DISCUSSION

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In the Introduction, the author states the ASME Code [1] contains two errors. However, a comparison of the author's proposed analysis method with the Code method reveals that the Code is not in error, but that there are differences in the assumptions made in the development of the design equations. The differences in the assumptions concern: 1) the significance of considering or neglecting the effect of the shell thickness in the development of the equations for calculating the moment distribution in the vessel; and 2) on how much affect the area out for the holes in a perforated side plate has on the bending stiffness, I_2, for the side plate, and subsequently the moment distribution in the vessel.

The design philosophy of Division 1 of Section VIII of the Code does not require a detailed evaluation of the higher, more localized stresses which are known to exist; but insofar as practical design rules for details are written to limit such stresses to a safe level consistent with experience (UG-23(c)). In developing the moment equations, one of the goals was to keep the equations as simple as possible while still providing a satisfactory degree of accuracy within the Code design philosophy. Therefore, the effect of the shell thickness was intentionally neglected. In the author's proposed equations, the forces due to the membrane stresses were considered to act in the center plane of the side plates, and this resulted in additional terms being added to the moment equations.

A comparison of the stresses in the first two examples shows that the stresses calculated by the Faupel and Code equations only differ by 382 psi under to 271 psi over those calculated using the author's proposed equations. Therefore, the simpler equations appear to be satisfactory for these vessels, which have a ratio of span to depth of nine or greater for each side.

In the third example where the shell thicknesses are much greater, the differences between the stresses calculated by the two methods are much greater, as would be expected, with the Code stresses being 2813 psi under to 1687 psi over that calculated using the author's equations. However, in this vessel, the accuracy of the stresses obtained using either set of equations is questionable, since the span-to-depth (thickness) ratios are $h/t_2 = 5$ and $H/t_1 = 3$; and according to Roark [2], when the span-to-depth ratio is less than eight, the beam formulas yield results that at best are approximate and may be grossly in error. Therefore, vessels with span-to-thickness ratios less than eight may require more detailed analyses which consider factors other than just the bending effects on which the Code and the author's analyses are based.

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In order to consider the effect of a reduction in the bending stiffness due to holes in the side plates, the author proposes to multiply I_2 by the ligament efficiency factor to obtain $I_2 = et_2^3/12$ for use in all equations when determining the moment distribution in the vessel. Multiplying I_2 by e has the same effect as cutting the side plates into (vertical) strips with effective widths equal to 6 $(p-d)$ and with gaps between the strips equal to the diameter of the holes. This results in an overcorrection for the bending stiffness and induces errors in the calculated moment distribution for the top and bottom plates, as well as in the side plates, which for practical ligament configurations will be much greater than if the effect of the holes on the bending stiffness is neglected, (i.e., using $I_2 = t_2^3/12$). The overcorrection will be even greater if the actual number of rows of holes (horizontally) is less than the maximum number of rows possible (assuming the vertical pitch is the same as the horizontal pitch).

The author also proposes to disregard the effect of the holes on I_2 when calculating the bending stresses by using $I_2 = t_2^3/12$. This would result in very unconservative and unrealistic calculated stresses in the perforated side plates, as can be seen by comparing the stresses shown in the corrected Summary of Stresses for Example 4. In the Faupel and Code equations for calculating the bending stresses, the effect of the area out for the holes is accounted for at section m by the value used for E, which by definition is the lessor of: 1) the joint efficiency per Paragraph UW-12 of the Code, or 2) the ligament efficiency e_b or e_m. In the summary of stresses for Example 4, the author used $E = 1.0$ instead of $E = e_b = e_m = 0.60$ when calculating the bending stresses t section m by Faupel's and the Code method. (Reference [1] example in 13–17 (*a*).)

Summary of stresses for Example 4

Location	Equation no.		Stresses (psi)			
	Faupel	code	Faupel eqs.			Author eqs.
			$K = 1.80$	$K = 1.82$	$K = 1.08$	$K = 1.08$
n	28	3	4971	4981	7646	8154
$q1$	29	4	12,920	12,930	15,593	16,103
m	30	5	10,672	10,695	9628	9850
			▷17,787	▷17,825	▷16,045	▷16,416
$q2$	31	6	5047	5025	6091	5870

$E = 1.0$ when there are no holes, all butt welds are Type 1 per Table UW-12, and all butt joints are radiographed.
▷$E = 0.60$ at Section m for perforated side plates.

Summary

- The author's proposal to consider the effect of the shell thickness in the equations for calculating the moments in the shell is a refinement in the analysis, but the difference in the stresses calculated by the Code equations and the author's proposed equations is small for vessels with span-to-thickness ratios greater than eight. Therefore, the addition of the thickness terms to the basic moment equations in the Code does not appear to be justified. However, to cover any cases where the designer feels the extra calculations are warranted, consideration should be given to adding provisions in the Special Calculations section of the Code (Appendix 13) for including the shell thickness effects. A statement should also be added to limit the use of the equations to vessels with span-to-thickness ratios greater than eight.
- Reducing the bending stiffness of a perforated plate by multiplying the moment of inertia (without perforations) by the ligament efficiency factor ($I_2 \times e$) as proposed by the author results in overcorrection errors, which are greater than the errors incurred when the Code method of neglecting the effect of the perforations on the bending stiffness is used.
- When calculating the bending stresses at the midpoint of a perforated side plate or at any other section containing a

row of holes, the moment of inertia (without perforations) must be multiplied by the ligament efficiency factor to account for the area out for the holes.

References

1 ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, 1980 Edition.

2 Roark, R. J., and Young, W. C., *Formulas for Stress and Strain*, Fifth Edition, 1975, p. 89.

N. Gilbert[3]

In his paper the author presents two contentions; namely: 1) the equations in Appendix 13 of the ASME Pressure Vessel Code Section VIII, Division 1, related to rectangular square-cornered vessels are in error – in the nonconservative direction; and 2) the equations in Appendix 13 do not account for the reduced stiffness of side plates resulting from holes (for tubes or plugs).

The basis for the author's first statement is his derivation of the moments in a square-cornered box section in which he considers the corners as separate elements instead of being rigid attachments of the side elements. This is a refinement which is not considered in conventional rigid frame analysis because its effect is negligible until the length-to-thickness ratios of the side plates become small. Examples 1 and 2 of the "Comparison of Results" consists of box headers with ratios in the order of 13:1 and 32:1 and the maximum deviation is 4 percent with Appendix 13 yielding lower stresses. A deviation of this magnitude is well within the margin of safety and can be ignored. In Example 3 the author intends to demonstrate the discrepancy when the ratios are smaller – $h/t_2 = 5$ and $H/t_1 = 3$. This example shows the Code formula being 16 percent lower than the proposed formula for the controlling stress at the short-side corner location. It is important to note that these proportions (ratios of 5 and 3) are outside the acceptable limits for the flexure stress formula, which is the basis for the bending stress in the analysis. It is my opinion that the errors in applying the flexure equation to such very short beams may exceed the 16 percent revealed in the author's exercise. Although the short-beam errors in applying the flexure formula are probably nonconservative, the local yielding resulting from the high local stresses (beyond the yield strength) causes the structure to "shake down" and strengthen after the first few load cycles. Because of this phenomenon it is common practice to use the (elastic) flexure equation in the design of short beams without concern for the actual high stresses in local regions. I might add that a finite element analysis is required for a more precise determination of the actual complex stress system. However, since were are concerned with a practical design procedure and not a rigorous stress analysis, no changes in Appendix 13 need be considered. In short, the "error" in stress revealed by the author in Example 3 is meaning less because the basic technique (namely, use of elastic analysis and the flexure equation) is not applicable to short beam stress analysis.

In Example 4 the author feels compelled to demonstrate that the stresses are different between a vessel with a solid-wall long-side plate and one that is perforated. As would be expected, the perforated model reveals reduced stresses in the middle of the short-side plates and in the corners. Of course, the important consideration is: how much reduced stiffness is produced by two or four rows of holes? Certainly it is not as expressed by the author in Example 4. Revising the factor K by the ligament efficiency e as a multiplier assumes that the stiffness of a perforated plate is the same as for one consisting of slots (instead of holes) extending the entire width (h) of the plate. This, of course, is very unrealistic and cannot be considered as an appropriate model of a plate with 2 or 4 rows of holes. Also, overestimating the reduced stiffness of the long side plates results in an unconservative stress in those plates. In Conclusion 2 the author recognizes this dilemma and suggests a design procedure which can result in grossly overconservative stresses.

In summary, the following comments are pertinent:

- The subject paper presents a refinement in the analysis of rigid frames which is of negligible effect, and consequently is ignored in conventional structural analysis. In the range of h/t ratios where the refinement may show an appreciable effect, the use of the flexure equation becomes unacceptable for the determination of stress (particularly, the maximum stress). For this reason, I recommend that no changes to Appendix 13 be made on the subject.
- The author's second point concerning the reduction of stiffness in plates containing rows of holes is valid and was recognized by the Working Group when formulating the equations in Appendix 13. This is the reason for using the moment of inertia, I, in the equations instead of the conventional $t^3/12$ for solid plates. It is true that no specific explanation exists in the text of Appendix 13 (although this effect is mentioned in Paragraph 13-14(a) (2) for external pressure) and the reduction in stiffness resulting from a row of holes is ignored in the examples of 13-17. In light of this, I recommend that Appendix 13 expand on this and add some cautionary notes and perhaps some guidelines for the designer. However, the author's suggestion that the entire plate be treated as slotted is not an acceptable procedure – especially when it can lead to unconservative stress in the side plates containing the holes.

In conclusion, it might be added that Appendix 13 of the ASME Boiler and Pressure Vessel Code Section VIII, Division 1 was developed by a Working Group of six members (Dr. Joseph H. Faupel, Chairman). The procedures were based on established techniques in structural design and analysis. In addition to the member's input, the progress of the work was monitored by the ASME Boiler and Pressure Vessel Committee at large. Also, individuals representing a wide spectrum of interest and expertise attended the public meetings of the Working Group and followed its progress. This Working Group is still in force and is continually receptive to critical discussion. If one has a disagreement with any of the Code's procedures or rules, it is advisable to address it to the Boiler and Pressure Vessel Committee as indicated in the Foreword of the Code.

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