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Stress Analysis of a Vessel Due to Nozzle Loads

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Abstract

The associated stresses and deformations developed due to thermal piping loads resulted in significant deformation of the pressure vessel shell. A typical pressure vessel and piping configuration is examined in this paper, and it is proposed a comparisons between WRC 107 and FE analysis aproach.

Keywords: *pressure vessel, piping loads, WRC 107, finite element analysis.*

Introduction

Loads from piping attached to vessels induce in the vessels walls, in the form of membrane and bending stresses. These stresses normally must be evaluated against the requirements of the ASME Boiler and Pressure Vessel Cod, Section VIII, Division 2. Accurate calculation of stresses in a vessel wall is difficult without a finite element analysis; the best means of doing a calculation otherwise is to use a reference which parameterizes results of finite element analyses. The most common reference of this type is Welding Research Council Bulletin 107.

Welding Research Council Bulletin No.107

Welding Research Council Bulletin No.107 is a parameterization of the results of a set finite element analyses examining stresses in vessels due to loaded attachments. WRC 107 contains equations and non-dimensional curves (based upon parameters such as ratios of the nozzle to vessel diameter and the vessel diameter to vessel thickness) which are used to extract coefficients for the calculation of stresses in the vessel wall at the point of attachment. Note that WRC 107 computes stresses in the vessel shell at the nozzle/vessel interface – stresses in the nozzle wall (which in some cases can be higher than the stresses in the vessel wall) are not computed. Stresses in the nozzle wall may become greater than the stresses in the vessel wall as the t/T (nozzle to vessel thickness) ratio becomes less than one[1]. The WRC-107 Analysis calculates thecombined primary membrane and local stress (primary local or secondary stressdepending on many factors) at the junction of an attachment and a shell or head.

Before attempting to use WRC 107 to evaluate the stress state of any nozzle/vessel junction, one shall always make sure that the geometric restrictions limiting the application of WRC 107 are not exceeded.

Some criteria when a finite element analysis is beneficial are listed below[3]:

1) When the d/D ratio for a loaded nozzle is greater than 0.5 and WRC 107/297 is considered for use;
2) When the t/T ratio for a loaded nozzle is less than 1.0 and WRC 107/297 is considered for use;
3) When the nozzle is pad reinforced and WRC 107/297 is considered for use;
4) When the number of full range pressure cycles is greater than 7000 cycles and the nozzle is subject to external loads;
5) When the D/T ratio is greater than 100 and SIFs or flexibilities are needed for a pipe stress program;
6) When the D/T ratio is greater than 100 and a dynamic analysis including the nozzle is to be performed using a piping program;
7) When a large lug is used in a heavily cyclic service;
8) When pad-reinforced lugs, clips, or other supports are placed on the knuckle radius of a dished head;
9) When seismic horizontal loads on vessel clips or box supports are to be evaluated;
10) Pad reinforced hillside nozzles subject to pressure and external loads;
11) Large run moments, but small branch moments in a piping system;
12) Overturning Moments on Skirts;
13) Effect of Integral vs. Non-Integral Pad on Nozzle in Head Should be Studied;
14) Different thermal expansion coefficients or temperatures between the header and branch;
15) Where loads on nozzles are high because of the assumption that the nozzle connection at the vessel is a rigid anchor;
16) Heat Transfer in An Axisymetric Model Geometry;
17) When the effect of adding a radius to weld geometries on nozzles in heads should be investigated;
19) For saddle supported horizontal vessels with or without wear plates including tapered saddles with many design options;
20) To evaluate effects of axial or transverse loads due to internal sloshing, wind loads, seismic loads, or general external loads. Zick's methods do not consider axial or transverse loads;
21) Design of Pipe Shoes for self-weight, liquid weight, and external loads.

WRC-107 coordinate system

For both heads and cylinders as hosts, the radial load P is positive if it is inward. For heads is need to choose arbitrary 1-1 and 2-2 axes that are normal to each other. A shear load V2 acts in the 1-1 direction and causes the M1 moment. A shear load V1 acts in the 2-2 direction and causes the M2 moment, figure no. 1.

For cylindrical hosts the axes are the longitudinal direction and the circumferential direction. In this case a positive shear load VC acts in the positive circumferential direction and creates the positive moment MC. The positive shear load VL acts in the positive longitudinal direction and creates the positive moment ML.

ASME Setion VIII Division 2 – Elastic Analysis of Nozzle

Ideally in order to address the local allowable stress problem, the user should have the endurance curve for the material of construction and complete design pressure/temperature loading information. If any of the limits are approached, or if there is anything out of the ordinary about the nozzle/vessel connection design, the code should be carefully consulted before performing the local stress analysis.

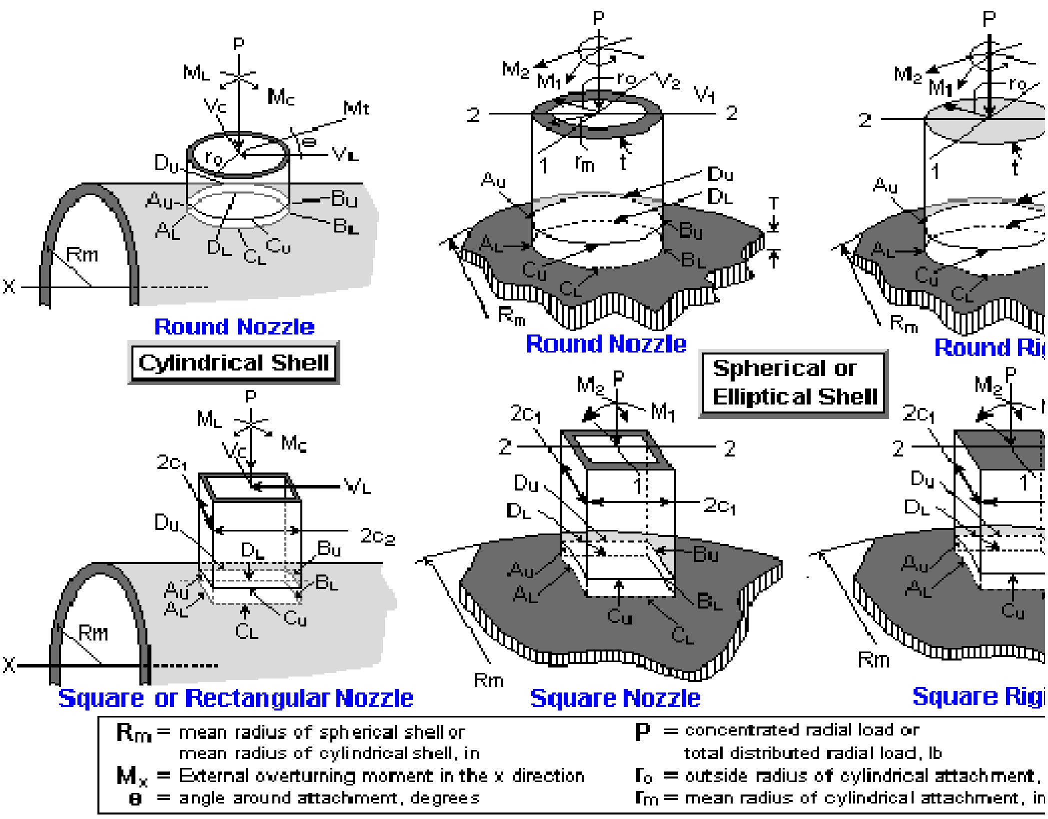

Fig. 1. WRC coordinate system.

There are essentially three criteria that must be satisfied before the stresses in the vessel wall due to nozzle loads can be considered within the allowable. These three criteria can be summaried as[1]:

$$P_m < kS_{mh} \quad (1)$$

$$P_m + P_l + P_b < 1.5kS_{mh} \quad (2)$$

$$P_m + P_l + P_b + Q < 3S_{mavg} \quad (3)$$

Where P_m , P_l , P_b and Q are the general primary membrane stress intensity, the local primary membrane stress intensity, the local primary bending stress intensity, and the total secondary stress intensity (membrane plus bending), respectively; and k, S_{mh} and S_{mavg} are the occasional stress factor, the hot material allowable stress, and the average material allowable stress intensity $(S_{mh} + S_{mc})/2$.

Due to the stress classification defined by Section VIII, Division 2 in the vicinity of nozzles, the bending stress terms caused by any external load moments or internal pressure in the vessel near a nozzle or other opening, should be classified as Q, or the secondary stresses, regardless of whether they were caused by sustained or expansion loads. This causes P_b to disappear, and leads to a much more detailed classification:

P_m – General primary membrane stress intensity (primarily due to internal pressure);

P_l – Local primary membrane stress intensity, which may include:

Membrane stress due to internal pressure;

Local membrane stress due to applied sustained forces and moments.

Q – Secondary stress intensity, which may include:

Bending stress due to internal pressure;
Bending stress due to applied sustained forces and moments;
Membrane stress due to applied expansion forces;
Bending stress due to applied expansion forces and moments;
Membrane stress due to applied expansion moments.

Each of the stress terms defined in the above classifications contain three parts: two stress components in normal directions and one shear stress component. To combine these stress, the following rules apply:

1) Compute the normal and shear components for each of the three stress intensities, i.e. P_m, P_l, and Q;
2) Compute the stress intensity due to the P_m and compare it against kS_{mh};
3) Add the individual normal and shear stress components due to P_m and P_l; compute the resultant stress intensity and compare its value against $1.5S_{mh}$;
4) Add the individual normal and shear stress components due to P_m, P_l and Q, compute the resultant stress intensity and compare its value against $3S_{mavg}$.
5) If there is an occasional load as well as a sustained load, these types may be repeated using a k value of 1.2.

The procedure for checking stresses in vessel shells using WRC 107 can be summarized as follows:

Step 1 – Check that no geometric limitations invalidate the use of WRC 107;

Step 2 – If WRC 107 is applicable, check to see whether or not the elastic approach as outlined in Section VIII, Division 2, AD-160 is satisfactory;

Step 3 – Compute the sustained, expansion and occasional loads in the vessel shell due to the applied nozzle loads. Consider the local restraint configuration in order to determine whether or not the axial pressure thrust load ($P*A_{in}$) should be added to the sustained (and/or occasional loads). If desired by the user, this thrust load will be automatically calculated and added to the applied load;.

Step 4 – Calculate pressure stresses, P_m, on the vessel shell wall in both longitudinal and circumferential (hoop) directions for both sustained and occasional cases. Notice that two different pressure terms are required in carrying out the pressure stress calculations. P is the design pressure of the system (sustained), while P_{var} is the difference between the peak pressure and the design pressure of the system, which will be used to qualify the vessel membrane stress under the occasional load case. Note that the P_m stresses will be calculated automatically if a pressure value is enter by user.

Step 5 – Run WRC 107 to calculate the P_l, and Q stress as defined earlier. Note that the local stresses due to sustained, expansion and occasional loads can be computed simultaneously.

Step 6 – Various stress components can be obtained from combining the stress intensities computed from applying the sustained, expansion and occasional loads. These stress intensities can then be used to carry out the stress summations and the results are used to determine acceptability of the local stresses in the vessel shell. Notice now CAESAR II can provide the WRC 107 stress summation module in line with the stress calculation routines.

Under the above procedure, the equations used in different programs to qualify the various stress components can be summarized as follows:

$$P_m(SUS) < S_{mh} \tag{4}$$

$$P_m(SUS + OCC) < 1.2S_{mh} \tag{5}$$

$$P_m(SUS) + P_l(SUS) < 1.5S_{mh} \tag{6}$$

$$P_m(SUS + OCC) + P_l(SUS + OCC) < 1.5(1.2)S_{mh} \quad (7)$$

$$P_m(SUS + OCC) + P_l(SUS + OCC) + Q(SUS + EXP + OCC) < 1.5(S_{mc} + S_{mh}) \quad (8)$$

Study case

Due to anticipated piping system changes, it was necessary to evaluate a diameter nozzle-to-cylindrical shell junction to assess the stresses in the nozzle, insert plate, and shell. The loadings considered were due to weight, pressure, and piping thermal expansion.

The vessel cylindrical shell is 78 in I.D. and 0.875 in. thick; the nozzle is 13.620 in. 0.D. and 2.030 in. thick; and the nozzle opening in the shell is not reinforced. The base material of the shell, and nozzle neck is SA 516-70. The vessel was originally designed to the ASME Code Section VIII, Division 1. Since the Division 1 Code does not explicitly treat assessment of stresses at nozzle-to-shell junctions due to external loadings, the integrity of the nozzle was assessed using Division 2 allowable stress criteria.

In the evaluation, it was assumed that a nozzle *pressure thrust load* due to internal pressure acting on the nozzle internal area should be included, and this is incorporated into the commercially available nozzle stress analysis programs. Although there are numerous papers written on the subject of nozzle load evaluation, the pressure thrust load is often overlooked without a good explanation. *Pressure thrust* must be considered when doing a detailed nozzle load evaluation, and stresses due to the pressure thrust cannot just be ignored. Since there are various methods for handling the pressure thrust load, a comparison of stresses due to internal pressure and the pressure thrust load was made using commercially available WRC 107 and WRC 297 programs , and a commercially available FEA program.

WRC 368 includes 2 loading components, the surface stress due to internal pressure and the pressure thrust load. Pressure thrust is the force exerted on the vessel-nozzle junction due to the internal pressure. Figure 2 shows the arrangement of a typical vessel-nozzle junction. applied load between them.

The WRC 107 program calculates the stresses in the shell at eight points on the inside and outside surface at the nozzle junction due to external loads (see table 2) on the nozzle based on Bijlaard's method. The program also calculates the nominal circumferential and longitudinal stresses in a cylindrical shell due to internal pressure and adds these general membrane stresses to the stresses due to external loads. The program has an option which includes the effect of the pressure thrust load on the nozzle by adding it to the imposed external loads. The WRC 297 program also calculates the stresses in a nozzle-to-cylindrical shell junction due to external loads. The WRC 297 method also calculates the stresses in the nozzle neck as well as at eight points on the inside and outside surface around the nozzle. In the program used, the effect of the end pressure thrust load is also accounted for by assuming that the pressure thrust load should be added to the external nozzle loads.

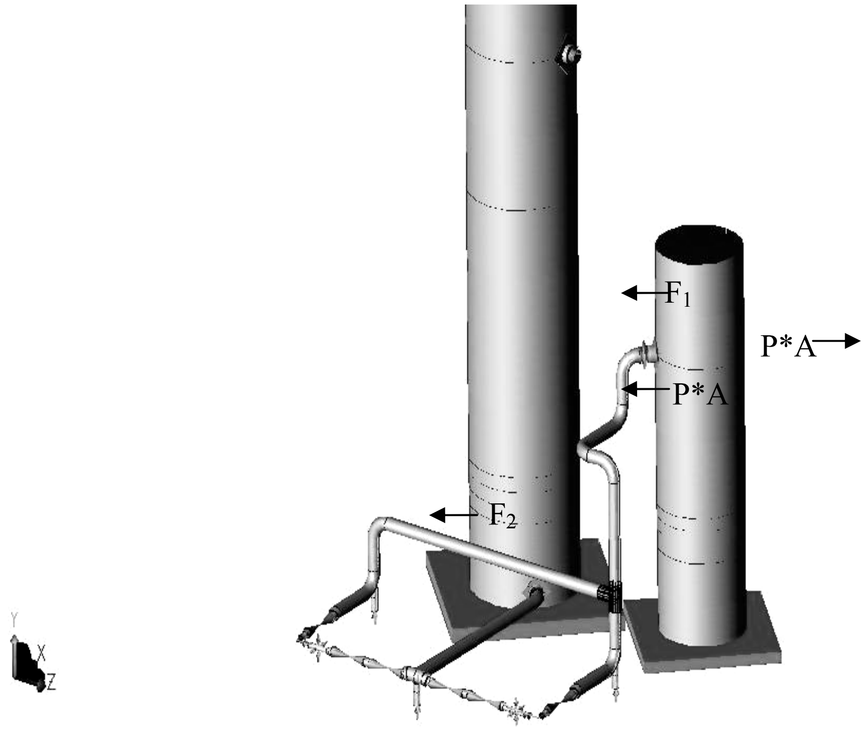

Fig. 2. Pressure thrust effect.

However, the stresses due to pressure thrust at then added to stresses calculated based on multiplying the nominal membrane stress in the shell by fatigue stress indices from the ASME Code Section VIII, Division 2. These intensified stresses are then added to those due to external loads.

A comparison of the results among the methods for the case of internal plus(minus) pressure thrust load is shown in table 3.

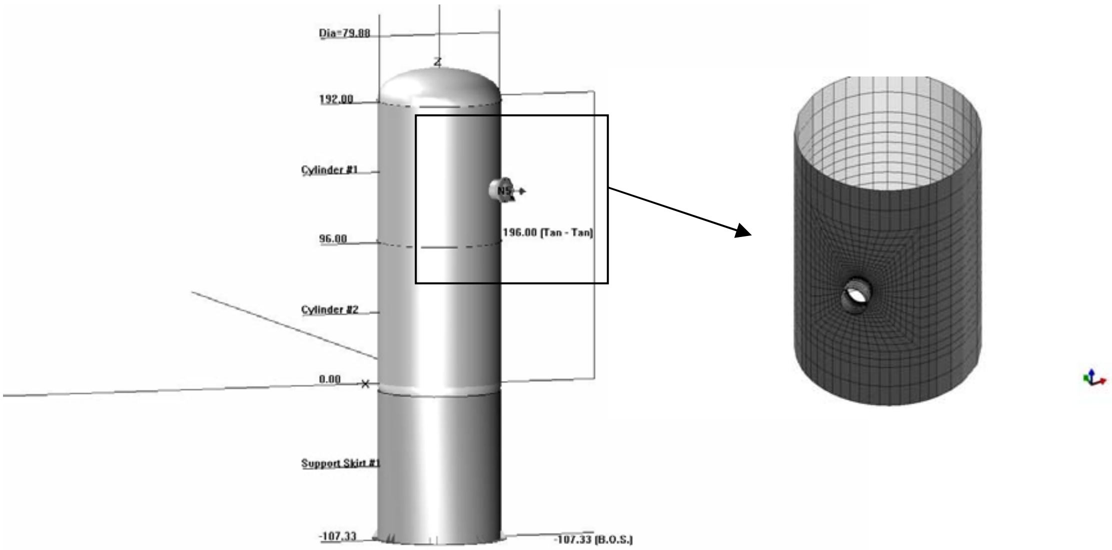

Fig. 3. Pressure vessel FE detail.

Table 1. Forces and Moments indicated at the nozzle to shell junction (after piping analyses, figure 2, using CAESAR II program).

EQUIPMENT: 039-V-012						
NOZZLE NO.: N5 – 10" Nozzle			NODE NO.: 5081			
LOAD CASES	FORCES (lbs)			MOMENTS (ft-lbs)		
	Fx	Fy	Fz	Mx	My	Mz
HYDROTEST	351	734	109	1457.9	-329.3	-1244.5
OPERATING AT DESIGN TEMPERATURE	-1070	-1932	41	-4540.1	4232.0	1199.2
SUSTAINED ONLY	357	769	111	1508.4	-332.5	-1292.4
90-MPH WIND(MAX)	191	-445	-110	690.9	-227.3	386.8
SEISM(MAX)	638	-1626	346	2460.4	-695.6	1452.3
MAXIMUM EXPANSION ONLY	-1635	-3940	-463	-6534.9	5305.0	9515.7

Table 2. Results comparison (stresses, in psi).

Method	Header Next to Nozzle Weld P_1	Branch Next to Header Weld P_1	Header Next to Nozzle Weld P_1+P_b+Q	Branch Next to Header Weld P_1+P_b+Q
WRC 107 *No P.T.*	**1404** (SUS, membrane)	N.A.	**25310** (total)	N.A.
WRC 107 *P.T included.*	**13559** (SUS, membrane)	N.A.	**52437** (total)	N.A.
FEA	**17417** (SUS, membrane)	**13892** (SUS, membrane)	**53297** (inner) **51085** (outside) (total) see figure 4, and ec.8	-

Conclusion

It appears that the WRC 107 (PT included) program may under-predict the membrane stress in the shell, while the total surface stress seems reasonable.

The WRC 107 (PT included) solution also appears to be in agreement with the FEA solution.

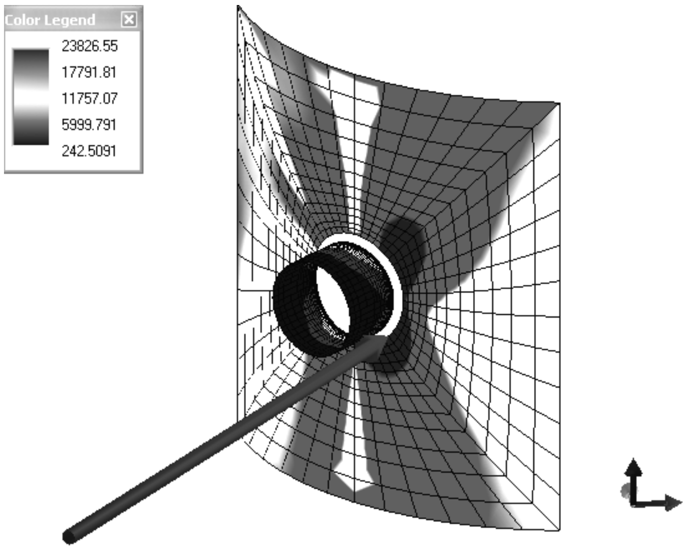

Fig. 4 Stresses distribution on Header Next to Nozzle Weld.

In summary, for the nozzle in this particular case, it was determined that the nozzle and header would not be overstressed considering the combination of pressure, weight, and piping loads due to thermal expansion. *It was determined that the stresses at the nozzle-to-shell junction due to pressure and pressure thrust loads should be considered*.However, not all analytical methods are equivalent since the various methods can yield stresses that vary by a factor of at least three.

References

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Analiza stării de tensiuni în zona racordurilor recipientelor presurizate

Rezumat

În prezenta lucrare se propune, un studiu comparativ între metodele analitice şi cele numerice privind evaluarea stării de tensiuni în zona racordurilor recipientelor presurizate uniform la interior.