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COMPARISON OF THEORETICAL AND EXPERIMENTAL STRESSES ON A SPHERICAL SHELL WITH A SINGLE RADIALY ATTACHED NOZZLE

F. J. Witt R. C. Gwaltney
B. L. Greenstreet

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SPHERICAL SHELL WITH A SINGLE RADially ATTACHED NOZZLE

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COMPARISON OF THEORETICAL AND EXPERIMENTAL STRESSES ON A SPHERICAL SHELL WITH A SINGLE RADIALY ATTACHED NOZZLE

F. J. Witt R. C. Gwaltney
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Abstract

The stresses on a series of steel spherical-shell models with single nozzles radially attached were examined with strain gages. The parameters studied were nozzle dimensions and length of internal nozzle protrusions. The loads were internal pressure on the shell and nozzle and axial thrust and moment loadings on the nozzle. Both experimental and theoretical stresses are reported for six configurations for which the sphere dimensions were equal and the outside diameter of the attached nozzle was constant. In most instances the nozzle protruded through the vessel.

The work was performed primarily to obtain empirical design data and to assess the adequacy of theoretical methods. The most significant conclusion is that for each configuration and all loadings the maximum experimental stresses are well approximated by the theoretical analyses. The approximations improved for the thinner nozzles. The theoretical stress distributions also agreed quite well in general with those obtained experimentally.

Introduction

The development of analytical models that provide realistic values for stresses and displacements in actual or proposed configurations is fundamental to the design of pressure vessels and shell structures. With this concept in mind the limitations of and restrictions on the analytical methods take on considerable significance; thus, model testing should not only provide design data but also a basis for comparing experimental data with theoretical results.

An experimental testing program is being conducted at the University of Tennessee under subcontract with Oak Ridge National Laboratory. This work is, in turn, sponsored by the U.S. Atomic Energy Commission in close cooperation with the Subcommittee on Reinforced Openings and External

Loadings of the Design Division of the Pressure Vessel Research Committee. The program encompasses an extensive investigation of steel models consisting of a nozzle attached to a spherical shell and includes the investigation of nonradial, as well as radial, attachments. In these models, the fillet radius between shell and nozzle is essentially zero so that comparisons between theoretical and experimental results are not encumbered by stress distortions caused by fillets. The effects of the nozzle protruding through the vessel are also being investigated. The loadings of principal interest are internal pressure on the shell and nozzle, axial thrust on the nozzle, and a bending moment applied to the nozzle.

Two computer programs developed at Oak Ridge National Laboratory are being used to obtain the theoretical results. With these programs, CERL-II code¹ and the Unified Shell Analyzer (USA code),² nozzles radially attached to spherical shells may be examined.

The CERL-II code is a revised version of a computer code obtained by J. L. Mershon,* USAEC, from the Central Electricity Research Laboratory in London, England.[†] The code computes the stress distributions in a spherical vessel and the radially attached nozzle for internal pressure loadings and for axial thrust, bending moment, and pure shear loads applied to the nozzle. For the moment and shear analyses it is assumed that the stresses vary circumferentially about the junction as the cosine of the circumferential angle measured from the plane of loading. The nozzle may protrude through the vessel; however, its junction with the sphere is assumed to be remote from any additional edge effects. The details of the analysis are given in Ref. 3.

The USA code analyzes an axisymmetrically loaded pressure vessel composed of several different types of elemental shell structures. The edges of any elemental shell may be close or remote and each shell can be joined to any logical combination of shells around it. Boundary conditions may be applied to any or all edges. The recent version of the code accounts for

*Vice-Chairman of the Subcommittee on Reinforced Openings and External Loadings, Pressure Vessel Research Committee.

[†]Letter from B. R. Tozer, Central Electricity Generating Board, to J. L. Mershon, Feb. 4, 1963.

pressure, axial thrust on the nozzle, and thermal loadings. The theoretical solutions for spherical and cylindrical shells are given in detail in Ref. 4. Both the USA and CERL-II codes are for shells joined at their middle surface. The codes produce identical results for duplicate analyses.

The first phase of the testing program at the University of Tennessee was an investigation of a relatively small, radially attached nozzle with a fixed outside diameter.⁵ In the following, a comparison is made between the experimental data and results from the two computer programs. The variation of experimental stresses as a function of nozzle protrusion length is discussed, as well as the variation of stresses due to moment loading as a function of the angle measured from the plane of loading. One example of superposed loadings is presented. In the appendix a comparison is made between theoretical and experimental results for two models that fall well into the thin-shell range.

Comparison of Theoretical and Experimental Results

The basic model from which the experimental data were obtained is shown in Fig. 1. As previously noted, there is essentially no fillet at the junction. Additional test models were obtained by boring out the nozzle to a nominal thickness of 0.125 in., cutting off the internal penetration at two stub lengths, removing the stub altogether, and then again boring out the nozzle to a nominal thickness of 0.0625 in. The dimensional and loading data for the six models obtained by the machining processes described above are summarized in Table 1. Each model was loaded by an internal pressure, an axial compressive thrust, and a bending moment applied to the nozzle. One loading on Model 1 was a combined pressure and axial compressive thrust load.

Comparisons of theoretical and experimental stresses are shown in Figs. 2 through 19. The theoretical results were obtained with both the CERL-II and USA codes. Since in these codes the configurations are joined at the middle surfaces, the theoretical nozzle stresses are plotted from that origin. The experimental data for the nozzle are also plotted at distances measured from the middle surface of the sphere. Gages were

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Fig. 1. Cross-Sectional View of Hemisphere and Nozzle Assembly - Model 1.

Table 1. Dimensional and Loading Data for the Six Spherical Shell Models

Sphere radius: 15.25⁴ in.

Sphere thickness: 0.375 in.

Outside diameter of nozzle: 2.625 in.

Model No.	Internal Nozzle Length (in.)	Nozzle Radius ^a (in.)	Nozzle Thickness (in.)	Nozzle Radius-to-Thickness Ratio	Loading		
					Internal Pressure (psi)	Axial Compressive Thrust (lb)	Bending Moment (in.-lb)
1	4.125	1.188	0.25	4.75	400 ^b	6000 ^b	8000
2	4.125	1.25	0.125	10.0	300	3000	4500
3	0.75	1.25	0.125	10.0	300	3000	4500
4	0.375	1.25	0.125	10.0	300	3000	4500
5	0.0	1.25	0.125	10.0	300	3000	4500
6	0.0	1.281	0.0625	20.5	200	1500	2400

^aAt middle surface.

^bModel 1 was also subjected to a simultaneous loading of 400-psi internal pressure and a 6000-lb axial compressive thrust.

Fig. 2. Comparison of Theoretical and Experimental Stresses on Model 1 for an Internal Pressure of 400 psi.

Fig. 3. Comparison of Theoretical and Experimental Stresses on Model 1 for an Axial Compressive Thrust Load of 6000 lb on the Nozzle.

Fig. 4. Comparison of Theoretical and Experimental Stresses in Plane of Loading on Model 1 for a Pure Moment Loading of 8000 in.-lb on the Nozzle.

Fig. 5. Comparison of Theoretical and Experimental Stresses on Model 2 for an Internal Pressure of 300 pci.

Fig. 6. Comparison of Theoretical and Experimental Stresses on Model 2 for an Axial Compressive Thrust Load of 3000 lb on the Nozzle.

Fig. 7. Comparison of Theoretical and Experimental Stresses in Plane of Loading on Model 2 for a Pure Moment Loading of 4500 in.-lb on the Nozzle.

Fig. 8. Comparison of Theoretical and Experimental Stresses on Model 3 for an Internal Pressure of 300 psi.

Fig. 9. Comparison of Theoretical and Experimental Stresses on Model 3 for an Axial Compressive Thrust Load of 3000 lb on the Nozzle.

Fig. 10. Comparison of Theoretical and Experimental Stresses in Plane of Loading on Model 3 for a Pure Moment Loading of 4500 in.-lb on the Nozzle.

Fig. 11. Comparison of Theoretical and Experimental Stresses on Model 4 for an Internal Pressure of 300 psi.

Fig. 12. Comparison of Theoretical and Experimental Stresses on Model 4 for an Axial Compressive Thrust Load of 3000 lb on the Nozzle.

Fig. 13. Comparison of Theoretical and Experimental Stresses in Plane of Loading on Model 4 for a Pure Moment Loading of 4500 in.-lb on the Nozzle.

Fig. 14. Comparison of Theoretical and Experimental Stresses on Model 5 for an Internal Pressure of 300 psi.

Fig. 15. Comparison of Theoretical and Experimental Stresses on Model 5 for an Axial Compressive Thrust Load of 3000 lb on the Nozzle.

Fig. 16. Comparison of Theoretical and Experimental Stresses in Plane of Loading on Model 5 for a Pure Moment Loading of 4500 in.-lb on the Nozzle.

Fig. 17. Comparison of Theoretical and Experimental Stresses on Model 6 for an Internal Pressure of 200 psi.

Fig. 18. Comparison of Theoretical and Experimental Stresses on Model 6 for an Axial Compressive Thrust Load of 1500 lb on the Nozzle.

Fig. 19. Comparison of Theoretical and Experimental Stresses in Plane of Loading on Model 6 for a Pure Moment Loading of 2400 in.-lb on the Nozzle.

located on the inner surface of the nozzle at the middle surface of the sphere. These data were plotted at the origin of the nozzle and reflect the sharp gradients that exist inside a junction, as shown in photoelastic studies.

Comparisons of theoretical and experimental stresses for pressure, moment, and axial thrust loadings on Model 1 are given in Figs. 2 through 4. It should be noted that the ratio of the radius at middle surface to the thickness for the nozzle is 4.75, which is outside the range of thin-shell theory.⁶ For internal pressure results, as shown in Fig. 2, the discrepancies of theoretical and experimental results for the spherical shell are rather extreme. The meridional outer and circumferential inner theoretical stresses fail to have even the correct curvature. Of the data for the six models tested the results shown in Fig. 2 exhibit the greatest discrepancy. It is significant that theoretical and experimental data agree quite well for both the outer and inner nozzles. In Figs. 3 and 4, excellent agreement is shown between theoretical and experimental data; that is, no significant discrepancies exist for either moment or axial thrust loading.

For configuration 2 the nozzle was bored out to a thickness of 0.125 in. to give a radius-to-thickness ratio of 10, and the tests performed on Model 1 were repeated. Comparisons of these data with theoretical results are shown in Figs. 5 through 7. Significantly the results for pressure loading of the sphere agree much better than for Model 1. The circumferential results, which are the largest for the sphere, are in acceptable agreement; however, considerable discrepancy still exists for the outer meridional stresses. As seen in Figs. 6 and 7, good agreement exists for axial thrust and moment loading, with the exception of the meridional stresses on the sphere. There the theoretical results vary somewhat from the experimental in contrast to the results for Model 1 (see Figs. 3 and 4).

Model 3 was obtained by cutting off the internal nozzle to a length of 0.75 in.* In Model 4 the length was reduced to 0.375 in., and in Model

*Since the CERL-II code cannot be used for analyzing short cylinders, the theoretical results for moment loadings in Models 3 and 4 were calculated on the assumption that the internal nozzle was infinitely long.

5 the internal stub was removed altogether. The results for these models are shown in Figs. 8 through 16. The same general agreement of theoretical and experimental stresses exists for these three models as for Model 2. It is significant that the length of the internal nozzle has little effect on the experimental data. This was also predicted by the theoretical results obtained from the USA code.

Model 6 was obtained by boring out the 0.125-in.-thick nozzle to a thickness of 0.0625 in. The results are shown in Figs. 17 through 19. The results for pressure loading are improved over those for the other models. Very good agreement exists for the circumferential stresses, with less agreement for the meridional stresses. The axial thrust and moment results agree satisfactorily, with less agreement for the outer meridional stresses on the sphere.

Additional Experimental Results

Models 2, 3, 4, and 5 were alike except for the length of inward protruding nozzle. The lengths of the protrusions are given in Table 1, along with dimensional and loading data for the six models. Stresses at points on both the outer nozzle and the sphere are plotted in Fig. 20 as a function of inner nozzle length for the three types of loadings. The locations for which data are presented are not those at which the highest stresses were found. The locations were chosen on the basis of having complete sets of data for them. They are such, however, that no change in curvature is indicated between the location in question and that of a stress recorded closer to the junction. The curves in Fig. 20 were obtained by joining the values for inner nozzle lengths of 0, $3/8$, $3/4$, and $4\ 1/8$ in. and are not consistent with the nozzle approaching a semi-infinite length.

The variations among the three loadings in Fig. 20 are significant. For pressure loading there is little effect due to the inner nozzle, except for the outer circumferential stress on the nozzle. This stress, which is the largest stress at that location on the nozzle, increased by over 20% as the length of the inner nozzle increased. This is in contrast to the results for axial loading, in which case the larger stresses

Fig. 20. Variation of Sphere Stresses (on the Left) and Outer Nozzle Stresses (on the Right) as a Function of Inner Nozzle Length.

changed very little, with the exception of the axial inner stress on the nozzle, which showed a considerable decrease as the inner nozzle length increased. For moment loading the larger stresses consistently decrease as the inner nozzle length is increased.

In the CERL-II code the stresses for moment loading are assumed to vary about the junction as the cosine of the angle measured from the plane of loading. Experimental stresses for moment loading were obtained at 15° intervals, and therefore the validity of the assumption made in the theoretical analysis may be examined.

The experimental results obtained for moment loading at a given location on the sphere and on the nozzle are plotted at 15° intervals in Figs. 21 and 22, respectively, for all six models. Also plotted is $\sigma_0 \cos \theta$, where σ_0 is the measured stress in the plane of loading and θ is the circumferential location about the junction measured from the plane of loading. Very good agreement exists between the experimental data and the assumed variation, except for some disagreement for circumferential stresses.

One loading on Model 1 consisted of a combined axial compressive thrust and internal pressure, as shown in Table 1. A comparison of some of these results and those obtained by superposing results from individual loads is given in Table 2. Only the comparisons of large stresses are meaningful, since experimental error combined with superposing large stresses of opposite sign can lead to large discrepancies. An inspection of the larger stresses in Table 2 reveals no real trend and indicates that the factors ignored in the assumption of superposition for the loading considered play no significant role when the vessel loadings are applied simultaneously.

Finally, to indicate the accuracy of the codes in predicting stresses for shells that are well within the thin-shell range, comparisons for two test models are given in the Appendix, where the ability to predict both magnitudes and distributions with thin-shell theory is examined.

Fig. 21. Variation of Stresses on the Sphere Around the Junction for Moment Loading as a Function of Circumferential Angle Measured from the Plane of Loading.

Fig. 22. Variation of Stresses on the Outer Nozzle Around the Junction for Moment Loading as a Function of the Circumferential Angle Measured from the Plane of Loading.

Table 2. Superposition of Pressure and Axial Thrust Loading on a Nozzle Attached to a Spherical Shell^a

Location ^b	Surface ^c	Stresses ^d (psi)				
		Pressure and Axial Thrust (A)	Pressure (B)	Axial Thrust (C)	B + C	(B + C)/A
5.4°	Outer	-4,875	14,354	-19,757	-5,403	1.11
	Outer	1,747	17,146	-15,857	1,289	0.74
	Inner	15,653	1,124	15,834	16,958	1.08
	Inner	14,176	8,347	6,580	14,927	1.05
	Outer	-6,076	13,209	-18,867	-5,658	0.93
	Outer	847	15,633	-14,390	1,243	1.47
	Inner	16,978	2,479	14,186	16,665	0.98
	Inner	14,693	8,964	5,486	14,450	0.98
0.31 in.	Outer	-5,703	11,034	-17,459	-6,425	1.13
	Outer	1,889	14,080	-12,798	1,282	0.68
	Inner	1,678	-6,762	8,934	2,172	1.29
	Inner	5,393	12,462	-7,220	5,242	0.97

^aThe pressure loading was 400 psi and the axial compressive load was 6000 lb on Model 1.

^bLocation of nozzles is given in inches from outer surface of sphere; location on sphere is given in degrees from apex.

^cThe same locations refer to duplicate gages.

^dAt a given location, the first stress is an axial or meridional stress and the second is a circumferential stress.

Conclusions

The most significant fact obtained from these comparisons is that for each configuration and all loadings the maximum experimental stresses are well approximated by the theoretical analyses. This is in spite of the fact that for Model 1 the experimental and theoretical stresses on the sphere were in considerable disagreement for pressure loading. It is also significant that in all the models there were no fillets. Thus, stress concentrations due to the absence of a fillet apparently do not affect the results to any great degree. It is also worth noting that for the thinner nozzles the agreement of theory and experiment for the sphere improves for the maximum values under pressure loading. For these cases (Models 2 through 6) the only significant disagreement is for the meridional outer surface stresses for the sphere, and this disagreement exists for all three loadings considered.

In addition both the CERL-II and USA codes were effective in general in determining the stress distribution in each model for internal pressure loading, axial thrust on the nozzle, and a pure bending moment applied to the nozzle. The stresses around the junction for moment loadings do vary as the cosine of the circumferential angle, with some discrepancies in the circumferential stresses for the thicker models. In some instances the stresses in the meridional direction decrease at a greater rate than the $\cos \theta$ function as the nozzle thickness becomes thinner. In all cases the $\cos \theta$ distribution is conservative.

The experimental and theoretical stresses are in considerable disagreement on the sphere for pressure loading in the configuration with the thickest nozzle. This may be caused by the nozzle not being a thin shell; this could allow the transverse shear deformation in the nozzle to effect the stress level in the sphere. The significant disagreements are in the meridional stresses on the outer surface, but the agreement improves as the nozzle becomes thinner. In fact, when the nozzle and sphere are well within the thin-shell range, very good correlations have been obtained (see Appendix).

The nozzle stresses are affected considerably by changing the length of the nozzle extension. Under pressure loading, as the length of the

nozzle extension is reduced, all the stresses remain approximately the same except for the outer circumferential stress, which is reduced as the length is reduced. When the nozzle extension is completely removed, all the stresses are reduced except for the circumferential inner surface stress, which remains the same. Under axial loading as the length of the nozzle extension is reduced, the axial inner surface stress increases and the axial outer surface stress decreases, while the rest of the stresses remain approximately the same. When the nozzle extension is removed completely, all the stresses are increased except for the circumferential inner surface stress, which remains almost the same. For moment loading, the stresses remain about the same as the length of the nozzle extension is reduced. When the nozzle extension is completely removed all the stresses increase, except for the inner surface circumferential stresses, which remain almost constant.

This discussion leads to the conclusion that the length of the nozzle extension has little effect on the main spherical shell and its effect on the nozzle depends on the type of loading. The greatest effect on the spherical shell stresses was caused by removing the extension altogether. In some cases, such as pressure loading, the stresses were reduced by removing the extension. For local loads (such as axial and moment loads) the overall stress level was increased.

In one test the vessel was subjected to internal pressure and axial thrust simultaneously. The results confirmed that stresses from the combined loadings could be obtained by superposition.

In the theoretical analyses the shells are joined at the intersection of the middle surfaces, and stresses are computed from this point. For comparing theoretical and experimental results, this type of junction analysis may not be preferred (unless the shells are very thin shells); however, from a design standpoint it is not significant.

Appendix. Comparison of Theoretical and
Experimental Stresses on Two Thin-Shell Models

The results from two experimental shell-model tests are compared here with results of theoretical analyses to illustrate the accuracy of the computer codes for thin-shell models. A sketch of the first model is shown in Fig. A1. This was a photoelastic model and is identified as S-5AZ in Ref. 7. The final results for the model became available after publication of the reference, however. This model was chosen not only because of the dimensions but also because there was only a nominal fillet at the junction. With this model, experimental data were obtained to the intersection of the middle surfaces at the junction. The model was subjected to a pressure load of 0.056 psi, and the mechanical properties for the sphere and cylinder were taken as $E = 3410$ psi and $\mu = 0.5$, where E is elastic modulus and μ is Poisson's ratio.

Comparisons of theoretical results obtained by the USA and CERL-II codes with the experimental results are shown in Fig. A2. The location of the outer surface at the junction is shown by the dashed lines on the spherical shell plots, while the experimental stresses within the wall at the junction are plotted to the left of the zero ordinate in the cylindrical shell plots. The results compare favorably, except for the stresses inside the junction.

The second model was IITRI model S-1.⁸ The spherical shell had a radius of 11.899 in. and a thickness of 0.101 in., and the cylindrical portion had a radius of 5.90 in. and a thickness of 0.100 in. The radius-to-thickness ratio for the spherical shell was 108 and for the cylindrical shell, 59. Both shells were well within the thin-shell range. The junction between the cylinder and sphere had a fillet radius of $1/8$ to $5/32$ in. For an internal pressure of 60 psi the comparison of theoretical and experimental results is shown in Fig. A3. The stresses compare favorably. For an outward axial thrust on the nozzle of 6000 lb and an applied moment on the nozzle of 18,000 in.-lb, the comparisons are shown in Figs. A4 and A5. The agreement is very good. In particular, Figs. A2 and A3 show that for internal pressure loading the computer codes can give accurate results for models within the thin-shell range.

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Fig. A1. Cross-Sectional View of Model S-5AZ.

Fig. A2. Comparison of Theoretical and Experimental Stresses on Model S-5AZ for an Internal Pressure of 0.056 psi.

Fig. A3. Comparison of Theoretical and Experimental Stresses on IITRI Model S-1 for an Internal Pressure of 60 psi.

Fig. A4. Comparison of Theoretical and Experimental Stresses on IITRI Model S-1 for an Axial Load on the Nozzle of 6000 lb Outward.

Fig. A5. Comparison of Theoretical and Experimental Stresses on IITRI Model S-1 for a Pure Bending Moment of 18,000 in.-lb on the Nozzle.

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- 148. K. N. Tong, Syracuse University, Syracuse, N. Y.
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- 172. Research and Development Division, AEC, ORO
- 173-174. Reactor Division, AEC, ORO