stress ratios for the 90° or transverse plane of the model are shown in Fig. 3(c) and (d) (see Fig. 2 for reference).

As Fig. 3 indicates, both sets of finite-element results agree fairly well with the photoelastic results, except for the maximum values, which occur at the junction. This is a consequence of the thin-shell formulations describing the behavior of the finite elements, and the agreement is not substantially improved by increasing the number of elements used in the calculations; 504 elements were used in the medium-fine mesh, and approximately twice this number were used in the fine mesh. The inability to

25. E. C. Rodabaugh, *Phase Report No. 115-1 on Stress Indices for Small Branch Connections with External Loadings*, USAEC Report ORNL-TM-3014, Oak Ridge National Laboratory, August 1970.

26. M. M. Leven, *Photoelastic Determination of the Stresses in Reinforced Openings in Pressure Vessels*, Welding Research Council Bulletin No. 113, April 1966.

27. M. M. Leven, *Photoelastic Determination of Stresses at an Opening in a Thin-Wall Cylindrical Pressure Vessel*, Report No. 67-9D7-PHOTO-R1, Westinghouse Research Laboratories, August 1967.

28. M. M. Leven, *Photoelastic Determination of Stresses in a Reinforced Steam Connecting Connection with a Thermal Sleeve*, Report No. 69-9D7-PHOTO-R1, Westinghouse Research Laboratories, Jan. 14, 1969.

29. N. Krishnamurthy, *Comparison of Results from Finite-Element Analysis and Photoelastic Tests of a Cylinder-Nozzle Attachment* (to be published).

(a) STRESSES NORMAL TO SECTION

LONGITUDINAL SECTION (0° SLICE)

(b) STRESSES IN THE PLANE OF SECTION

(c) STRESSES NORMAL TO SECTION

TRANSVERSE SECTION (270° SLICE)

(d) STRESSES IN THE PLANE OF SECTION

Fig. 3. Comparison of Finite-Element Analysis and Experimental Photoelastic Stresses for a Small Branch Connection (Westinghouse Model WC-2AY) Loaded with Internal Pressure.

Table 4. Maximum Stresses for Model 1 (WC-2AY) Loaded with Internal Pressure

	Quantity	Stress Ratios		Location of Maximum
		Magnitude	Conservatism ^a	
Experimental (photoelastic results)	σ_p/S	2.96	1.00	Inner surface, 0° plane
Medium-fine-mesh finite-element analysis	σ_p/S	4.76	1.61	Outer surface, 0° plane
Fine-mesh finite-element analysis	σ_p/S	3.8	1.28	Outer surface, 0° plane

^aDefined as the ratio of the calculated maximum stress to the experimental stress.

account for the fillet on the outside surface in the analyses appears to have significant consequences. The maximum stresses are compared in Table 4, where it is clearly evident that conservative values for maximum stresses were calculated.

Structural Analyses of Elbows

Piping components being studied under this general heading are curved or bent pipe, welding elbows that meet the requirements of standards ASA B16.9 or ASA B16.28 (refs. 18 and 30) for butt-welding fittings, and miter bends fabricated in accordance with the rules of the nuclear piping code USAS B31.7 (ref. 9). No distinction is normally made between curved pipe and elbows in the calculation of elastic stresses because of the assumed ideal shape of the component. However, present manufacturing practices assure much better control over the shape of welding-end elbows than that of bent pipe. The latter has not only the dimensional variations of commercially available pipe but also dimensional variations such as ovaling of the cross section caused by the bending operation. Thus one of the studies of elbows (machined-model studies) was planned specifically to determine the influence of dimensional variations on the elastic stresses. These studies are described below.

In addition to a discussion of machined-elbow studies, this section contains a discussion and summary of results of experimental limit-load studies of carbon steel and stainless steel elbows. Analytical studies of the effects of pipe welded to elbows (end-effect studies) and the development of theoretical solutions and stress indices for miters were continued but at a much reduced level from that reported last year. Both these studies are important and will be resumed and reported when funding levels permit their conclusion.

Machined-Elbow Model Studies

Experimental and analytical elastic-response studies are to be conducted on the four machined-elbow test models listed in Table 5. All the models are 90°-bend long-radius elbows (bend radius equal to three times the pipe radius), with nominal outside diameters of 10.75 in., which are to be welded to matching pipe stubs approximately two pipe diameters long on each end. The first model, ME-1, is designated an ideal torus with a uniform wall thickness of 0.365 in.; each of the other three models has well-defined geometric deviations from this basic shape. The elastic response of each model will be studied for internal pressure loading; for in-plane, out-of-plane, and torsional moments; and combinations of moment loadings and internal pressure.

The experimental models will each be instrumented with approximately 80 three-gage strain rosettes on lines around the circumference and along the length of the model on both the inner and outer surfaces. About one-third of the gages will be located on the inner surface, and the pipe stubs will be gaged, as well as the elbows. Deflection gages will also be used to measure model deflections to obtain data for determining flexibility factors.

Elastic-response data from the first model, ME-1, will be used as the standard base-line data for comparison with the results from the other models and for determination of end effects caused by interaction with the attached pipe. The effects of out-of-roundness and nonuniform wall thickness will be determined by comparing the results from the second and third models, respectively, with the results from the first model. The fourth model, ME-4, is included primarily to test the assumption of superposition of stresses for nonlinear geometry changes. This assumption is inherent in simplified design procedures but cannot be supported on the basis of theoretical results alone. The assumption may, however, be reasonably accurate in the range where geometric deviations are not large.

During the past year all four elbows were machined, dimensionally inspected, and accepted from the vendor.

30. ASA Standard Wrought Steel Butt Welding Short Radius Elbows and Returns, ASA B16.28-1964, The American Society of Mechanical Engineers, New York.

Table 5. 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 may 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.

Since the only available manufacturer could not fabricate the specimens to a tolerance of less than ± 0.032 in. on the wall thickness, the models are not as dimensionally perfect as desired, but they represent the best that could be obtained.

The test assembly for the first model, ME-1, was fabricated. The model was strain-gaged on both the inner and outer surfaces with a total of 83 three-gage strain rosettes, and then it was mounted in the loading frame. Two sizes of 45° rosettes were used, 0.062- and 0.125-in. gage length; 67 are on the outer surface and 16 are on the inner surface. Figure 4 shows the test assembly prior to mounting in the loading frame.

Analytical studies are also planned for the four 10 in. machined-elbow models. Finite-element elastic analyses will be made for each of the models with a modified version of the thin-shell finite-element computer program JOINT. The modifications being made will also allow use of the program for studies of the effects of a wider range of dimensional variations than those in the planned test program. This computer program should be operational by the time the experimental studies of the first model are completed.

Limit-Load Studies for ASA Standard Elbows

Load-deflection responses were determined experimentally for a series of 14 commercial 6-in. (nominal) carbon steel elbows and one 6-in. stainless steel elbow. The results were interpreted in terms of limit analysis concepts and used to determine the so-called collapse loads. A paper giving the details of the work was written and submitted for publication.³¹

The simplified design procedure of USAS standard B31.7 (ref. 9) for primary stresses in piping components is based on satisfaction of the equation

$$B_1 \frac{pD_o}{2t} + B_2 \frac{D_o}{2I} M_i \leq 1.5S_m,$$

where S_m is the allowable stress intensity, p and M_i are internal pressure and applied moment loading, respectively, $D_o/2t$ and $D_o/2I$ are normalizing factors based on stresses in straight pipe, and B_1 and B_2 are stress indices based on plastic limit loads.

Design values for both the primary stress indices B_1 and B_2 , are contained in the piping code USAS B31.7-1969 for elbows manufactured in accordance with the ASA standards, and the B_1 values are supported by the pressure-burst requirement of the standards. The code values for B_2 were estimated from elastic analyses because of the lack of either detailed analytical limit-load studies or experimental collapse data. Although the code value of $B_2 = 0.75C_2$, where C_2 is the secondary bending stress index, was believed to be conservative, a series of experimental plastic collapse studies was undertaken as part of the ORNL piping program to provide more direct information for establishing design values of the B_2 stress index.

Plastic collapse loads (limit loads) were experimentally determined for the 15 commercial 6-in.-nominal-diameter elbows for butt welding listed in

31. S. E. Bolt and W. L. Greenstreet, *Experimental Determination of Plastic Collapse Loads for Pipe Elbows* (to be published).

Fig. 4. Test Assembly for Experimental Stress Analysis of a 10-in. Uniform-Wall-Thickness Machined Elbow, Model ME-1.

Table 6. 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 and stainless steel elbows. Fourteen of the elbows were made from ASTM A-106, grade B, carbon steel and one, PE-15, from ASTM type 304L stainless steel. All the elbows had a bend angle of 90° .

The test assemblies were made by welding pipe extensions of the same type of 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, and a single normal force loading was applied near the end of the free extension four to five pipe diameters from the elbow. This produced essentially only a moment loading on the elbow. Five of the sched-40 carbon steel elbows were also tested with an internal pressure of 1500 psi, which was held constant throughout the test. Two dial indicators, $7\frac{1}{4}$

and 15 in., respectively, from the plane of the pipe-to-elbow weld, were mounted on a frame attached to the loading frame pedestal to measure deflections as a function of the applied force. In some cases strain gages were also mounted on the elbow in the region of expected high stresses. In all cases the specimens were loaded with a maximum external force of sufficient magnitude to produce predominantly plastic response. Figure 5 shows the stainless steel specimen, PE-15, during the test. The dial indicators, hydraulic ram, and load cell can also be seen.

The load-deflection response curves for each test specimen were then used to determine numerical values for the limit loads. In all but one case the experimentally determined limit load indicated that the code value for the stress index B_2 is perhaps more conservative than necessary. The one exception was the stainless steel elbow PE-15 for which the collapse load was about one-half that for the comparable carbon steel elbow

Fig. 5. Limit-Load Test of a 6-in. Long-Radius Sch40 Stainless Steel Elbow, Specimen PE-15.

PE-2. Load-deflection curves obtained for the two elbows, PE-2 and PE-15, are shown in Fig. 6. The two sets of curves are for the No. 1 and No. 2 dial indicators shown in Fig. 5, both of which were in the same

position for tests on each elbow. The small ν on each load-deflection curve marks the experimental collapse load. These two tests in the series are especially significant because the B_2 stress index values currently

Table 6. Plastic Limit-Load Tests on 6-in. Elbows

Elbow			Loading Conditions			
Designation ^a	Wall Thickness (sched)	Bend Radius (in.)	Moment		M_y Out-of-Plane	Pressure
			In-Plane ^b			
			$+M_z$	$-M_z$		
PE-1	40	9	X			
PE-2	40	9		X		
PE-3	40	9			X	
PE-4	40	9	X			X
PE-5	40	9		X		X
PE-6	40	9			X	X
PE-7	80	9	X			
PE-8	80	9		X		
PE-9	80	9			X	
PE-10	40	6	X			
PE-11	40	6		X		
PE-12	40	6			X	
PE-13	40	6	X			X
PE-14	40	6			X	X
PE-15 ^c	40	9		X		

^aThe letters PE stand for plastic collapse, elbow.

^bA positive in-plane moment ($+M_z$) causes the elbow to open; a negative in-plane moment ($-M_z$) causes the elbow to close.

^cType 304L stainless steel elbow.

Fig. 6. Corresponding Load-Deflection Curves from Limit-Load Tests of Carbon Steel and Stainless Steel Elbows with an Applied In-Plane Bending Moment.

in the piping code are not functions of the material; they are functions only of the dimensions of the component.

Further experimental confirmation of this apparent difference in the behavior of carbon steel and stainless steel elbows is needed before firm recommendations can be made to the code groups.

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 SP48 standards.^{18,19} The ratio of the diameter of the branch pipe to that of the run pipe is normally greater than one-half but may be as small as one-third. These fittings are characterized by a smooth transition between the branch and run and are normally forged as a single unit. The wall thickness is large in comparison with the diameter, so thin-shell theory is not expected to apply.

Essentially no information was available in the literature that could be applied directly in the development of stress indices and flexibility factors for these fittings. Consequently, at the request of the code committees and the PVRC, an extensive series of experimental and analytical investigations of standard tees was included in the ORNL piping program. Commercially available tee-joint fittings sold under registered trade names are not being studied in this program.

Various experimental and analytical studies are being conducted with the series of 15 tees listed in Table 7 as models. These tees represent the standard, or normal, product of three of the major suppliers of pipe fittings and include both full outlet and reducing tees ranging in nominal size from 6 to 24 in. The nominal wall thicknesses of sched 40, 80, and 160 span the range of normal usage in water-cooled nuclear power plants, and two materials, A-106, grade B, carbon steel and type 304L stainless steel, are represented.

Table 7. Descriptions of ASA B16.9 Tees for Experimental and Analytical Studies

Tee	Identification ^a					Experimental Tests ^b				Analytical Studies ^b		
	Run (in.)	Branch (in.)	Wall (sched)	Steel ^c	Mfr.	5.1.1 ^d	5.1.2 ^e	5.2 ^f	5.3 ^g	5.4 ^h	5.5 ⁱ	5.6 ^j
T-1	6	6	40	S	I	1B	1M				4B-F	
T-2	6	6	160	C	I	1B	1P				4B-F	7ΔT
T-3	6	6	40	S	I	1B	1M				4B-F	
T-4	12	12	80	C	I	1A	1M				4B-F	
T-5	12	12	80	C	II	1B	1P				4B-F	
T-6	12	12	80	C	III	1A	1M	8B-M	8A-T	4A-F	4B-F	
T-7	12	12	160	S	II	1A	1M	8B-P	8A-T	4A-F	4B-F	
T-8	12	6	40	S	II	1A	1M	8B-M	8A-T	4A-F	4B-F	7ΔT
T-9	12	6	160	S	II	1A	1P	8B-P			4B-F	
T-10	24	24	40	C	III	1B	1M				4B-F	
T-11	24	24	160	C	III	1B	1P				4B-F	7ΔT
T-12	24	10	40	C	III	1B	1M				4B-F	
T-13	24	10	160	C	III	1B	1P				4B-F	7ΔT
T-14	12	6	40	S	III	1B	1U				4B-F	
T-15	12	6	40	S	I	1A	1M				4B-F	

^aDimensions are nominal.

^bNumerical headings indicate subtask numbers in the ORNL piping program.

^cS = type 304L stainless steel; C = type A-106, grade B, carbon steel.

^dElastic response tests: 1A indicates highly instrumented test for determination of stress distributions and intensities; 1B, test with fewer strain gages to measure maximum stress intensities.

^eLow-cycle fatigue-to-failure tests; loadings: M indicates moment; P, internal pressure; U, as yet undecided.

^fPhotoelastic stress analyses; loadings: P indicates internal pressure; M, out-of-plane bending moment applied to the branch.

^gThermal stress tests.

^hDevelopment of finite-element computer code and stress analyses.

ⁱDetailed finite-element elastic analyses.

^jAnalyses of the response of mixing tees to step and monotonic changes in fluid temperatures in branch and run.

Planned experimental studies include elastic-response tests for 13 loadings, including internal pressure and applied bending moment and thrust loads on the branch and run pipe extensions, low-cycle fatigue-to-failure tests, photoelastic model studies, and thermal stress tests. 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 data from three of the tees. The computer code will have thermal stress capabilities and will be used to analyze all 15 tees listed in the table. Analytical studies have also been outlined for mixing tees, as requested by the PVRC (see ref. 1), but are currently not a part of the ORNL program.

At present, portions of the B16.9 tee studies are being conducted at Oak Ridge National Laboratory and under subcontract at the University of California, Southwest Research Institute, Combustion Engineering, Inc., and Westinghouse Research Laboratories. Accomplishments during the past year for each of these studies are discussed below.

Elastic-Response Tests

The elastic-response tests have been divided into two groups, phase A and phase B (see Table 7, seventh column and footnote d), with the amount of instrumentation required being the major difference. The objective of the phase A series is to determine both the general stress distributions over the body of the tee and the locations and magnitudes of the maximum stresses caused by each of the 13 different loadings. The objective of the phase B series is primarily to obtain maximum stress data after the expected locations have been identified from the phase A tests. Fewer strain gages and perhaps fewer loading conditions will be used for the phase B tests. Data from the phase A tests will be used for comparison of otherwise identical tees made by different manufacturers, comparison with the photoelastic model studies, and development of the finite-element computer code. Data from both series of tests, that is, from all 15 tees, will be used to develop empirical correlations that can be used for piping system design purposes.

Since the majority of the tests are designated phase B and since the phase A test results are needed to identify the number and locations of strain gages required to measure the maximum stresses, it follows that the phase A tests should be completed before the phase B tests are started. However, in order to complete the entire test series in a reasonable amount of time, subcontracts were written for the experimental stress analysis (and

low-cycle fatigue tests) of nine of the tees listed in Table 7, each of which was to be fully instrumented with approximately 225 three-gage strain rosettes, with about one-half the gages on the inner surface. Since the largest fraction of the cost is related to fabrication of the test assembly and to setup and operation of the tests, the total cost of each test is not appreciably influenced by the number of strain gages. The tees currently being experimentally analyzed are the five 12-in. tees (T-4, T-6, T-7, T-8, and T-15 at Southwest Research Institute) and the four 24-in. tees (T-10 through T-13 at Combustion Engineering, Inc.).

During the past year elastic-response tests were completed for all five of the 12-in. tees being examined at Southwest Research Institute. Each tee assembly was instrumented with approximately 225 three-gage strain rosettes, 0.062-in. gage length, on the body of the tee and a minimum of 16 dial indicators to measure pipe deflections. Strain gages were placed on two diametrically opposite quadrants of each tee along lines at 0, 22½, 45, 67½, and 90° from the axial plane of the tee. Matching gages were placed on both the inner and outer surfaces, and additional gages were positioned in the crotch of the tee. The strain gages on the 0 to 90° quadrant of tee T-4, which is a 12-in. full-outlet sched-80 carbon steel tee, are shown in Fig. 7. Some of the lead wires to the gages on the opposite quadrant may also be seen in the photograph.

For each of the 13 loading cases, strain-gage data were obtained for two complete loading cycles at nine loading positions for each cycle — zero load, four loading positions, and four unloading positions. Stresses were calculated for each rosette site, and the numerical data were punched on IBM cards and transmitted to ORNL for evaluation. For each tee tested there were over 150,000 individual strain readings to be evaluated.

The data were examined at ORNL for accuracy and consistency with a numerical evaluation scheme developed from the hypothesis that the strain-gage readings should be proportional to the loads in a linear-elastic test. Much care was taken during the tests to assure that the maximum strains never exceeded the elastic range of materials behavior. An ORNL data-evaluation computer code, called NQSEY,³² was written that computes a linear least-squares fit to the strain data for each gage as a function of the recorded

32. W. G. Dodge and J. E. Smith, "A Diagnostic Procedure for the Evaluation of Strain Data from a Linear Elastic Test," paper to be presented at the Spring Meeting, Society for Experimental Stress Analysis, Salt Lake City, Utah, May 18-21, 1971.

Fig. 7. Outer Surface Strain Gages in the 0 to 90° Quadrant for the Elastic-Response Tests of a 12 x 12 x 12-in. Sched-80 Carbon Steel ASA B16.9 Tee, Specimen T-4.

load, flags the data points that differ from the fitted line by more than a predetermined amount, makes certain other diagnostic checks of the data, and calculates normalized stresses based on the best determined strain-load relationship. Also, curves of the calculated stresses as a function of gage position are plotted for further evaluation.

Most of the data obtained from the SwRI tee tests passed the diagnostic evaluations with little difficulty, indicating that the experimentally determined stresses were reliable. However, the data from several of the loading cases showed distinct departures from linear behavior for a large percentage of the gages; the torsional moment loadings appeared to be the worst. The trouble was finally traced to a small unbalance in

the loading system, and Southwest Research Institute was asked to rerun six loading cases on three of the tees. The reruns were completed, and the data were evaluated and accepted for two of the tees. The last tee, T-6, was reinstalled in the loading frame, and the reruns should be completed within the next few weeks.

During the coming year, the elastic-response data from the five 12-in. tees tested at Southwest Research Institute and the data from the 24-in. tee tests discussed below will be used to develop stress indices that can be used with simplified design procedures. A series of reports will also be written to describe the tests and document the results.

Elastic-response tests for one of the 24-in. ASA B16.9 tees (T-10) being examined at Combustion Engineering,

Inc., at Chattanooga, Tennessee, were completed during the past year; and the second tee, T-11, was partially instrumented. Tee assemblies for the other two 24-in. tees, T-12 and T-13, were fabricated, and the tees will be tested during fiscal year 1972. All four tests were to have been completed during FY-1971, but available funds were not sufficient. The test assemblies are presently being held in storage. Results of tests of T-11 will be obtained before July 1971. A paper³³ describing the experimental stress analysis of T-10 was prepared, and the complete set of results will be issued as an ORNL report.

The elastic-response tests for T-10 were similar to the tests described above for the 12-in. tees. Before the test assembly was fabricated, plaster-of-paris casts were made of both the inner and outer surfaces in order to retain a permanent record of the shape and dimensions of the tee, which would be destroyed during the fatigue test. Strain-gage positions were established on the tee prior to welding the pipe extensions, and three-dimensional coordinate positions were measured for each strain rosette relative to the intersection of the center lines of the branch and run outlets. Pipe extensions, 66 in. long, with hemispherical end caps, of SA-106, grade B, carbon steel were welded to each of the run outlets by using the gas-tungsten-arc welding method. Both the welders and welding procedures were qualified in accordance with Section IX of the ASME *Boiler and Pressure Vessel Code*. After all the welds passed radiographs, dye penetrant, and metallographic examinations, the tee assembly and branch piping assembly were stress relieved at 1150°F.

Approximately 230 rectangular rosette strain gages with 0.125-in. gage lengths were installed on the tee assembly in two diametrically opposite quadrants with matching gages on the inner and outer surfaces. After the inner surface was instrumented the branch pipe was welded to the tee and the test assembly was positioned in the loading frame. Figure 8 shows the assembly in the loading frame with the outer surface instrumentation in place. Four of the six LVDT deflection-measuring devices used to obtain data for determining flexibility factors and the supporting hardware may also be seen.

Tee T-10 was tested with 12 different loading conditions: internal pressure, six bending moments, and

five direct-force loads. The in-plane force load on the branch pipe was omitted because of fixations in the strength of the test frame. Strain-gage data were obtained with a 2000-channel computer-controlled data-acquisition system and evaluated both at Combustion Engineering and ORNL by the diagnostic scheme described earlier. The maximum normalized stress intensities determined experimentally for the internal pressure and applied moment and force loadings are given in Table 8.

Table 8. Maximum Normalized Stress Intensities for T-10

Internal pressure, $\bar{S}_{max} = 2.8$

Branch Loading	$\bar{S}_{max}$ (s)	Run Loading	$\bar{S}_{max}$ (s)
In-plane moment	3.8	In-plane moment	3.2
In-plane force		In-plane force	3.1
Out-of-plane moment	3.7	Out-of-plane moment	1.6
Out-of-plane force	3.5	Out-of-plane force	2.0
Torsional moment	3.6	Torsional moment	3.5
Axial force	6.5	Axial force	4.3

$\bar{S}_{max}$ is the normalized value of the maximum principal stress, where the normalizing factor was M/Z for the moment loadings, F/A for the axial force loadings, and $pD_o/2t$ for the pressure loading. Z = section modulus, A = cross-sectional area, D_o = outside diameter, and t = wall thickness of the branch pipe or run pipe, respectively.

Low-Cycle Fatigue Tests

Low-cycle fatigue-to-failure tests are also scheduled for the 15 tees listed in Table 7. The fatigue tests will be run with either a cyclic internal pressure or applied moment, as indicated (eighth column and footnote e). All the moment-loading fatigue tests are to be conducted with a constant internal pressure equal to the nominal design pressure, except for T-3, which will be pressurized only enough to show the presence of a leaking crack (2 to 5 psi). The direction of the particular moment loading and whether it is to be applied on the branch pipe or the run pipe will depend on the results from the elastic-response tests. In most cases the moment loading used in the fatigue tests will correspond to the loading that gives the highest maximum stresses in the tee. The magnitude of the loading will be based on an expected fatigue failure within the range of 500 to 100,000 fully reversed controlled-displacement cycles. All the fatigue tests

33. J. K. Hayes and B. Roberts, "Experimental Stress Analysis of 24-inch B16.9 Tees," paper to be presented at the First National Congress on Pressure Vessels and Piping, American Society of Mechanical Engineers, San Francisco, May 10-12, 1971.

Fig. 8. Test Assembly Mounted in the Loading Frame for the Elastic-Response Tests of a 24 x 24 x 24-in. Sched-40 Carbon Steel ASA B16.9 Tee, Specimen T-10.

conducted thus far have been for a target failure of 7000 cycles, which is approximately equal to the log-mean value between 500 and 100,000 cycles.

Fatigue tests were conducted this year on three of the 12-in. tees - T-4, T-7, and T-8 - at Southwest Research Institute and on the 24-in. tee, T-10, at Corvusion Engineering. In these tests each tee was pressurized to the nominal design pressure, p , as determined from Eq. (2) of USAS B31.7-1969:

$$p = \frac{2S_m(t_m - a)}{D_o - 0.8(t_m - a)},$$

where S_m is the code-allowable primary stress, t_m is the minimum wall thickness of the pipe (87.5% of nominal), D_o is the nominal outside pipe diameter, and a

(0 in these tests) is additional wall thickness for corrosion allowance. Fatigue tests were run with an alternating moment load applied to the branch pipe of sufficient magnitude to produce a maximum equivalent elastic stress amplitude, S_e , of $\pm 83,500$ psi at the gage site with the highest stresses in the elastic-response tests. The value of S_e was determined from Markle's relationship:³⁴

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

where N was taken as 7000 cycles.

34. A. R. C. Markle, "Fatigue Tests on Piping Components," *Trans. ASME*, 74: 402 (1951).

Since plastic deformation was expected during the fatigue test, the maximum deflection, $\delta_{\max}$, of the branch pipe (to be used as a control point) was determined experimentally by extrapolating the elastic portion of a load-deflection curve measured during the first quarter cycle. During the remainder of the test the branch pipe was cycled through a deflection of $\pm\delta_{\max}$. The results from the fatigue tests conducted to date are summarized in Table 9, and a closeup view of the through-the-wall crack for T-4 is shown in Fig. 9.

Photoelastic Model Studies

Photoelastic model studies are being conducted under subcontract at Westinghouse Research Laboratories for

Table 9. Results from Fully Reversed Fatigue Tests of ASA B16.9 Tees

Tee	Constant Internal Pressure (psi)	Direction of Applied Branch Moment ^a	Number of Cycles to Failure ^b
T-4	1925	Out of plane	2,070
T-7	3240	In plane	11,475
T-8	950	Out of plane	8,249
T-10	1025	In plane	18,532

^aMaximum alternating equivalent elastic stress intensity $S_a = 83,500$ psi.

^bTarget point failure = 7000 cycles; average failure point = 10,081 cycles.

Fig. 9. Closeup of the Crack Developed During the Low-Cycle Fatigue Test of a 12 x 12 x 12-in. Sched-80 Carbon Steel ASA B16.9 Tee, Specimen T-4.

the four tees (T-6 through T-9) identified in Table 7 (ninth column and footnote f). One-half-scale epoxy models of the 12-in. prototype tees were to be made, loaded with either internal pressure or out-of-plane bending moment as indicated in the table, and analyzed by using the frozen-stress technique. It was reported last year that two of the photoelastic test assemblies had been fabricated, but the bulk of the work was completed this year. The entire series of studies will be completed and reported³⁵ by midyear.

The principal objective of this investigation is to identify nonlinearities in the stress distributions through the wall thickness of the tees at positions of maximum surface stress. This information will provide guidance for separating the total stress components into the three categories (primary, secondary, and peak) used by the nuclear vessel and piping codes to fix allowable stress values. Surface stresses are to be determined from slices cut from one quadrant of the models at angular locations $22\frac{1}{2}^\circ$ apart around the branch opening, corresponding to strain-gage lines on the prototype elastic-response tests. At three positions on each photoelastic model, the principal stress distributions along lines through the wall thickness will be determined and used to determine (1) the average membrane stress, (2) the linear equivalent stress distribution (corresponding to the secondary bending stress), and (3) the peak stress or maximum difference between the measured stress distribution and the linear-equivalent distribution.

The present status of this work is that one-half-scale photoelastic models have been fabricated for all four tees and tested with the appropriate loading. The test models were machined from blocks of epoxy on a contour milling machine, with very carefully constructed half-scale wooden patterns used as templates. The shape and wall thickness of the wooden patterns were hand finished to incorporate all the dimensional variations of the prototype tees at the proper scale. (The prototype tees either have been or will be tested in other parts of the program.) Details of the welds at the junctions between the tee and the attached pipe were hand finished to conform with the requirements of USAS B31.7 for butt-weld alignment between components of different thicknesses. A photograph of the completed photoelastic model for T-9, ready for testing with an internal pressure, is shown in Fig. 10.

All four models have been sliced, and the surface stresses have been determined for three of the tees, T-7, T-8, and T-9. The results from the analysis of T-9 loaded with internal pressure are shown in Figs. 11 and 12, where the stresses are plotted in terms of the dimensionless ratio σ/p , stress per unit pressure. The inside diameters of this particular tee for both the run and branch outlets were machine bored, and as a result there was a sharp inside corner, as shown in the cross-sectional view of the slices. The maximum stress in the model, $\sigma/p = 13.9$, occurred at this inside corner in the 0° slice (Fig. 12). In terms of the nominal membrane stress in the pipe, $\sigma_{nom} = pD_o/2t$, where $D_o = 12.75$ in. and $t = 1.312$ in., the maximum principal stress ratio is $\sigma_{max}/\sigma_{nom} = 2.86$. Stress distributions through the wall thickness are needed in order to identify the secondary and peak components of this value before the maximum experimental stresses can be compared with stresses calculated by using the indices presently in the piping code, USAS B31.7.

Experimental Thermal Stress Analysis

In addition to the experimental studies for mechanical loadings on ASA B16.9 tees, stresses due to thermal loadings are also being investigated. Two of the tees, T-6 and T-8 (see Table 7, tenth column and footnote g), were selected for thermal stress tests at Oak Ridge National Laboratory; T-6 is carbon steel and T-8 is stainless steel. The tees actually being tested 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.

Test assemblies have been 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. One of the tees, T-6, has been instrumented with 68 high-temperature (600°F maximum temperature), 45° three-gage strain rosettes, with 0.125-in. gage lengths, on both the inner and outer surfaces, as shown in Fig. 13. Thermocouples will be attached at the same locations and at additional positions on the model to measure temperature distributions. Cooling coils and electrical heating elements will be wrapped around the ends of the pipe extensions, and the entire assembly will be insulated. The tests will be run by heating one of

35. M. M. Leven and R. L. Johnson, *Photoelastic Determination of Stresses in ASA B16.9 Tees* (to be published).

Fig. 10. One-Half-Scale Photoelastic Model of the 12 x 12 x 6-in. Sched-160 ASA B16.9 Tee, T-9, Analyzed for Internal Pressure Loading.

Fig. 11. Normalized Surface Stresses on the Transverse (90°) Section of the Photoelastic Model for ASA B16.9 Tee, T-9, Loaded with Internal Pressure.

Fig. 12. Normalized Surface Stresses on the Longitudinal (0°) Section of the Photoelastic Model for ASA B16.9 Tee, T-9, Loaded with Internal Pressure.

the pipe ends and cooling the other two to produce temperature gradients in the tee. Temperatures and strains will be measured and used to calculate the resulting thermal stresses. Because of a shortage in funds this fiscal year, the test assemblies, the test stand, and related hardware are being stored in their present condition until funds are available to continue the experiment.

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 for generating parameters over a wide range of sizes and wall thicknesses and, with these results, to provide stress indices for ASA B16.9 tees that may be included in codes and standards and used in the design of piping systems.

Included in the scope of the work being done at Berkeley is the development of a computer program specifically for the analysis of B16.9 tees under arbitrary loadings, including temperature change, internal pressure, and combinations of bending moment, torsional moment, axial force, and shear force applied through the connecting pipes. The computer program is expected to require only a small amount of data preparation and to produce stress output in an easily

Fig. 13. ASA B16.9 Tee Test Assembly (T-6) with Attached High-Temperature Strain Rosettes for Thermal Stress Tests of a 12 x 12 x 12 in. Sched-80 Carbon Steel Tee.

interpreted form. The detailed stress analysis of three of the tees, T-6, T-7, and T-8 (Table 7, eleventh column and footnote *h*), and comparison with experimental results are included as part of the proof of the analysis.

Because of the complex geometry of B16.9 tees, the finite-element method was selected as the only feasible method of analysis. Nevertheless, application of this method is not a routine matter, and a number of theoretical and practical problems must be solved before a completely satisfactory computer program can be written. These problems include the following:

1. selection of the best finite-element model from a variety of candidate elements,
2. determination of the best procedure for subdividing the structure into finite elements,
3. definition of the geometry of the structure, hopefully in terms of a small number of parameters, and development of procedures for generating the finite-element data automatically,
4. presentation of the computed stresses in a convenient form for interpretation,
5. development of an efficient computer program to permit the analyses to be carried out at reasonable cost.

Progress during the past year has been excellent. Finite-element models of both the flat-plate and three-dimensional types were considered, and a modified version of one of the existing three-dimensional elements was developed for application to the B16.9 tee problem. This element, designated ZIB8R5, belongs to

the family of isoparametric hexagonal (six-sided) elements originated by Zienkiewicz and Irons.³⁶ It has eight nodes with five additional modes of deformation that permit the edges of the element to curve in the direction normal to the shell surface but not within the shell surface. A number of example structures, including straight and curved beams, and thick and thin plates and shells were analyzed in order to compare the characteristics of the different elements. On the basis of these comparisons and preliminary calculations for the 12-in. tee tested at SwRI, it was concluded that the modified element ZIBSR5 had the best overall characteristics, combining simplicity with the ability to provide accurate results for tees with a wide variety of shapes and thicknesses.³⁷

Logarithms of the final computer program are now updating, and the remaining parts have been mapped out. The program is based on a general-purpose structural analysis program developed by Wilson,³⁸ which is exceptionally efficient and can accommodate large structures.

Comparative results of the finite-element calculations, the strain-gage data, and the photoelastic analysis of ASA B16.9 tee T-8 (2 X 12 X 6 in., sched 40) for the case of an out-of-plane bending moment on the branch pipe are shown in Figs. 14 and 15. Maximum and minimum principal stresses normalized to the nominal bending stress in the pipe, $\sigma_{nom} = M/Z$, where Z is the section modulus, are shown in Fig. 14(a) and (b), respectively, for the longitudinal section of the tee. Surface stresses for the transverse section of the tee are shown in Fig. 15. σ_x/σ_{nom} in Fig. 15(a), directed along the axis of the main pipe section, and σ_y/σ_{nom} in Fig. 15(b), directed normal to the plane of the section. In general the agreement between the calculated stresses and the strain-gage and photoelastic data is very good, although there were some errors in the photoelastic results for the stresses lying in the plane of the transverse section (270° slice). These were apparently

Fig. 14. Minimum and Maximum Principal Stresses from Finite-Element Calculations and Strain-Gage and Photoelastic Experimental Results for the Longitudinal Section of a 12 x 12 x 6-in. Sched-40 ASA B16.9 Tee (T-8) Loaded with an Out-of-Plane Bending Moment on the Branch Pipe.

due to deformations in the plastic model that occurred during the stress-freezing operation.

Currently procedures are being developed for automatically subdividing the tee structure into finite elements and generating the computer program input data. A procedure has been devised for dividing virtually any shape into well-conditioned quadrilateral elements for use in generating mesh layouts for the surfaces of the tee. This procedure is being further developed to complete the element descriptions (through-the-wall direction) and to allow for the many different sized tees that would be considered in a parameter study.

36. B. M. Irons and O. C. Zienkiewicz, "The Iso-Parametric Element System - A New Concept in Finite Element Analysis," *Proceedings, Conference on Recent Advances in Stress Analysis*, Royal Aeronautical Society, London, England, 1968.

37. G. H. Powell, R. W. Clough, and A. N. Gentayat, *Stress Analysis of B16.9 Tees by the Finite Element Method* (to be published).

38. F. L. Wilson, *SAP - A General Purpose Structural Analysis Program*, Report No. 78-20, Structural Engineering Laboratory, Department of Civil Engineering, University of California, Berkeley, September 1979.

Fig. 15. Direct Surface Stresses Along the Gage Line and Normal to the Gage Line from Finite-Element Calculations and Strain-Gage and Photoelastic Results for the Transverse Section of a 12 x 12 x 6-in. Sched-40 ASA B16.9 Tee (T-8) Loaded with an Out-of-Plane Bending Moment on the Branch Pipe.

Conclusions

Progress during this past year has been excellent, even though some adjustments in program activity were required because of the constrained budget situation. Several studies were completed, and many of the projects initiated last year are nearing completion. The literature survey and engineering study on stress analysis information and design techniques developed prior to initiation of the ORNL piping program was published. Stress indices and rules for the design analysis of straight pipe with out-of-round cross sections were developed and submitted to the code committees. Stress indices were also developed for integral attach-

ments to straight pipe, girth-welded joints, tapered transition joints, and concentric reducers. Reports covering these studies are being written.

Experimental and analytical studies on thin-walled idealized tees (cylinder-to-cylinder models) and small branch connections were continued at ORNL. Studies on the first of four steel models were completed, and a report was written; the second and third models were fabricated; and most of the experimental work on one of these was finished. A portion of the analytical finite-element studies on branch connections initiated at Auburn University last year was completed, and the balance of the work is being done at ORNL.

Experimental studies of ASA standard B16.9 tees were completed at Southwest Research Institute and Combustion Engineering, Inc., that included strain-gage tests on four 12-in. tees and one 24-in. tee and low-cycle fatigue tests of three 12-in. tees and one 24-in. tee. Most of the photoelastic stress analysis work at Westinghouse Research Laboratories on half-scale models of 12-in. tees was completed. The main portion of the finite-element computer code being developed at the University of California, Berkeley, for the analysis of ASA B16.9 tees was written. For several test-case loadings, especially good agreement was obtained between calculated stresses and the experimental data obtained at SwRI on 12-in. tees. Preparations for experimental thermal stress tests of two 12-in. ASA B16.9 tees were initiated at ORNL. Both test assemblies were fabricated, and one of the specimens was partially instrumented.

Plastic collapse (limit-load) studies of ASA standard 6-in. elbows were completed that included the examination of 14 carbon steel elbows and one type 304L stainless steel elbow. The limit load for the stainless steel elbow was considerably lower than the limit loads for the carbon steel elbows. Four machined 10-in.-nominal-diameter elbows for experimental elastic response studies of the effects of geometric distortions were received from the fabricator. The test assembly for one of these was completed and instrumented.

Project work for the coming year, as well as for the remainder of the program, was planned. In addition to completing the studies now under way, plans have been made to do limit-load studies on tees and to start the experimental stress analysis and fatigue tests of the remaining B16.9 tees during this next year.

Part II. Experimental and Analytical Investigations of Nozzles

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Experimental and analytical investigations of stresses and strains in the region of vessel penetrations and nozzle connections are being conducted to establish analytical procedures for use in the design of nuclear reactor pressure vessels. Emphasis of the program is on both development and complete experimental substantiation of analytical methods. Included are studies of single nozzles radially and nonradially attached to cylindrical and spherical shells and clusters of nozzles attached to flat plates and to cylindrical and spherical shells.

The loadings of principal interest are axial thrust forces, bending moments, and torsion, all applied to the nozzles, and internal pressure. Parameters being studied are the nozzle diameter-to-thickness ratio (d/t), the shell diameter-to-thickness ratio (D/T), the nozzle-to-shell diameter ratio (d/D), and the nozzle-to-shell nominal membrane stress ratio (s/S). Other parameters of interest are different lengths of internal protrusions attached to external nozzles and attachment angles of nozzles to spherical shells. With respect to nozzle clusters, additional parameters include the number, spacing, and geometric pattern of the attached nozzles.

The program is divided into an AEC (ORNL)-sponsored portion and a joint AEC and industry portion sponsored by the Pressure Vessel Research Committee (PVRC) of the Welding Research Council. The AEC (ORNL) program is directed primarily toward

multiple-nozzle connections; whereas the AEC-PVRC effort is concerned primarily with single-nozzle connections. Much of the research is being done under subcontract to either ORNL or to the PVRC, depending on the source of funds. Table 10 lists the projects currently active under both programs.

The joint AEC-PVRC program is coordinated by ORNL through the PVRC Subcommittee on Reinforced Openings and External Loadings. Included in the management functions being performed by ORNL are (1) review and evaluation of the research program, (2) direction of AEC-sponsored work and participation therein, (3) generation of analytical parameter studies in support of correlation and evaluation studies being made at Battelle Memorial Institute, (4) reporting and making recommendations to the Subcommittee on non-AEC sponsored projects, and (5) soliciting recommendations from relevant PVRC subcommittees on AEC-sponsored projects.

ORNL staff members attend the PVRC Design Division meetings and meetings of the subcommittees as part of the coordination task. Periodically, visits are made to the facilities of each program subcontractor to maintain up-to-date knowledge of the status of the work. This information is used in making evaluations and reports to the Subcommittee. A complete description of the program and status reports on the work being done by the participating organizations are

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Table 10. Current Experimental and Analytical Investigations of Nozzle Attachments

Program	Subcontractor	Funding Agency	Program
Joint AEC (ORNL)- FVRC; single-nozzle studies	University of Tennessee (I)	AEC	Experimental stress analyses of single radial and nonradial nozzles attached to spherical shells; determination of the internal pressure required for plastic collapse of a thin cylindrical shell with a single radially attached nozzle
	Westinghouse Research Laboratories	AEC	Photoelastic studies of single radial and nonradial nozzles attached to cylindrical shells; photoelastic studies of bolts without nozzles; photoelastic studies of closely spaced reinforced nozzles attached in pairs to cylindrical shells
	Auburn University (II)	AEC	Theoretical analysis of stresses in a single nonradial nozzle attached to a spherical shell
	University of Waterloo (I)	FVRC	Analytical and experimental investigations in connection with plastic limit design for spherical shells with single, right radial nozzles in which the composite structures are subjected to combinations of internal pressure and to force and moment loadings on the nozzles
	University of Waterloo (II)	FVRC	Derivation of upper-bound-limit equations and associated computer programs for analysis of cylindrical shells with radially and nonradially attached nozzles where the com- posite structure is subjected to internal pressure, external loadings on the nozzle, or combinations of internal pressure and external loads; experimental verification of selected limit loads
	University of Shefford	FVRC	Theoretical and experimental investigations of flat plates with single skewed holes; derivation of lower-bound-limit equations for analysis of cylindrical shells with radially and nonradially attached nozzles where the composite structure is subjected to internal pressure or combinations of internal pressure and external force and moment loadings on the nozzle
	Oak Ridge National Laboratory	AEC	Equation derivations for analysis of radial nozzle-to- cylindrical shell attachments and computer programming of the equations; finite-element studies of reinforcement configurations for spherical shells; finite-element studies of shell attachment regions; comparisons between analytical and experimental results; miscellaneous testing; parametric studies for program guidance and code use; construction of test systems and experimental stress analyses of thin-walled steel cylindrical shells with single nozzle attachments
AEC (ORNL); multiple- nozzle studies	University of Tennessee (II)	AEC	Experimental stress analyses of flat plates with clusters of holes (applicable to small openings in large vessels)
	Battelle Memorial Institute (II)	AEC	Theoretical analysis of stresses in flat plates with clusters of holes and clusters of nozzles attached to flat plates
	Auburn University (I)	AEC	Experimental stress analysis of spherical shells with clusters of nozzles and holes, and flat plates with clusters of nozzles (applicable to small nozzles on large vessels)
Joint AEC (ORNL)- FVRC; general	Battelle Memorial Institute (I)	AEC	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

contained in a project report.³⁹ An addendum was published in November 1970 that provides an up-dated status report of the work.⁴⁰ Status reports on the various projects were also made at the Second Annual Information Meeting - Studies in Applied Solid Mechanics, sponsored by ORNL. The 1970 meeting was held April 2-3 in Oak Ridge.

The work currently under way at ORNL is described in the following section of this report. A summary is

39. W. L. Greenstreet and R. C. Gwaltney, *Experimental and Analytical Investigations of the Structural Behavior of Nozzle-to-Shell Attachments*, USAEC Report ORNL-TM-2526, Oak Ridge National Laboratory, April 1969.

40. W. L. Greenstreet and R. C. Gwaltney, *Experimental and Analytical Investigations of the Structural Behavior of Nozzle-to-Shell Attachments*, USAEC Report ORNL-TM-2526, Addendum, November 1970.

then given of the work being done by the other participating organizations. For more detailed discussions of the program the reader is referred to the project report³⁹ and the 1970 addendum.⁴⁰

Oak Ridge National Laboratory Projects

In addition to program management responsibilities, ORNL is actively participating in work on the structural behavior of shells and nozzle-to-shell attachments. A description of the specific studies and a report on the status of this work follows.

Development of Analytical Methods for Nozzle-to-Cylindrical Shell Attachments

Nozzle-to-cylindrical shell attachments are being analyzed by using a closed-form solution for the governing differential equations. The equations were programmed for solution on a computer, and it is presently possible to analyze stresses due to internal pressure and out-of-plane bending moments applied to the nozzle. The analytical methods are based on shallow-shell theory, which is limited in application to thin-walled shells with a nozzle-to-shell radius ratio (ρ_o/R) no greater than $1/3$. The parametric limitations of the methods are illustrated in Fig. 16. According to Novozhilov⁴¹ it is accepted practice to utilize shallow-shell theory for shells having an h/L ratio (see Fig. 16) no greater than $1/5$. Another suggested limit shown in the figure is $1/8$. Results for cases where the parameters lie in the region between these two ratios are subject to question.

The analytical results for an internal pressure loading have been compared with experimental data for photoelastic model WC-2AY,⁴² which was tested at Westinghouse Research Laboratories. The model parameters are $\rho_o/R = 0.117$, $\rho_o/t = 14.8$, and $R/T = 29.6$. Comparisons of analytical predictions and experimental results are presented in Figs. 17 through 20 for the longitudinal plane (0° plane) containing the center lines of both the nozzle and shell. In these figures the circumferential and longitudinal stresses (indicated as σ) were normalized by dividing them by S_{nom} , the nominal stress based on the mid-wall diameter of the

Fig. 16. Regions of Applicability of Shallow-Shell Theory.

cylindrical vessel. Although the experimental data do not fall exactly on the theoretical curves, good agreement is generally observed. A report describing this work is being prepared.

Finite-Element Studies of Reinforcing for Nozzle-to-Spherical Shell Configurations

One important area of design of reinforced openings in pressure vessels is the determination of acceptable compact reinforcement configurations for single radial nozzles attached to spherical shells loaded with internal pressure. A limited parameter study is being conducted by using the finite-element method of analysis to investigate the effectiveness of various amounts of internal and external compact reinforcement for the spherical-vessel radial-nozzle configuration. The primary purpose of this study is to evaluate the effectiveness of inside compact reinforcement. Outside reinforcing in the form of circular fillets at the nozzle-to-shell junction is also being examined for comparison.

41. V. V. Novozhilov, *Thin Shell Theory*, P. Noordhoff, Ltd., Groningen, The Netherlands, 1964.

42. M. M. Leven, *Photoelastic Determination of Stresses in a Reinforced Steam-Charging Connection with a Thermal Sleeve*, Report 69-9D7-PHOTO-R1, Westinghouse Research Laboratories, Jan. 14, 1969.

Fig. 17. Comparison of Analytical and Experimental Results for Internal Pressure Test of Photoelastic Model WC-2AY Nozzle Inner Surface, 0° Plane.

Fig. 19. Comparison of Analytical and Experimental Results for Internal Pressure Test of Photoelastic Model WC-2AY Shell Inner Surface, 0° Plane.

Fig. 18. Comparison of Analytical and Experimental Results for Internal Pressure Test of Photoelastic Model WC-2AY Nozzle Outer Surface, 0° Plane.

Fig. 20. Comparison of Analytical and Experimental Results for Internal Pressure Test of Photoelastic Model WC-2AY Shell Outer Surface, 0° Plane.

The first series of calculations was performed for the relatively thick-shell nozzle-to-shell structures shown in Fig. 21, which have the parameters $D_i/T = 24$, $d_i/D_i = 0.20$, and $d_i/t = 11.43$, where D_i and d_i are the inside diameters of the shell and nozzle, respectively, and T and t are the corresponding wall thicknesses. A second series of calculations was performed for the relatively thin-shelled structures shown in Fig. 22. The dimensional parameters in this case are $D_i/T = 72$, $d_i/D_i = 0.20$, and $d_i/t = 35.71$. Each series consisted of one unreinforced case, two configurations of outside reinforcement, two configurations of inside reinforcement, and one configuration of reinforcement balanced between the outside and inside.

The specific finite-element analysis used in the study was the SAFE-2D computer program,⁴³ which is based on a two-dimensional axisymmetric solid-body formulation. Triangular ring elements were used with linearly varying displacement fields assumed within each element. This type of analysis is advantageous for this application because it is based on a solid-body formulation and is not subject to the limitations of shell theory.

43. D. C. Cornell, *SAFE-2D, A Computer Program for the Stress Analysis of Plane and Axisymmetric Composite Structures, A User's Manual*, USAEC Report GA-9076, Gulf General Atomic, February 1969.

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Fig. 21. Details of the Five Configurations Examined in the Finite-Element Studies of Area Reinforcing ($D_i/T = 24$).

Fig. 22. Details of the Five Configurations Examined in the Finite-Element Studies of Area Reinforcing ($D/T = 72$).

The compact reinforcements of the nozzle-to-shell configurations chosen were designed according to the reinforcement rules given in Welding Research Council Bulletin 133,⁴⁴ as follows:

$$A_r = dT \cos \phi,$$

$$\phi = \sin^{-1} (d/D),$$

$$L_c = 1.5(T/D)^{2/3} D.$$

44. R. L. Cloud and E. C. Kotsabough, *Proposed Reinforcement Design Procedure for Radial Nozzles in Spherical Shells with Internal Pressure, Phase Report No. 1*, Welding Research Council Bulletin No. 133, September 1968.

In these equations, A_r is the cross-sectional area required for 100% reinforcement of all planes passing through the center of the opening and normal to the surface of the vessel. Only the reinforcement included within a circle defined by the radial distance L_c , measured from the center of the opening, is considered to be applicable.

All the relatively thick-shell models were analyzed by using 1196 nodes and 2048 elements in the overall finite-element layout shown in Fig. 23. The membrane stresses resulting from the applied internal pressure loading are given in Fig. 23. Each model in the thin-shell series was analyzed separately by using individual finite-element layouts. The membrane stresses throughout were the same for both series. In

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Fig. 23. Overall Finite-Element Layout for the Relatively Thick-Shell Models.

both studies the membrane stress in the vessel, $S = 10,000$ psi, was employed as the stress-normalizing factor.

Examples of the pressure-deformed configurations for 65% inside and 65% outside reinforcement are shown in Figs. 24 and 25, respectively, for the thin-shell model. It should be kept in mind that the deformed shapes shown in these figures have been exaggerated to show the mode of deformation clearly.

The maximum normalized stresses for each configuration analyzed are listed in Table II. For the

Fig. 24. Displaced Configuration for the 65% Inside Reinforcement of the Thin-Shell Model Subjected to Internal Pressure Loading.

thick-shell models, the two outside reinforcement configurations gave essentially the same stress distributions and maximum stresses. The maximum stress levels for these cases were about 28% less than for the unreinforced case. A slight increase in maximum stress was found for inside reinforcement when compared with the unreinforced model. On the other hand, the balanced reinforcement produced a 33% reduction in maximum stress.

In the study of the thin-shell model, the two outside reinforcement configurations gave essentially the same stress distributions, with the maximum stress being 65 to 75% less than for the unreinforced case. A slight increase in maximum stress was noted for the 65% inside reinforced case, while a slight decrease was noted for the 100% inside reinforced case. The balanced reinforced case of the thin-shell model gave a 75% reduction in maximum stress.

Table 11. Maximum Normalized Stresses for Reinforced Radial Nozzles Attached to Spherical Shells

Configuration	Maximum Normalized Circumferential Stress (σ_θ/S)	Maximum Normalized Meridional-Plane Stress (σ_ϕ/S)
Models with parameters of $D/T = 24$, $d/D_i = 0.20$, $d/r = 11.43$, and $s/S = 1.0$		
Unreinforced	2.5	2.1
30% outside fillet reinforced	1.8	1.3
44% outside fillet reinforced	1.7	1.3
80% inside reinforced	2.6	2.6
112% inside reinforced	2.4	2.5
Balanced reinforced case with 51% reinforcing inside and 30% outside		
Models with parameters of $D/T = 72$, $d/D_i = 0.20$, $d/r = 35.71$, and $s/S = 1.0$		
Unreinforced	4.6	4.2
65% outside fillet reinforced	1.7	1.6
100% outside fillet reinforced	1.4	1.4
65% inside reinforced	4.3	5.1
100% inside reinforced	4.1	4.7
Balanced reinforced case with 65% reinforcing outside and 35% inside	1.3	1.2

Based on the results of this study, it was concluded that inside reinforcement alone was not effective for either the relatively thick- or thin-shell structures. On the other hand, compact outside fillet reinforcement produced relatively large reductions in maximum stress values for both thin- and thick-shell models. A combination of reinforcement on both the inside and outside appears to be the most effective in these particular applications.

Finite-Element Studies of Cylindrical Shells with Step Changes in Wall Thickness

A limited finite-element parameter study was conducted to determine the stress concentrations in stepped cylindrical shells with various fillet radii. A typical stepped cylinder with an outside fillet is shown in Fig. 26. Twenty-seven cases were analyzed to establish bounds on the stress-concentration effects. The parameters of interest are

$T/t = 1.5, 4.0,$

Fig. 25. Displaced Configuration for the 65% Outside Fillet Reinforcement of the Thin-Shell Model Subjected to Internal Pressure Loading.

Fig. 26. Typical Stepped-Cylinder Configuration.

$$r/t = 0.05, 0.15, 0.225, 0.3, 0.45, 0.6,$$

$$a/t = 5.0, 10.0, 20.0,$$

where T and t are thicknesses of the thicker and thinner sections, respectively, r is the fillet radius, and a is the inner radius of the cylindrical shell. These parameters are further identified in Fig. 26. The stepped cylinders were loaded by internal pressure, and both ends were considered to be capped at sufficient distances from the region of interest to eliminate any influence of end effects. A version of Wilson's code,⁴⁵ which is based on a two-dimensional axisymmetric solid-body formulation, was the specific finite-element analysis employed.

In order to provide verification of the applicability of the specific computer program used, comparisons between finite-element predictions and experimental results were made for two stepped cylinder models. The first comparison was for a stepped cylinder model

tested at Southwest Research Institute.⁴⁶ The model had an inside radius, a , of 9 in., a wall thickness, T , of 0.375 in. in the thicker section, and a wall thickness, t , of 0.1875 in. in the thinner section. The fillet radius, r , at the transition region was 0.1875 in. The experimental results were obtained both with strain gages and with a birefringent photoelastic coating. Different Poisson's ratios for the model and coating, along with the added thickness of the coating, tend to cause the birefringence data to be less accurate than the strain-gage data.⁴⁷ The results, shown in Fig. 27, indicate good agreement between the finite-element results and the experimental strain-gage data. The magnitudes of the birefringence results were higher; however, the trends of the stress distributions were the same. The

45. E. L. Wilson, "Structural Analysis of Axisymmetric Solids, *AIAA Journal*, 3(12): 2269-2274 (December 1965).

46. D. Bynum, Jr., *Fillet and Groove Stress Concentrations for Uniaxially and Biaxially Applied Loads, Final Report, Report No. T-1677-3, Southwest Research Institute, Aug. 27, 1962.*

47. W. H. Tupper, Jr., and A. S. Kofayachi, *Manual on Experimental Stress Analysis*, Society for Experimental Stress Analysis, 2nd ed., 1965.

Fig. 27. Comparison of Calculated and Experimental Stresses on the Outer Surface of a Stepped Cylinder ($T/t = 2.0$, $a/t = 48.0$, $r/t = 1.0$).

wirefringe data are not considered to be accurate, as noted above.

The second comparison between finite-element results and experimental data was for a stepped cylinder model⁴⁸ tested photoelastically by Leven.⁴⁸ The model had an inside radius, a , of 4.352 in., a wall thickness, T , of 0.486 in. in the thicker section, and a wall thickness, t , of 0.404 in. in the thinner section. The fillet radius, r , in the transition region was 0.050 in. Comparisons of analytical and experimental results for the outer surface, in Fig. 28, show good agreement. The stress ratio shown in the figure is defined as the stress in the structure at any point divided by the hoop membrane stress in the thinner section, with the mean radius used to calculate the membrane stress.

As a further verification of the applicability of Wilson's code to this particular problem, duplicate analyses for two specific cases were performed with another two-dimensional finite-element code, the SAFE-2D program.⁴⁹ The results of the duplicate analyses were described in the previous report in this series.⁴⁹ Good agreement was found for stress contours calculated by the two independent analyses.

48. M. M. Leven, *Stress Distribution in a Cylinder with an External Circumferential Fillet Subjected to Internal Pressure*, Research Memorandum 65-9D7-520-M1, Westinghouse Research and Development Center, July 21, 1965.

Fig. 28. Comparison of Calculated and Experimental Stresses in a Stepped Cylinder ($T/t = 1.2$, $a/t = 10.8$, $r/t = 0.12$).

The parameters of the 27 cases analyzed are given in Table 12. For convenience in establishing the finite-element representation, the thickness, t , of the thinner section was the same for every case. The calculated Lamé stresses are given in Table 13 for reference, along with the nominal membrane hoop and axial stresses in the thinner section, which were calculated by using the inside radius, a , of the cylindrical shell. The finite-element layout used with Wilson's code for the analysis of the configuration 1 model, having a T/t ratio of 1.5, an a/t ratio of 5, and an r/t ratio of 0.3, is shown in Fig. 29. The transition region for the model with a fillet is shown enlarged on the left; for the model without a fillet, the mesh was modified as shown in the contour. A total of 723 nodes and 618 elements was used for the model with a fillet, and 681 nodes and 568

49. B. L. Greenberg et al., "Experimental and Analytical Investigations of Nozzles," pp. 156-166, *Nuclear Safety Program Ann. Prog. Rept. Dec. 31, 1969*, USAEC Report ORNL-4511, Oak Ridge National Laboratory.

Fig. 29. Finite-Element Layout for Configuration 1 Model 31 ($T/t = 1.5$, $a/t = 5$, $r/t = 0.3$).

Table 12. Model Parameters

Configuration	Model No.	Parameters ^a						
		t (in.)	T (in.)	T/t	r (in.)	r/t	d (in.)	d/t
I	11	0.5 ^b	0.75 ^b	1.5 ^b			2.5	5
	12				0.875	0.05	5.0	10
	13						10.0	20
	21						2.5	5
	22				0.875	0.15	5.0	10
	23						10.0	20
	31						2.5	5
	32				0.150	0.30	5.0	10
	33						10.0	20
	41						2.5	5
	42				0.225	0.45	5.0	10
	43						2.5	20
II	11	0.5 ^c	2.0 ^c	4.0 ^c			2.5	5
	12				0.875	0.15	5.0	10
	13						10.0	20
	21						2.5	5
	22				0.1125	2.725	5.0	10
	23						10.0	20
	31						2.5	5
	32				0.150	0.30	5.0	10
	33						10.0	20
	41						2.5	5
	42				0.225	0.45	5.0	10
	43						10.0	20
	51						2.5	5
	52				0.100	0.60	5.0	10
	53						10.0	20

^a t = thickness of the thinner section, in.; T = thickness of the thicker section, in.; r = fillet radius, in.; d = inside radius of the cylindrical shell, in.; configuration = a set of models with the same T/t ratio; model No. = a single model within a configuration, with different r/t and d/t ratios.

^b For all models of configuration I.

^c For all models of configuration II.

Table 13. Calculated Load and Nominal Circumferential and Axial Stresses^a

Section	Configuration	Model No. ^b	Circumferential Stress (psi)			Axial Stress (psi)	
			Load Stress		Avg. Nominal Stress ^c	Load, Either Surface	Nominal ^c
			Inner Surface	Outer Surface			
Thin	I, II	11, 21, 31, 41, 51	1109	909	1009	455	909
		12, 22, 32, 42, 52	2105	1905	2005	952	1909
		13, 23, 33, 43, 53	4102	1102	4000	1951	2000
Thick	I	11, 21, 31, 41	700	500		300	
		12, 22, 32, 42	1400	1200		600	
		13, 23, 33, 43	2770	2570		1207	
	II	11, 21, 31, 41, 51	379	179		89	
		12, 22, 32, 42, 52	617	417		300	
		13, 23, 33, 43, 53	1109	909		455	
		11, 21, 31, 41, 51	1109	909		455	
		12, 22, 32, 42, 52	2105	1905		952	
		13, 23, 33, 43, 53	4102	1102		1951	

^a Load radial stresses are -200 psi at the inner surface and 0 psi at the outer surface.

^b Models 51, 52, and 53 listed for the thin section apply to configuration II only.

^c P , internal pressure, psi; a , inside radius of cylindrical shell, in.; t , thickness of thin section, in.

elements were used for the model without a fillet. Most of the elements were quadrilaterals, which were internally divided by the program into four triangular elements with linearly varying displacements.

An example of the type of finite-element idealization used with Wilson's code for the analyses of the configuration II models is shown in Fig. 30. These models had T/t ratios of 4.0, and this particular model had an a/t ratio of 5 and an r/t ratio of 0.6. Figure 31 shows the circumferential stress contours and the

maximum principal stress contours in the plane of the cross section for configuration II model 31 ($T/t = 4.0$, $a/t = 5$, and $r/t = 0.3$). Examination of this figure shows the very localized discontinuity effects in configuration II models; therefore, a quick transition to large elements in the surrounding region was made for this configuration. Examination of Figs. 29 and 30 shows that more elements could be placed in the fillet radius region of configuration II models than in configuration I models because of the limitation of the

Fig. 30. Finite-Element Layout for Configuration II Model 31 ($T/t = 4.0$, $a/t = 5$, $r/t = 0.6$)

Fig. 31. Circumferential Stress Contours and Maximum Stress Contours in the Plane of the Cross Section for Configuration II Model 31 ($T/t = 4.8$, $a/t = 1$, $r/t = 0.3$).

number of elements that can be used in the computer program.

The stress concentration factor (SCF) for each model was calculated by dividing the maximum stresses on the outer surface of the stepped cylinder within the transition region by the nominal shell membrane stresses given in Table 13. These SCF's were plotted as functions of the r/t ratio for the different a/t and T/t ratios. Two examples of the plots are shown in Fig. 32 for $a/t = 10$ and $T/t = 1.5$ and in Fig. 33 for $a/t = 10$ and $T/t = 4$. In these figures, K_z denotes the axial stress concentration factor, which was obtained by dividing the maximum calculated axial stress by the nominal membrane axial stress in the thin section. The

maximum principal stress concentration factor in the plane of the cross section, K_{max} , was calculated by dividing the highest maximum principal stress by the nominal membrane axial stress in the thin section, and by definition $K_{max} \geq K_z$. The circumferential stress concentration factor, K_θ , was obtained by dividing the maximum calculated circumferential stress by the nominal membrane circumferential stress in the thin section. Figure 32 represents four configuration I cases; Fig. 33 represents five configuration II cases.

No results for the cases without fillets ($r = 0$) were included, because the stresses in the vicinities of the corners theoretically approach infinity as the r/t ratios approach zero. Although finer and finer meshes would predict higher and higher stresses, the predicted values

Fig. 32. Stress Concentration Factors for Stopped. Voids of Configuration I with Ratios $a/t = 10$ and $T/t = 1.5$.

Fig. 33. Stress Concentration Factors for Stopped Cylinders of Configuration II with Ratios $a/t = 10$ and $T/t = 4$.

for zero fillet radii would not reach the theoretical maximum. However, in actual application the limiting case of zero radius would not be employed.

The study showed that variations in fillet radius have very little effect on circumferential stress concentration factors in that they remained approximately 1.0 for all r/t ratios studied. The axial stresses or maximum principal stresses in the plane of the cross section are greatly affected by the r/t ratios, particularly in the cases where the T/t ratio is 4.0. At r/t ratios above 0.6, the values of K_z and K_{max} approach 0.91 and 0.975, respectively, for a/t ratios of 5 to 20. Since only the values of K_z and K_{max} can be affected by increasing the r/t ratios, these r/t ratios in a stepped cylinder should be sufficiently large to give values of K_z and K_{max} less than 2.0.

Parameter Studies for Radial Nozzles Attached to Spherical Shells

Parameter studies currently being conducted at ORNL in support of code-rule development work at Battelle Memorial Institute include both triangular-ped and fillet reinforcement for radial nozzles attached to

spherical shells. Triangular-ped-reinforced nozzles thus far analyzed for an internal pressure loading by using the axisymmetric shell analysis program STATIC are listed in Table 14. This parameter study also included models with shell diameter-to-thickness ratios of 80, 160, and 250, which will give a total of 66 models. After completing the analysis for pressure loading the same models will be analyzed for moment loading and for a thermal gradient on the nozzle.

A second parameter study of 42 nozzles with fillet reinforcement is also being conducted with the STATIC program. These models were analyzed previously for internal pressure loading, and those shown in Tables 15 and 16 are being analyzed for moment loading. In addition, calculations were made for several models

Table 14. Models Thus Far Analyzed in Study of Nozzles in Spheres with Triangular-Ped Reinforcing Subjected to Internal Pressure Loading

Model No.	Parameters ^a		
	D/T	d/D	α (deg)
30-1	10	0.08	30
60-1	10	0.08	60
30-2	10	0.16	30
30-3	10	0.32	30
30-5	20	0.04	30
60-5	20	0.04	60
30-6	20	0.08	30
60-6	20	0.08	60
30-7	20	0.16	30
30-8	20	0.32	30
30-10	40	0.04	30
60-10	40	0.04	60
30-11	40	0.08	30
60-11	40	0.08	60
30-12	40	0.16	30
60-12	40	0.16	60
30-13	40	0.32	30
30-14	40	0.50	30

^a D = inside diameter of spherical vessel; d = inside diameter of nozzle; T = wall thickness of vessel; t = wall thickness of nozzle; α = angle of inclination of nozzle.

Table 15. Models Analyzed in Study of Moment Loading of Nozzles in Spherical Vessels with Fillet-Radius Reinforcing, $d/t = D/2T$

Model No.	Parameters ^a				
	D/T	d/D	r_o/t	$r_o/\sqrt{dt}$	$2A_f/tT$
20	10	0.01	0.25	5.67	0.246
21	10	0.02	0.25	2.81	0.119
22	10	0.04	0.25	1.40	0.565
8	10	0.08	1.72	4.82	0.888
23	10	0.16	3.22	4.50	0.987
24	10	0.32	6.82	4.78	0.947
25	10	0.50	15.9	7.13	0.866
1	20	0.01	0.25	3.96	0.126
2	20	0.02	0.25	1.98	0.0616
3	20	0.04	1.11	4.40	0.515
4	20	0.08	2.47	4.90	0.997
5	20	0.16	4.16	4.12	0.987
6	20	0.32	8.35	4.14	0.947
7	20	0.50	16.9	5.35	0.866
14	640	0.01	3.72	10.4	1.00
16	640	0.02	5.67	7.94	1.00
17	640	0.04	8.30	5.82	0.999
13	640	0.08	12.35	4.32	0.997
18	640	0.16	19.4	3.39	0.987
19	640	0.32	34.2	2.99	0.947
15	640	0.50	56.4	3.15	0.866
8	10	0.08	1.72	4.82	0.888
4	20	0.08	2.47	4.90	0.997
9	40	0.08	3.36	4.70	0.997
10	80	0.08	4.60	4.55	0.997
11	160	0.08	6.36	4.45	0.997
12	320	0.08	8.83	4.37	0.997
13	640	0.08	12.35	4.31	0.997

^a A_f = cross-sectional area of reinforcing provided by the fillet; r_o = outer radius of the nozzle attachment fillet.

Table 16. Models Analyzed in Study of Moment Loading of Nozzles in Spherical Vessels with Uniform-Wall Reinforcing on Vessel ($h > 1.0$) or Uniform-Wall Reinforcing on Nozzle ($g > 1.0$), $r_o/T = 0.25$

Model No.	T/T (h)	D/T	D/T	d/D	$r_o/\sqrt{dt}$	$2A_f/dT$ (e)
26	1.0	200	200	0.01	1.25	0.013
27	1.0	100	100	0.0141	1.25	0.018
28	1.0	40	40	0.0224	1.25	0.028
29	1.0	20	20	0.0316	1.25	0.038
30	1.0	10	10	0.0447	1.25	0.050
31	1.5	1000	667	0.016	0.52	0.004
32	1.5	250	167	0.034	0.49	0.007
33	1.5	50	33.3	0.097	0.39	0.010
34	2.0	500	250	0.031	0.51	0.003
35	2.0	250	125	0.069	0.32	0.005
36	2.0	50	25	0.22	0.23	0.006
r/h						
(g)						
37	3.0	200	200	0.01	1.25	0.013
38		50	50	0.022	1.14	0.023
39	6.0	200	200	0.011	1.13	0.012
40		50	50	0.025	1.00	0.019
41	12.0	250	250	0.01	1.12	0.010
42		100	100	0.024	0.74	0.010

^a A_f = cross-sectional area of reinforcing provided by the fillet. Models 26-30 do not require reinforcing. Reinforcing is provided by increase in vessel wall (models 31-36) or increase in nozzle wall (models 37-42); r = wall thickness of nozzle with uniform-wall reinforcing on nozzle; T = wall thickness of spherical vessel with uniform-wall reinforcing on vessel.

with thermal gradient loadings consisting of step temperature changes between the nozzle and the shell. Additional analyses will be done for thermal loadings.

Joint AEC (ORNL)-PVRC Cooperative Program - Single-Neck Studies

University of Tennessee (II) - Experimental Stress Analysis of Single Nozzles in Spherical and Cylindrical Shells

A continuing experimental study is being conducted on spherical shells with single radially and nonradially attached nozzles. The data obtained from this program provide a means for evaluating the predicted results of various methods of theoretical analysis.⁵⁰

The test models consist of nozzles having outer diameters of 2.625 and 7.875 in. that are welded to

steel hemispheres with a middle surface radius of 15.250 in. The models are instrumented with resistance-type strain gages and tested under internal pressure and external forces and moments applied to the nozzles. The nozzle loads include axial loads, pure bending moments, pure torsional moments, and shear loads. A summary of the nozzle configurations being studied is presented in Table 17. Tests of the configurations underlined in the table have not yet been started; footnote *b* indicates those configurations currently being tested; and the remainder have been tested.

The strain-gage data obtained from the tests were processed by means of a computer program⁵¹ to give normalized stresses in tabulated form at each rosette position and plots of the stresses versus model profile. The normalized stresses were obtained by dividing the principal stresses by normalizing factors that corresponded to the loadings employed.

A report summarizing the first program and results for the 7.875-in.-OD 0.187-in.-thick radial nozzle having a 10.0-in. inner protrusion was published.⁵² The dimensional details for the hemisphere and nozzle are given in Fig. 34. Typical results are presented as plots of normalized principal stresses in Figs. 35 through 38 for the following loading cases: internal pressure, an axial force on the nozzle, a couple on the nozzle in the axial plane, and a couple on the nozzle in the radial plane. The normalizing factor used in each case is indicated on the figure.

Westinghouse Research Laboratories - Photoelastic Studies

Internal pressure tests are being conducted on two cylindrical photoelastic models to provide information concerning the strain distributions for two closely spaced reinforced radial nozzles for comparison with

50. F. J. Witt, R. C. Gwaltney, R. L. Maxwell, and R. W. Holland, "A Comparison of Theoretical and Experimental Results from Spherical Shells with a Single Radially Attached Nozzle," *Trans. ASME, J. Eng. Power*, 89, Ser. A3, pp. 323-340, (1967).

51. G. R. Stong, *A Computer Program for the Generation of Stress Versus Profile Drawings*, Report ME-7-70-3, The University of Tennessee, Department of Mechanical and Aerospace Engineering, June 1970.

52. R. L. Maxwell, R. W. Holland, and G. R. Stong, *Experimental Stress Analysis of the Attachment Region of Hemispherical Shells with Attached Nozzles, Part 2b. Radial Nozzle, 7.875-in. OD-7.500 in. ID, 14.00 in. Penetration*, Report ME-7-70-1, The University of Tennessee, Department of Mechanical and Aerospace Engineering, June 1970.

Table 17. Nezzle Configurations

Model Designation	Dimensions ^a (in.)				
	L_o	L_i	d_o	d_i	t
7.875-in.-OD Radial Nozzle					
TS 887-881	14.0	10.0	7.875	7.125	0.375
TS 887-882	14.0	10.0	7.875	7.500	0.187
TS 887-883 ^b	14.0	0.0	7.875	7.500	0.187
7.875-in.-OD 45° Nozzle					
TS 887-451 ^c	14.0	0	7.875	7.125	0.375
2.625-in.-OD Radial Nozzle					
TS 882-881	8.5	4.0	2.625	2.125	0.25
TS 882-882	8.5	4.0	2.625	2.375	0.125
TS 882-883	8.5	0.75	2.625	2.375	0.125
TS 882-884	8.5	0.375	2.625	2.375	0.125
TS 882-885	8.5	0.0	2.625	2.5	0.062
2.625-in.-OD 22½° Nozzle					
TS 882-221	8.5	4.0	2.625	2.125	0.25
TS 882-222 ^d	8.5	4.0	2.625	2.500	0.0625
TS 882-223 ^d	8.5	0.0	2.625	2.500	0.0625
2.625-in.-OD 45° Nozzle					
TS 882-451 ^c	8.5	4.0	2.625	2.125	0.25
TS 882-452 ^c	8.5	4.0	2.625	2.500	0.0625
TS 882-453 ^c	8.5	0.0	2.625	2.500	0.0625

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

Configuration being prepared for testing.

Configurations to be tested.

^dConfigurations being tested.

Fig. 34. Cross-Sectional View of Hemisphere and Nozzle Assembly.

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Fig. 15. Normalized Principal Stress Versus Profile for 0° Plane and Internal Pressure Load. Normalized stress = stress/ N , where $N = pD/4T$, p = pressure, D = dome diameter, and T = dome wall thickness.

ORNL-DWG 70-14334

Fig. 16. Normalized Principal Stress Versus Profile for 0° Plane and Axial Force on Nozzle. Normalized stress = stress/ N , where $N = F/A$, F = axial force, and A = nozzle cross-sectional area.

ORNL-DWG 70-11335

Fig. 37. Normalized Principal Stress Versus Profile for 0° Plane and Couple on Nozzle in Axial Planes. Normalized stress = stress/ N , where $N = MC/I$, M = moment, C = nozzle radius, and I = nozzle moment of inertia.

ORNL-DWG 70-11336

Fig. 38. Normalized Principal Stress Versus Profile for 0° Plane and Couple on Nozzle in Radial Planes. Normalized stress = stress/ N , where $N = TC/J$, T = torque, C = nozzle radius, and J = nozzle polar moment of inertia.

similar information on single reinforced and unreinforced nozzles. One model represents a relatively thick-walled vessel, while the other model has relatively thin walls. In addition, hillside nozzles and openings were included to obtain information needed in support of earlier studies.

Photoelastic model 3, which contained the four nozzles and two openings listed in Table 18, was tested. The model was relatively thick walled and had an inside diameter-to-thickness ratio (D_i/T) of 12. The two closely spaced reinforced nozzles (WC-12DD) were mounted along the longitudinal axis. A single reinforced nozzle (WC-12D) identical to each of the closely spaced nozzles was included to provide a means for comparing the behavior of single and closely spaced nozzles. Nozzle WC-12E had the same basic dimensions as the above three, but the configuration had an internal protrusion, and the reinforcement, except for an outside fillet, was internal to the vessel. Nozzle WC-12C3 was an unreinforced hillside nozzle. In the case of the two openings, WC-12B was a lateral opening and WC-12C2 was a hillside opening. The bore of the

latter was made tangential to the inner surface of the shell. Details of the model are shown in Fig. 39. The results from tests of WC-12C3, WC-12B, and WC-12C2 were published.⁵³

A second photoelastic cylindrical model with a diameter-to-thickness ratio (D_i/T) of 100 was fabricated that contains the five nozzles listed in Table 19 and shown in Fig. 40. The nozzle types included in this model correspond directly to those included in model 3. Two closely spaced reinforced nozzles (WC-100DD) are mounted along the longitudinal axis of the model, and a single reinforced nozzle (WC-100D) with dimensions identical to those of the closely spaced nozzles is included to be used for comparing the effect of spacing nozzles close together. Nozzle WC-100E has the same basic dimensions as the above three, except it has an

53. W. L. Greenstreet, S. E. Moore, and R. C. Gwaltney, *Progress Report on Studies in Applied Solid Mechanics*, USAEC Report ORNL-4576, Oak Ridge National Laboratory, August 1970.

Table 18. Nozzles and Openings Included in Cylindrical Photoelastic Model 3 ($D_i/T = 12$)

Nozzle or Opening Designation	Parameters ^a					Description
	d_i/D_i	s/S	α , Hillside (deg)	γ , Lateral (deg)	γ_o/T	
WC-12DD	0.129	1.0	0	0	0.50	Two closely spaced outside reinforced nozzles
WC-12D	0.129	1.0	0	0	0.50	Single outside reinforced nozzle
WC-12E	0.129	1.0	0	0	0.50	Single inside reinforced nozzle
WC-12C3	0.0667	1.0	60.0	0		Hillside nozzle
WC-12B	0.0207	0	0	45		Lateral opening
WC-12C2	0.0257	0	77.3			Hillside opening

^a d_i = inside diameter of nozzle; D_i = inside diameter of model; σ = membrane stress in model; S = membrane stress in shell; α = angle of inclination of hillside from normal; γ = angle of inclination of lateral from normal; γ_o = outside radius of nozzle fillet; T = thickness of model.

Table 19. Nozzles Included in Cylindrical Photoelastic Model 4 ($D_i/T = 100$)

Designation	Parameters					Description
	d_i/D_i	s/S	α , Hillside (deg)	γ , Lateral (deg)	γ_o/T	
WC-100DD	0.110	1.0	0	0	1.1	Two closely spaced outside reinforced nozzles
WC-100D	0.110	1.0	0	0	1.1	Single outside reinforced nozzle
WC-100E	0.110	1.0	0	0	1.1	Single inside reinforced nozzle
WC-100C1	0.110	1.0	60.0	0		Unreinforced hillside nozzle

DIMENSIONS IN INCHES

Fig. 40. Westinghouse Research Laboratory Thin-Walled Photoelastic Model 4 ($D_i/T = 100$).

internal protrusion and is reinforced internally. Nozzle WC-100C1 is an unreinforced hillside nozzle.

The thin-walled model, model 4, is now being tested under internal pressure loading. A report will be prepared discussing the results of this test and the results for the nozzles of model 3 that have not yet been reported. This report will include an evaluation of the interaction of closely spaced nozzles on both relatively thick- and thin-walled cylindrical shells.

University of Waterloo (I) – Elastic and Plastic Analyses of Nozzle-to-Shell Attachments
(Sponsored by PVRC)

Methods of analysis and experimental data were developed to determine the elastic stresses at nozzle-to-vessel intersections, and methods were developed for plastic limit analyses and design of nozzle-to-vessel attachment configurations. Also experimental data were obtained for verification of the accuracy of these methods.

The remainder of the work in this investigation was completed this year with the publication of a report⁵⁴ describing upper- and lower-bound-limit analysis methods for predicting plastic limit loads for spherical shells with single radial nozzle attachments and the associated experimental investigation. A paper⁵⁵ describing the theory and analytical methods was prepared for journal publication. Also a paper⁵⁶ was prepared that describes six steel model tests conducted in support of the theoretical developments.

University of Waterloo (II) – Plastic Limit Analysis Studies for Tees and Laterals
(Sponsored by PVRC)

Methods are being developed for accurately determining upper-bound limits for pressure and for external force and moment loadings on the branch pipes of tees and lateral connections. An initial trial solution for lateral connections based on an approximately rigid,

perfectly plastic shell theory was developed first.⁵⁷ The analysis was subsequently refined by eliminating conservative approximations; and load minimization was done by using the nonlinear programming technique, CLIMES, employed at the Computer Center of the University of Waterloo.

The major difficulty in the analysis of lateral connections lies in the complexity of the geometric descriptions of plastic regions. As a result of this complexity and the presence of high strain concentrations at the lower crotch of laterals with branch-to-pipe diameter ratios (d/D) close to one, the analysis was divided into two parts. Part I is applicable to branch-to-pipe diameter ratios (d/D) varying from 0 to 0.7 and for a range of branch-pipe angles (α) between 0 and 60°, while part II is for d/D ratios varying from 0.7 to 1 and for branch-pipe angles ranging from 30 to 90° ($\alpha = 0^\circ$ for a tee). Both parts are limited to diameter-to-thickness ratios of both branch (d/t) and pipe (D/T) greater than 20 and for elastic branch-to-pipe hoop-stress ratios (s/S) from 0.25 to 3.0.

Fourteen models being tested to provide data for comparison with theoretical predictions are described in Table 20. All the models have been acquired; and, as indicated in the status column, two in this series remain to be tested. The type of loading to be employed in each case is also indicated in the table. Laterals Y_5 , Y_6 , Y_7 , and Y_8 were made by welding the nozzle to the shell, while the remainder, as well as all tees (T_1 , T_2 , etc.), were machined from one piece of material.

The results of the internal pressure test of lateral Y_1 are shown in Fig. 41. This lateral, which was machined from one forging, did not fracture, even at very large deformations. The value designated in the figure as P^* is the experimental limit pressure. The pressure test of welded lateral Y_2 resulted in fractures at the crotch and one side at pressures approximately twice the limit pressure of the connection and considerably greater than the limit pressure of straight pipe. Pressure versus displacement curves for lateral Y_5 are shown in Fig. 42.

In order to investigate the magnitude of strain concentrations at the crotch, machined lateral Y_2 , which was strain gaged and pressure tested under elastic conditions at ORNL,⁴⁹ was subjected to internal pressure loading. The strain versus pressure curves of six

54. S. Palusamy and N. C. Lind, *Limit Analysis of Rigid Radial Nozzles in Spherical Pressure Vessels Under Combined Loading*, report submitted Subcommittee on Reinforced Openings and External Loadings of the PVRC Design Division, University of Waterloo, December 1969.

55. S. Palusamy and N. C. Lind, *Limit Analysis of Non-symmetrically Loaded Spherical Shells*, Report No. 51, Solid Mechanics Division, University of Waterloo, July 1970.

56. N. C. Lind and S. Palusamy, *Experiments on Plastic Limit Behavior of Shell-Nozzle Juncures Subjected to Non-symmetrical Loading*, Report No. 47, Solid Mechanics Division, University of Waterloo, June 1970.

57. J. Schroeder and P. Rangarajan, "Upper Bounds to Limit Pressures of Branch-Pipe Tee Connections," *Proceedings First International Conference on Pressure Vessel Technology, Part I, Design and Analysis*, Delft, The Netherlands, September 29–October 2, 1967, pp. 277–291.

Fig. 41. Results from Limit-Pressure Test of 45° Machined Lateral Y_1 .

Fig. 42. Results from Limit-Pressure Test of 45° Welded Lateral Y_5 .

Table 20. Models for Limit Load Tests

Model	Parameters				Status	Type	Conditions
	d_m/D_m	D_m/T	s/S	α , Nozzle Angle (deg)			
Y ₁	1.0	34.5	1.0	45	Tested	Machined from one piece	Internal pressure
T ₁	1.0	34.5	1.0	0	Tested	Machined from one piece	Internal pressure
Y ₂	1.0	26.0	1.0	45	Tested	Machined from one piece	Internal pressure with special strain gages on crotch; to be retested to fracture
T ₂	0.6	34.5	0.9	0	Tested	Machined from one piece	Internal pressure
T ₃	0.35	25.0	1.0	0	Tested	Machined from one piece	Internal pressure
T ₄	0.5	25.0	1.0	0	Acquired	Machined from one piece	External out-of-plane couple on branch
T ₅	0.75	25.0	1.0	0	Tested	Machined from one piece	External in-plane couple on branch
T ₆	1.0	25.0	1.0	0	Tested	Machined from one piece	External out-of-plane couple on branch
Y ₃	0.75	25.0	1.0	45	Acquired	Machined from one piece	External in-plane couple on branch
Y ₄	0.6	26.0	1.0	45	Tested	Machined from one piece	Internal pressure
Y ₅	1.0	16.0	1.0	45	Tested	Commercially welded lateral	Internal pressure to fracture
Y ₆	1.0	20.0	1.0	45	Tested	Commercially welded lateral	Internal pressure to fracture
Y ₇	0.75	17.6	1.0	45	Tested	Commercially welded lateral	Internal pressure to fracture
Y ₈	1.0	18.0	1.0	45	Tested	Commercially welded lateral	Internal pressure to fracture

gages attached to the acute inside corner of the crotch are shown in Fig. 43. The highest strains were in the circumferential direction. These strains were about 1% when the limit pressure of the lateral was reached. The changes in slope of the strain versus pressure curve of gages 16 and 19, Fig. 43, indicate the limit pressure condition that was also indicated by displacement versus pressure curves of the test (not shown).

The theoretical results for the internal pressure loading case are shown in Fig. 44. The curves are plotted as ratios of upper bounds of the true limit pressure (experimental value) to the theoretical limit pressure of a branch-pipe connection; that is, p^*/p_p versus d/D for $D/T = 30$, $s/S = 1$, and branch-pipe angles (α) of 0 and 45°. The top curve in the figure represents

results obtained by using the initial solution, whereas the middle and bottom curves are solutions obtained by using the part I and part II equations for fillet radii equal to T at the crotch. The experimental points shown in Fig. 44 indicate a strengthening of the tee when $\alpha = 0$ but a weakening of the 45° lateral when the diameter ratio increases from 0.5 to 1.0.

Theoretical work was started on upper bounds to out-of-plane limit couples applied to the branch of tees exhibiting diameter ratios smaller than 0.7. The test apparatus developed for the application of external couples to branches of tees and laterals was modified to subject branch-pipe connections to out-of-plane couples.

Fig. 43. Results from Limit-Pressure Test of 45° Machined Lateral with 42 Strain Gages Attached at ORNL.

Fig. 44. Upper Bounds to Limit Pressures of Laterals and Tees.

University of Sherbrooke – Elastic Analysis of Skewed Holes in Flat Plates and Plastic Collapse Loads of Single Nozzle-to-Cylindrical Shell Attachments (Sponsored by PVRC)

Two separate investigations have been conducted at the University of Sherbrooke, and information is being developed that is applicable to determining the plastic collapse loads of cylindrical shells with single cylindrical nozzles attached and to the elastic stresses near single skewed holes in flat plates.

Lower- and upper-bound limit pressures were obtained for cylindrical shells with single radial nozzles. However, the upper-bound limit pressure formulation was preliminary, and since it appeared that an exact formulation for the limit pressure could be obtained from the lower-bound analysis, no further work was done on the upper-bound equations.

The equations for determining lower bounds to limit pressures for cylindrical nozzles attached to cylindrical shells were formulated.⁵⁸ With the limit pressure as the objective function, a lower bound for the limit pressure is found by maximizing the objective function over all admissible stress fields. The stress fields, being functions of arbitrary parameters and special coordinates, are chosen to satisfy the equilibrium equations, the symmetry conditions, and the boundary conditions at the ends. The computerized equations are solved by using SUMT, a nonlinear programming computer code.⁵⁹

The results of the analyses are compared with available relevant experimental results in Table 21. Except for model 30, the maximum deviation is less than 20%. The results of the lower-bound analysis are compared in Fig. 45 with approximate upper-bound results given in the literature for a vessel diameter-to-thickness ratio (D/T) of 50, a nozzle-to-vessel hoop-stress ratio (s/S) of 1, and a nozzle-to-vessel material-yield-stress ratio (η) of 1. The theories all show the same trend with respect to variation of geometric parameters and minimum strength; however, the "upper bounds" predicted by the analyses in refs. 57 and 60 cannot be

Fig. 45. Comparison of Present Lower Bound and Upper Bound Results.

directly compared with the lower-bound results because the analyses employ different yield surfaces and they contain certain approximations. The solution given in ref. 61 does not blend with either the upper bounds or the present lower bound.

The studies of elastic stresses near skewed holes in flat plates are described in Welding Research Council Bulletin No. 153.⁶³ The report contains comparisons of experimental and two-dimensional elasticity results obtained earlier, as well as some of the current

58. F. Ellyin and N. Türkkan, *Lower Bound to Limit Pressure of Nozzle-to-Cylindrical Shell Attachment*, Rapport Technique No. FE11-70, Université de Sherbrooke, September 1970.

59. A. V. Fiocco and P. G. McCormick, *Nonlinear Programming, Sequential Unconstrained Minimization Technique*, Wiley, New York, 1968.

60. R. L. Cloud and E. C. Rodabough, "Approximate Analysis of the Plastic Limit Pressure of Nozzles in Cylindrical Shells," *Trans. ASME, J. Eng. Power*, Series A, Vol. 90, No. 2, pp. 171-176, April 1968.

61. K. Krishnamoorthy, *Upper Bound Analysis of Normally Intersecting Cylindrical Shells*, M.A. Sc. thesis, Ecole Polytechnique, Montreal, Quebec, Canada, 1970.

62. E. J. Winkler, A. L. L. Venberg, and A. G. Pickett, *Experimental Investigation of Plastic Collapse Pressure Vessel Models*, Southwest Research Institute, San Antonio, Texas, Sept. 15, 1965.

63. F. Ellyin, *Elastic Stresses Near a Skewed Hole in a Flat Plate and Applications to Oblique Nozzle Attachments in Shells*, Welding Research Council Bulletin No. 153, pp. 32-51, August 1970.

Table 21. Comparisons of Lower Bound and Experimental Limit Pressures

Source	Model	Parameters ^a								
		d/D	s/T	D/T	s/S	$\rho = \frac{d}{L} \left(\frac{D}{T} \right)^{1/2}$	$\eta = \frac{\sigma_{ov}}{\sigma_{on}}$	$p_e = \frac{P_E}{P_{yv}}$	p^-	$\frac{p_e - p^-}{p_e} \times 100$
Ref. 62	4	0.405	0.20	20.1	2.02	1.82	1.00	0.40	0.35	12
	11	0.096	0.102	89.0	0.95	0.91	0.62	0.52	0.48	8
	12	0.374	0.43	88.3	0.87	3.12	0.57	0.25	0.22	12
	17	0.099	0.192	83.8	0.51	0.91	0.74	0.52	0.46	11
	18	0.395	0.78	84.2	0.51	3.63	1.00	0.27	0.24	11
	20	0.394	1.28	88.6	0.31	3.70	1.00	0.41	0.35	15
	21	0.096	0.073	128.1	1.31	1.08	0.59	0.35	0.29	17
	22	0.413	0.41	178.4	1.01	5.51	0.71	0.21	0.17	19
	24	0.380	0.20	88.5	1.88	3.57	0.70	0.18	0.15	17
	26	0.379	0.131	57.4	2.89	2.88	1.00	0.20	0.16	20
	27	0.097	0.20	178.6	0.48	1.29	0.54	0.37	0.31	16
	28	0.376	0.88	177.2	0.43	5.00	0.54	0.20	0.17	15
	29	0.097	0.32	177.9	0.30	1.29	0.51	0.49	0.40	18
	30	0.378	1.27	177.2	0.30	5.03	0.73	0.29	0.19	34
Ref. 57	T ₁	0.63	0.70	34.5	0.90	3.70	1.00	0.57	0.49	14
	T ₂	1.00	1.00	34.5	1.00	5.87	1.00	0.71	0.57	20

^a σ_{ov} = yield stress of vessel material, σ_{on} = yield stress of nozzle material, p^- = nondimensional lower bound to the limit pressure, P_E = experimental limit pressure, and $P_{yv} = 2\sigma_{ov} T/D$ = nominal yield pressure of vessel.

three-dimensional theoretical work. An approximate theoretical approach is suggested for computing the elastic stress concentration factor of oblique openings in spherical and cylindrical shells.

Auburn University (II) – Theoretical Analysis of Stresses in a Single Nonradial Nozzle in a Spherical Shell

The investigations at Auburn University are to provide analytical methods for obtaining the stresses in the junction region of spherical shells with single nonradial nozzle attachments. The first analysis to be developed was based on an extension of an existing analytical method, which in turn was based on shallow-shell theory.⁶⁴ The existing method was modified by using an exact description for the nozzle-to-shell intersection instead of an approximate description, such as was used in ref. 64. On this basis, the modified analytical method can be used to analyze nozzles with larger angles of obliqueness.

Formulations were developed for stress analysis of both nonprotruding nozzles and inwardly protruding

nozzles, where the shell and nozzle are loaded by internal pressure.⁶⁵ Computer programs were written for analyzing both types of nozzles,^{66,67} and preliminary comparisons were made between the theoretical predictions and experimental results from a model tested at the University of Tennessee.⁶⁸ The model, shown in Fig. 46, which consists of a 22½° oblique nozzle attached to a hemispherical shell, was subjected to internal pressure loading. The experimental and analytical results are compared in Figs. 47 and 48 for 0 and 90° planes, respectively. Although the experimental data are for a nozzle having an internal

64. D. E. Johnson, *Stresses Due to Forces on a Nonradial Nozzle in a Spherical Shell*. Ph.D. Dissertation, Cornell University, June 1963.

65. J. C. Yu and C. H. Chen, *Stresses in the Vicinity of a Nozzle Nonradially Attached to a Spherical Pressure Vessel*, Report ME-UC2889-2, Auburn University, April 1969.

66. J. C. M. Yu, C. H. Chen, and W. A. Shaw, *Stresses in the Vicinity of a Nozzle Nonradially Attached to a Spherical Pressure Vessel*, Report ME-UC-2889-3, Auburn University, August 1970.

67. J. C. M. Yu, C. H. Chen, and W. A. Shaw, *Stresses in the Vicinity of a Nozzle Nonradially Attached to and Penetrated into a Spherical Pressure Vessel*, Report ME-UC-2889-4, Auburn University, October 1970.

68. R. L. Maxwell, R. W. Holland, and G. R. Stengl, *Experimental Stress Analysis of the Attachment Region of Hemispherical Shells with Attached Nozzles*, Part 3a, Report ME-7-69-1, University of Tennessee, 1969.

ORNL-DWG 71-1296A

Fig. 46. Cross-Sectional View of Hemisphere and $22\frac{1}{2}^\circ$ Oblique Nozzle Assembly.

ORNL-DWG 71-1297A

Fig. 47. Comparison of Theoretical Results with Experimental Data for the 0° Plane of a $22\frac{1}{2}^\circ$ Oblique Nozzle Under Internal Pressure Loading.

Fig. 48. Comparison of Theoretical Results with Experimental Data for the 90° Plane of a $22\frac{1}{2}^\circ$ Oblique nozzle Under Internal Pressure Loading.

protrusion, while the theoretical results are for a nonprotruding nozzle, the comparisons were generally good. The calculated meridional, or axial, stresses near the junction on the outer surface in the 0° plane were less than the experimental values, as shown in Fig. 47. The normalizing factors used in the plots were the mean radii of the sphere (R_s) and nozzle (R_c) and the calculated membrane stress for the sphere (S).

Comparisons were also made between analytical results for a 10° oblique nozzle with and without an internal protrusion. The parameters for the nozzle are a nozzle radius-to-thickness ratio (r/t) of 20, a sphere-to-nozzle thickness ratio (t/T) of 2, and a sphere-to-nozzle radius ratio (R/r) of 12. Examples of the resulting comparisons are shown in Figs. 49 and 50 for the outer and inner surfaces of the 0° plane, respectively. There is little difference between results of the two analyses, which indicates that the internal nozzle protrusion has little effect on the resulting stress distribution for this particular angle of attachment.

The analyses will eventually be extended to include the various types of external loadings and nozzles having larger oblique angles.

AEC (ORNL) Program - Multiple-Nozzle Studies

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

For relatively small nozzles attached to large-radius cylindrical or spherical vessels, vessel curvature has little influence on stresses resulting from local nozzle loadings, such as axial thrust, moment, torsion, and shear. Consequently, flat plates with nozzle attachments can be used to study structural behavior in these cases.

The current investigation of flat plates containing clusters of holes is the first step toward an extensive study of flat plates containing clusters of closely spaced nozzles. Four models containing the hole configurations shown in Fig. 51 are being tested. The $36 \times 36 \times 0.375$ in. plates shown in Fig. 51 are loaded through

NORMALIZED STRESS

	AXIAL (σ_a/s)	CIRCUMFERENTIAL (σ_b/s)	
NOZZLE	—	---	
PROTRUDING	○	△	
FLUSH			

WHERE s = CALCULATED MEMBRANE STRESS IN SPHERE

Fig. 49. Comparison of Theoretical Results for Outer Surface of a 10° Oblique Nozzle With and Without an Internal Protrusion.

NORMALIZED STRESS

	AXIAL (σ_a/s)	CIRCUMFERENTIAL (σ_b/s)	
NOZZLE	—	---	
PROTRUDING	○	△	
FLUSH			

WHERE s = CALCULATED MEMBRANE STRESS IN SPHERE

Fig. 50. Comparison of Theoretical Results for Inner Surface of a 10° Oblique Nozzle With and Without an Internal Protrusion.

Fig. 51. Hole-Cluster Pattern and Details for Flat-Plate Model.

32 hydraulic rams positioned eight to a side along the edges of the specimen. Initially, two unpierced plates were tested to insure that the desired state of true biaxial stress was developed within the central region of the plate. These tests along with the uniaxial and biaxial tests of a plate containing a single hole, were described in the previous report in this series.⁶⁹

A report was published that describes the results of uniaxial and biaxial tests on a plate containing two holes.⁷⁰ The plate was instrumented on one side with electrical-resistance strain gages, as shown in Fig. 52. The symbol R in the table in the lower part of the figure denotes the hole radius. The tests were conducted by using essentially the same procedure as that described previously for the one-hole plate.⁷¹ Each test was conducted with the gaged side of the plate up in the loading apparatus and then repeated with the gaged side down. The two sets of readings for each gage were

Fig. 52. Location of Strain Gages for Two-Hole Flat-Plate Model.

69. B. L. Greenstreet et al., "Experimental and Analytical Investigations of Nozzles," pp. 173-176, *Nuclear Safety Program Ann. Progr. Rept. Dec. 31, 1969*, USAEC Report ORNL-4511, Oak Ridge National Laboratory.

70. A. Mathews and R. B. Michaels, *Experimental Stress Analysis of Clusters of Holes of Equal Diameters in Flat Plates, Part II, Two Holes*, Report EM-70-2, University of Tennessee, Department of Engineering Mechanics, June 1970.

71. A. Mathews, *Experimental Stress Analysis of Clusters of Holes of Equal Diameters in Flat Plates - Part I, A Single Hole*, Report EM-69-2, University of Tennessee, Department of Engineering Mechanics, April 1969.

averaged to compensate for possible bending or eccentricities in the loading system. Materials properties were determined from test coupons taken from the edges of the plate.

Typical radial and tangential stress distributions are shown in Figs. 53, 54, and 55 for the 90° plane (plane 9 in Fig. 52). The full curves in these figures represent theoretical results obtained at Battelle Memorial Institute,⁷² while the points indicate experimental results. The directions of loading 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

72. L. E. Hulbert, A. T. Hopper, and E. E. Rybicki, *Second Annual Report on the Stress Analysis of Perforated Plates and Plates with Single and Clustered Nozzles*, Battelle Memorial Institute, Oct. 9, 1970.

Fig. 53. Radial and Tangential Stresses in Uniaxial Tension for 90° Plane.

Fig. 54. Radial and Tangential Stresses in Uniaxial Tension for 90° Plane.

undisturbed tensile stress of 1650 psi. The abscissa is the distance from the center of the hole divided by the hole radius, R . The agreement between calculated and experimental results is generally good, except for the biaxial loading.

Plots of the tangential stresses around the edge of one hole for uniaxial and biaxial loadings, respectively, are given in Figs. 56 and 57. The orientation angle, θ , is taken as the abscissa, and the planes in which data were taken are indicated in Fig. 52. In general, the agreement between analytical and experimental results was good. In Fig. 57, which is for biaxial loading, the experimental data lie slightly below the analytical curve. The additional dashed curve shows the predicted results based on superposition of the experimental data for the two directions of uniaxial loading. Close agreement was found between the experimental biaxial loading data and the predictions based on superposition.

Testing of the three- and five-hole plates was completed, and reports are now being prepared.

Fig. 55. Radial and Tangential Stresses in Biaxial Tension for 1:1 Ratio in 90° Plane.

Fig. 56. Tangential Stresses at Edge of Hole in Uniaxial Tension.

Fig. 57. Tangential Stresses at Edge of Hole in Biaxial Tension for 1:1 Ratio.

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

As noted in the preceding section, methods of stress analysis applicable to flat plates with clusters of holes have been developed and programmed for the computer at Battelle Memorial Institute,⁷² and methods of analysis for flat plates with clusters of nozzles are presently being developed. Results obtained through the use of these methods are compared with experimental results obtained for perforated plates at the University of Tennessee and for plates with nozzles at Auburn University. The loadings include in-plane forces applied to perforated plates and plates with nozzles (used to simulate internal pressure loading of cylindrical and spherical vessels) and nozzle loadings consisting of axial forces, shear forces, and moments. As the computer programs are developed and verified experimentally, they will be used at ORNL to conduct parameter studies of nozzle clusters. The results of these studies are applicable to clusters of small nozzles attached to vessels having large diameter-to-thickness ratios.

A computer program has been developed and used for the analysis of square flat plates containing one and two nozzles. Loadings considered on the nozzles were internal pressure, thrust, moment, and a shear force. In

addition, the structures were analyzed for cases in which the plate edges were loaded to produce uniaxial and biaxial membrane stress states in the plate. For the two-nozzle configuration, loadings that were symmetric or antisymmetric about the symmetry lines were treated.

Numerical results were obtained for 20 cases involving flat plates with one and two attached nozzles. The plate was 16 in. square by $\frac{3}{8}$ in. thick. The nozzles were 8.5 in. long and 0.25 in. thick with a mean diameter of 2.375 in. A sketch of the two-nozzle plate is shown in Fig. 58. For the single-nozzle model, the nozzle was located at the center of the plate.

Typical results are shown in Figs. 59, 60, and 61 for 1:1 biaxial loading, equal moments acting in the same direction on both nozzles (antisymmetric loading), and equal moments acting in opposite directions

on the nozzles (symmetric loading), respectively. In these figures, h is the thickness, σ_x and σ_y are the stresses in the x and y directions, as specified in Fig. 59(a), and Q_ϕ is the shear in the nozzle. The s refers to the longitudinal direction along the nozzle for $\theta = 0^\circ$.

The most notable characteristic of the results for the biaxial loading is the rapid decay of the stress resultants away from the interface of the plate and nozzle. Similarly the shear, Q_ϕ , in the nozzle decays quite rapidly away from the interface. This rapid decay behavior was typical of all examples studied.

A complete description of the results obtained thus far is presented in ref. 72. Future work will include analysis of the five-nozzle cluster and comparisons between analytical and experimental results.

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

Two extensively strain-gaged hemispherical shell models and a series of flat-plate models are being tested at Auburn University. One of the hemispherical shell models was used for a study of stress distributions around four separate clusters of holes under internal pressure. The vessel has an outside diameter of 40.901 in. and a wall thickness of 0.503 in. The model was divided into quadrants containing two-, three-, four-, and five-hole clusters arranged as shown in Fig. 62. All holes are radial and have diameters of 1.572 in. The hole clusters are centered at a vertical angle of 47° from the base of the model to avoid possible influences of the vessel flange on the strain field in the vicinity of the holes. The holes were sealed by pistons and O-rings to simulate a pressurized vessel.

Three-gage electrical-resistance strain-gage rosettes were attached to the vessel along radial lines (stringers) designated as x -axis lines. The test consisted of subjecting the model to a static pressure of 200 psi and reading the strain gages while the pressure was maintained constant by means of a dead-weight tester. Examples of the resulting plots of the ratios of measured stress to nominal stress are shown in Figs. 63 through 66 for the two-, three-, four-, and five-hole clusters, respectively.

The stress ratios were found to approach a value of three (which is the theoretical stress concentration for a single hole in a flat plate under uniaxial tension) in the case of two stringers — stringer 1 of the two-hole cluster, Fig. 63, and stringer 7 of the four-hole cluster, Fig. 65. Generally a maximum stress ratio of

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Fig. 58. Two-Nozzle Flat-Plate Analytical Model.

Fig. 59. Results for Square Plate with Two Nozzles Attached and Biaxial 1:1 Tension Loading on the Plate.

Fig. 60. Results for Square Plate with Two Nozzles Attached and Loaded with Moments Applied in the x - z Plane in the Same Direction.

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Fig. 61. Results for Square Plate with Two Nozzles Attached and Loaded with Moments Applied in the x - z Plane in Opposite Directions.

ORNL-DWG 70-14152

Fig. 62. Hole Arrangement in the Auburn Hemispherical Vessel Model.

ORNL-DWG 70-14154

LOAD PRESSURE = 200 psi

NOMINAL STRESS:

$$S = \frac{PD}{4t} = 4050 \text{ psi}$$

- INSIDE $\frac{\sigma_y}{3}$
- OUTSIDE $\frac{\sigma_y}{3}$
- INSIDE $\frac{\sigma_x}{3}$
- OUTSIDE $\frac{\sigma_x}{3}$

Fig. 63. Strain-Gage Results for Stringer 1 of the Two-Hole Quadrant.

ORNL-DWG 70-14155

LOAD PRESSURE = 200 psi

NOMINAL STRESS:

$$S = \frac{PD}{4t} = 4050 \text{ psi}$$

- INSIDE $\frac{\sigma_y}{3}$
- OUTSIDE $\frac{\sigma_y}{3}$
- INSIDE $\frac{\sigma_x}{3}$
- OUTSIDE $\frac{\sigma_x}{3}$

Fig. 64. Strain-Gage Results for Stringer 9 of the Three-Hole Quadrant.

Fig. 65. Strain-Gage Results for Stringer 7 of the Four-Hole Quadrant.

Fig. 66. Strain-Gage Results for Stringer 9 of the Five-Hole Quadrant.

two was recorded for most stringers; the stress ratios recorded at locations relatively remote from the holes were one in each case. The O-ring seals were found to have an insignificant effect on the stress distributions, as indicated by the low values of radial stress near the edges of the holes. A report is being prepared on the results of this experiment.

The results of the experimental stress analysis of hemispherical model 2, which contained two closely spaced 7.002-in.-OD radial nozzles, were published.⁷³

This investigation was described in the previous report in this series.⁷⁴ Model 2 is now being modified by the addition of two nozzles to form a four-nozzle cluster. Holes for the additional nozzles have been bored by

using the setup shown in Fig. 67, and the machined nozzles are being welded to the shell. The finished weld will be ground in each case to produce a zero-radius intersection between the nozzle and shell.

Testing of the first of a series of flat plates with attached nozzles was initiated at Auburn University. A total of three flat-plate models containing a single nozzle, two nozzles, and a five-nozzle cluster, as shown in Fig. 68, will be tested.

The loading frame for this study, shown in Fig. 69, was designed and fabricated at ORNL and was modified and assembled at Auburn University. In order to ascertain that the desired stress state will be developed in the central region of the plates, an unpierced flat plate, shown positioned in the loading frame in Fig. 69, was loaded in biaxial tension by the 32 rams positioned in sets of eight along each side of the plate.

The three models to be tested in this study were fabricated at ORNL; also the first (the single-nozzle model) was instrumented with strain gages at ORNL.

73. R. W. Aderholdt and W. F. Swinson, *Experimental Stress Analysis of Spherical Pressure Vessels with Closely Spaced, Large Radial Nozzles Attached*, Technical Report ME-UC-2670-1, Auburn University, Engineering Experiment Station, March 1970.

74. Op. cit. ref. 69, pp. 180-185.

Fig. 67. Apparatus Used in Boring Holes for Two Additional Nozzles in Model 2.

Fig. 68. Nozzle Design and Cluster-Pattern Details for Flat-Plate Models.

Testing of the single-nozzle model is now under way at Auburn University.

Joint AEC (ORNL)-PVRC Cooperative Program - General

Battelle Memorial Institute (I) - Evaluation and Correlation of Experimental and Analytical Results

At Battelle Memorial Institute, correlations and evaluations of experimental and analytical data obtained under this and other programs are being made in order to develop proposed alternate code rules for nozzle-to-shell attachments. The rules are prepared in forms suitable for inclusion in the various codes. Proposed rules for reinforcement of radial nozzles attached to spherical and cylindrical shells⁷⁵ are being considered for inclusion in Sections III and VIII Division 1 and Division 2 of the ASME Boiler and Pressure Vessel Code.

75. E. C. Rodabaugh, Phase Report 117-3 on Proposed Alternate Rules for Use in ASME Codes, Battelle Memorial Institute, August 1969.

A review was made of known service experience and test data on nozzles in pressure vessels with nonintegral reinforcing.⁷⁶ Nonintegral reinforcing is defined as reinforcing that is not fully continuous in the shell, such as saddle reinforcement. This report is intended to provide information to be used in improvement of code design rules for such reinforcing. The results of the survey indicated the importance of insuring that microscopic cracks are eliminated from the toes of the fillet welds. It was recommended that inspection for such cracks be made as late as possible and preferably after application of the hydrostatic test. This report was approved for publication as a Welding Research Council Bulletin.

Reports are being prepared that will contain graphs and proposed rules covering internal pressure loads,⁷⁷ moments acting on the nozzle,⁷⁸ and thermal loads resulting from a step temperature change between nozzle and shell. The report on moment loadings will include factors to be used by the designer in the case of combined internal pressure and moment loadings to determine the reserve strength over the pressure load that would be available to support the moment loading.

Conclusions

Considerable progress was made in the continuing program of development and verification of analytical methods capable of determining the structural adequacy of nozzle-to-shell attachment configurations of nuclear pressure vessels. Included were studies on single and closely spaced nozzles both radially or nonradially attached. In the area of elastic behavior, analysis methods were developed and are being used to investigate reinforcement configurations for single, radial, nozzle-to-spherical shell attachments. In addition, the development of a method for analysis of single, unreinforced, nonradial nozzles attached to spherical vessels was completed, and experimental work for this configuration is being continued. Developments related

76. E. C. Rodabaugh, Phase Report 117-5, Review of Service Experience and Test Data on Openings in Pressure Vessel with Nonintegral Reinforcing, Battelle Memorial Institute, March 1969.

77. Battelle Memorial Institute, Phase Report 117-2, Additional Data on Elastic Stresses in Nozzles in Pressure Vessels with Internal Pressure Loading (to be published).

78. Battelle Memorial Institute, Phase Report 117-6, Elastic Stresses in Nozzles in Pressure Vessels with Pressure, Moment and Thermal Gradient Loadings (to be published).

Fig. 69. Setup for Conducting Tests of Flat Plates with Clusters of Nozzles Showing Unpierced Plate in Biaxial Testing Position.

to the first phase, the consideration of ideal geometries, are in progress for cylindrical shells. Experimental investigations of the effects of closely spaced holes in flat plates were completed, as was the test of a spherical vessel with closely spaced holes. Experiments on closely spaced nozzles and theoretical studies of such nozzles are in progress, with completion of some portions of each investigation achieved.

Parametric studies were made with available computer codes for design code use. In addition, design rules are being prepared on the basis of studies to date.

Studies being carried out under sponsorship of the Pressure Vessel Research Committee for determining plastic collapse loads for cylindrical and spherical vessels with single nozzles are in progress. Completion of portions of these studies was achieved.