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Local Stresses in Vessels—Notes on the Application of WRC-107 and WRC-297

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The welding Research Council (WRC) Bulletin No. 107 [1] and Bulletin No. 297 [2] are two of the most important design guides ever published for the design of pressure vessels. Widely used in the design of attachments and nozzle connections, they have become indispensable tools in recent years.

WRC-107 and WRC-297 are invaluable due to their combined consideration of theory, experimental data, and engineering judgment. In some instances the theoretical values have been adjusted several hundred percent upward to match the available experimental results. They are reliable tools without which the so-called "Design by Analysis" approach would have been impractical. Unfortunately many designers have misapplied the data presented, thus resulting in inconsistent designs. This article describes the data available in these two bulletins and explains the nature of inconsistency incurred in some designs. A supplemental formula is then developed for calculating the combined maximum stress intensity to be used in designs.

WRC Bulletin No. 107. WRC-107 was first published in August 1965 [1]. It was based on Professor Bijlaard's theoretical work, with some adjustments made based on available experimental data. A few revisions have been made since its first publication. The latest revision made in March 1979 relabeled some of the curves. WRC-107 has been used widely in the design of vessel nozzles and attachments. It was one of the major driving forces in promoting the "Design by Analysis" philosophy. The bulletin covers both spherical and cylindrical vessels.

In spherical vessels, the original theoretical work was based on round rigid inserts and round nozzle connections. Square inserts or connections can be analyzed using an equivalent round attachment having a diameter equal to 8/7 of the attachment width. On the other hand, the original theoretical work for the cylindrical vessel was based on square and rectangular-shaped uniform loads acting on unperforated vessels. Round attachments can be analyzed using an equivalent square having the width of 7/8 of the attachment diameter. Because of the assumption of the unperforated shell, the applicability of the cylindrical vessel portion of the bulletin is limited to rigid inserts. The Bulletin does not recommend any specific method in analyzing an actual nozzle connection in the cylindrical vessel. It is left largely to the designers to make their own judgment.

WRC-107 presents detailed tabular forms for calculating stresses at four major axis locations. Stresses at both inside and outside surfaces on these locations can be readily calculated following the step-by-step procedure outlined in the form. The final results are the total skin stress intensities at these four locations in the shell. No separate membrane stress intensity is given, nor is the stress in the nozzle calculated.

WRC Bulletin No. 297. WRC-297 was published in August 1984 [2]. It is a supplement to WRC-107 and is specifically applicable to cylindrical nozzles in cylindrical vessels. This bulletin was based on Professor Steele's theoretical work. It gives data for larger D/T ratios than in WRC-107, and also provides better readability for small values of d/D by plotting the curves using $\lambda = d/\sqrt{DT}$ as the abscissa. Most importantly, the new theory considers the opening on the shell together with the restraining effect of the nozzle wall. This is a better model than the unperforated shell used in WRC-107 for simulating the nozzle connections in cylindrical vessels.

Because most pressure vessel and piping codes have different allowable stress criteria for different stress categories, WRC-297 emphasizes the separation of membrane stress and skin bending stress. Two examples are given outlining the detailed procedures for calculating membrane and total stress intensities at points located in the longitudinal and the transverse planes. Both shell and nozzle stresses are calculated.

Location of Maximum Stress

The fact that the examples given in both WRC-107 and WRC-297 specifically outline the procedures for calculating stresses at the four major axis corners, has led designers to think that the maximum stress in the connection must be one of those stresses. This presumption introduces inconsistency and nonconservatism in the design of nozzle and attachment connections. The maximum stress is not normally located at these corners. Although the calculation involves only the secondary stress which itself involves various uncertainties, a

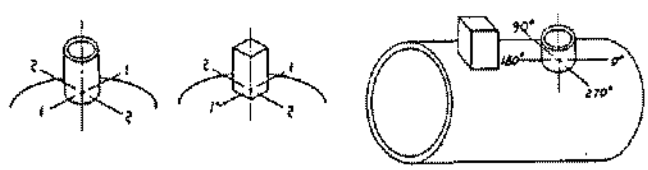

Fig. 1 Vessel attachments

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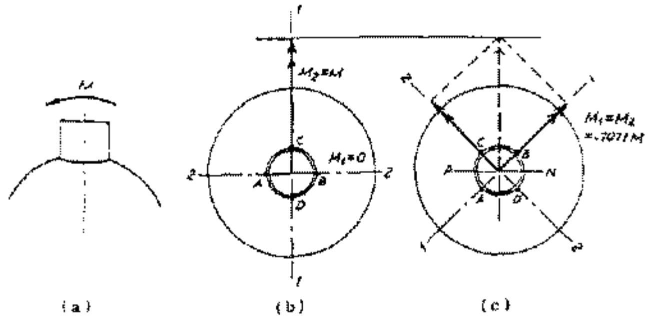

Fig. 2 Stress on spherical shell

certain amount of deviation is always expected. For instance, as much as a 10-percent difference may be made just from reading the chart by different persons. However, just because of its inherent uncertainty, effort is needed to make it as consistent as possible. Anything that can be done to improve its certainty should be done. In cases when deviation is unavoidable, it is preferred to deviate on the conservative side.

To demonstrate when an inconsistency may occur, the attachment on a spherical shell can be used as an example. Figure 2(*a*) shows a bending moment, M, acting on a nozzle connection at a spherical vessel. By choosing a coordinate system as shown in Fig. 2(*b*), the stresses at the four major axis points are SA = S, SB = S, SC = 0, and SD = 0. However, if a designer happens to have the coordinate system set up as shown in Fig. 2(*c*), the applied moment will be decomposed into $M_1 = M_2 = 0.7071M$ two components. The stresses at the four major axis points, in this case, are SA = SB = SC = SD = 0.7071S. This stress is about 40 percent below the expected maximum stress. The same nozzle connection and the same applied moment, yet the calculated stresses are substantially different depending solely on how the coordinate system is set up. From the foregoing demonstration, it is clear that in Fig. 2(*c*), the maximum stress is not located at the major axis points, but at the off-axis points P and N. In general, if the moments acting around both coordinate axes are nonzero, the maximum stress is not located at the major axes. This warning was properly stated in Paragraph 3.3.5, WRC-107, which said, "However, in the general case of arbitrary loading, one has no assurance that the absolute maximum stress intensity in the shell will be located at one of the eight points considered in the above discussion." The eight points mentioned are the inside and outside surfaces of the four major axis points. Strangely, this message has been largely ignored.

Maximum Stress Intensity

The calculation of the stresses at the four major axis points is helpful in understanding the stress distribution. However, for the design purposes it is essential to calculate the maximum stress intensity occurring throughout the entire connection. In developing the calculation procedure, the following notations are used:

P = radial load
M_C = circumferential moment (or moment in 1-1 direction)
M_L = longitudinal moment (or moment in 2-2 direction)
M_B = total bending moment
M_T = torsional moment
V_C = circumferential force (or shear force in 2-2 direction)
V_L = longitudinal force (or shear force in 1-1 direction)
V = total shear force
$S_{ij}(L)$ = i direction j category stress due to load L; $i=r$, θ, for radial and circumferential, respectively; $j=m,b$, for membrane and bending, respectively
SS = shear stress

The applied loading can generally be divided into four component groups. They are radial force, bending moment, torsional moment, and shear force. In the following discussion the stress contributed by each component group is described first. They are then combined to become the maximum stress. The discussion follows the WRC-297 stress orientation of radial and circumferential directions with respect to the nozzle. Because of the different stress orientation adopted in the cylindrical shell portion of the WRC-107, reorientation of the WRC-107 data is required for the cylindrical vessel. The procedure shows the method for calculating the stress in the shell. The same procedure can be used for calculating the stress in the nozzle.

Stress due to P. In a spherical vessel, it is obvious that the stresses created by the radial load are uniform around the entire attachment circumference. In a cylindrical vessel, though the stresses differ from location to location, it can also be regarded as uniform taking the maximum around the attachment circumference as the uniform value. This assumption introduces some conservatism but is not overly conservative. WRC-297 has already adopted this approach in developing the design curves. In reference to the attachment orientation, the stress created by the radial load can be written symbolically as

$$\begin{aligned} &\text{Membrane stress:} \quad S_{rm}(P),\ S_{\theta m}(P) \\ &\text{Bending stress:} \quad S_{rb}(P),\ S_{\theta b}(P) \end{aligned} \tag{1}$$

They are constant around the entire attachment circumference.

Stress due to M$_C$ *and* M_L. Stresses created by M_C and M_L are not independent as assumed by some designers. In a spherical vessel the M_C and M_L can be conveniently combined as

$$M_B = \sqrt{M_C^2 + M_L^2} \tag{2}$$

The combined stress can then be calculated based on M_B, rather than on M_C and M_L individually. However, to have a common method applicable to both spherical and cylindrical vessels, a more general approach is preferred. Professor Bijlaard has shown that the stress, due to bending moment, varies according to the cosine function in a spherical shell. As shown in Fig. 3(*a*), when the attachment is loaded simultaneously with M_C and M_L, each M_C and M_L commands a cosine shape stress distribution. With this type of cosine distribution, it can be shown that the maximum combined

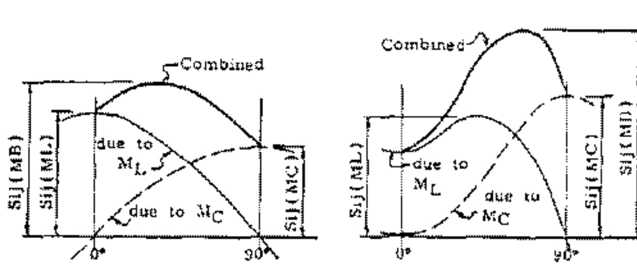

Fig. 3 Combined stress due to bending moments

stress is

$$S_{ij}(M_B) = \sqrt{(S_{ij}(M_C))^2 + (S_{ij}(M_L))^2} \quad (3)$$

The foregoing maximum stress located somewhere inside the 0–90-deg segment. There is another maximum stress, with a reversed sign, located within the 180–270-deg segment.

In a cylindrical vessel, the situation is complicated by the irregular distribution of the stress. The stress due to M_C is distributed in a shape close to a shifted cosine curve, but the stress due to M_L is humped toward the neutral axis. Due to this off-axis peaking, it appears that an absolute sum may have to be taken to calculate the combined maximum stress. Nevertheless, in considering the fact that the stress field due to M_C is considerably narrower than a cosine distribution, equation (3) can still be used for cylindrical shells with good representation. In fact, this equation has been used by the piping code [3] since the 1950's.

Since the purpose of the calculation is to find the maximum stress intensity, the relative signs between the radial stress and the circumferential stress is important. Fortunately, this sign reversal only occurs at some of the circumferential mem' ·ane forces in WRC-297. One way of maintaining the sign is to take the $S_{ij}(M_B)$ in equation (3) the same sign as that of the greater $S_{ij}(M_C)$ and $S_{ij}(M_L)$. Even with this sign-preserving arrangement, the maximum membrane stress intensity calculated may still be smaller than the ones calculated at the four major axis points. However, the difference is insignificant. The stresses calculated at the four major axis points still need to be considered.

Combined Normal Stress. The combined maximum normal stress is determined by P, M_C, and M_L. Since the stress due to P is uniform all around the attachment circumference, we can simply write

$$S_{ij} = S_{ij}(P) + S_{ij}(M_B) \quad (4a)$$

$$S_{ij} = S_{ij}(P) - S_{ij}(M_B) \quad (4b)$$

Equations (4*a*) and (4*b*) represent the maximum normal stresses at the two maximum points located on opposite sides of the attachment. Each equation further represents two stresses one at the outer, and the other the inner surface of the shell. These four locations are to be checked for the maximum stress intensity.

Shear Stress due to M_T. The shear stress due to torsional moment is uniform all around the attachment circumference. This stress can be expressed as $SS(M_T)$.

Shear Stress due to V_C *and* V_L. The shear stress due to V_C and V_L can be combined by

$$SS(V) = \sqrt{(SS(V_C))^2 + (SS(V_L))^2} \quad (5)$$

Total Shear Stress. The total maximum shear stress is the absolute sum of the shear stress due to torsion and the shear stress due to combined shear force. That is,

$$SS = SS(M_T) + SS(V) \quad (6)$$

This maximum shear stress generally does not occur at the same location as the maximum normal stress. However, since the shear stress is insignificant in most of the cases, it can be conservatively considered as occurring at the same location where the maximum normal stress occurs.

Maximum Stress Intensity. The stress intensity can be calculated by the maximum shear stress theory using the normal stress and shear stress calculated by equaitons (4) and (6), respectively. The WRC bulletins have given detailed formulas for this calculation. A total of four stress intensities representing the maximum and minimum stress points and both outside and inside surfaces should be calculated. The maximum value is then used for the design. To satisfy certain Code [4] requirements, the maximum membrane stress intensity and the total stress intensity may also need to be separated.

Conclusions

Regardless of the warning given by the WRC Bulletin 107 that there is no assurance that the absolute maximum stress intensity in the shell will be located at one of the eight points (four major axis points each having outside and inside surfaces) considered in the example calculations, many designers still use only the stresses calculated there for design. This practice creates inconsistencies in designs and may introduce as much as a 40-percent nonconservatism. The present article outlined the procedures for calculating the maximum stress intensities both at and off the major axis points. This maximum stress intensity should be used in the design evaluations.

References

1 Wichman, K. R., Hooper, A. G., and Mershon, J. L., "Local Stresses in Spherical and Cylindrical Shells Due to External Loadings," WRC Bulletin No. 107, Aug. 1965, revised Mar. 1979.

2 Mershon, J. L., Mokhtarian, K., Ranjan, G. V., and Rodabaugh, E. C., "Local Stresses in Cylindrical Shells Due to External Loadings on Nozzles—Supplement to WRC Bulletin No. 107," WRC Bulletin No. 297, Aug. 1984.

3 ANSI Code for Pressure Piping, ANSI/ASME B31.3 Chemical Plant and Petroleum Refinery Piping, ASME, New York, 1984.

4 ASME Boiler and Pressure Vessel Code, Section VIII, Pressure Vessels, Div. 2, Alternative Rules, ASME, New York, 1983.

DISCUSSION

R. Natarajan[2]

At the outset, I would like to congratulate the author for bringing out certain important points which a designer sometimes forgets while using design charts. However, there are some points which are worth mentioning about this paper:

1 While discussing the inconsistency about the location of the maximum stress in a nozzle-spherical sheet intersection, it is expected that the designer will define the geometry and the loading using the same coordinate system. The location of the maximum stress, and hence the inconsistency in defining the maximum stress location, is due to the misunderstanding by the designer and not due to the examples given in WRC-107 or WRC-297.

2 While calculating the combined stress due to bending moments, mention should be made that the flexibility of the nozzle has not been completely considered. Further, the boundary conditions at the nozzle and cylinder ends also affect the value and location of these maximum values.

K. Mokhtarian[3]

I have the following general comments to make on Peng's paper:

1 We have found that generally the maximum stress due to a longitudinal moment occurs at the 0-deg azimuth. We do not agree with the shape of the stress curve due to M_L in Fig. 3(*b*).

2 The last three sentences of the last paragraph in the subsection "Stresses due to M_C and M_L" are not clear and appear to contain conflicting statements.

3 Normally, the designer has to face the question of combining the stresses due to pressure with those due to

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mechanical loads. I do not know of any simple way of providing those guidelines now, but eventually this question will have to be addressed.

Z. F. Sang[4]

As stated in the paper by L. C. Peng, WRC-107 and WRC-297 published by PVRC are excellent references for calculating local stresses in nozzles and attachments. Indeed, they are widely used in the design of pressure vessels and have become indispensable tools.

The author summarizes inconsistencies occurring in some designs due to the designers misapplying the data presented in the aforementioned two documents. He also presents a method and procedure for calculating the maximum stress intensity. This is of importance and needs to be understood by designers. It should prove to be an aid in applying the two documents correctly.

I am in agreement with Dr. Peng's opinion about the inconsistency and nonconservation, which will be created in the design procedure if a designer cannot determine the maximum stress intensity. In the paper, the formula which is developed for calculating intensity seems to hold only for round radial nozzles and attachments on spherical shells. Only in this case are the stresses due to radial load P and torsional moment M_T uniform. For other shapes, particularly in the case of a rectangular attachment on a cylinder, the stresses are not uniform along the perimeter of the attachment.

In the section "Location of Maximum Stress" the author states that "the calculations involve only the secondary stress." From a stress classification point of view, stresses due to external load on an attachment include not only secondary stresses, but also primary ones. This is important, because there are different allowable stresses associated with different stress categories.

With reference to the calculation of the maximum stress intensity, it is noted that the maximum shear stress generally is not located at the same point where the maximum normal stress occurs. But the author assumes that they do occur at the same locations. Is this a conservative assumption?

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AUTHOR'S CLOSURE

In thanking Messrs. R. Natarajan, K. Mokhtarian, and Z. F. Sang for the valuable discussions, the author would like to make a brief closure.

This paper's main concern is the misapplication of the bulletins, not the validity of the bulletins which are excellent works. The nozzle flexibility and the vessel end condition, just as other geometrical parameters, have definite effects on the stress shape. The main point is if the interaction exists between the two moment components.

The off-axis peak stress due to M_L may not exist on small d/D vessels, but it does exist on other vessels, as demonstrated by Prof. Steel, and various pipe branch tests. There is indeed some confusion in the last three sentences concerning the stresses due to M_C and M_L. Because of the combination method proposed, the stress loses the orientation after the calculation. With the proposed sign tracking method, the maximum calculated membrane stress intensity may be occasionally smaller than the stress calculated at the four major corners. One way to correct the problem is to reverse one of the stress signs when the situation is detected. The author agrees that there is no simple way to combine the pressure and the mechanical load effects. Publication of some of the NRC approved methods, for instance, should be encouraged.

The secondary stress mentioned by Dr. Sang should have been more accurately stated as local stress. The inclusion of higher shear stress is always conservative in the calculation of the stress intensity when it is taken as twice the maximum shear stress.

The above and the pressure membrane stresses have been incorporated in SIMFLEX and LOCALS and LOCALC.