Modeling of Internal Pressure and Thrust load on Nozzles using WRC 368

(by: Mandeep Singh and Dave Diehl)

The vessel-nozzle junction presents an unusual situation for stress analysis. Local areas of high stress occur near the junction because of the presence of the hole in shell wall and welds that attach the nozzle to the shell. The loads on the vessel-nozzle junction can be external (such as from the piping system) or can be due to internal pressure.

The Welding Research Council (WRC) Bulletins 107, 297 and 368, provide empirical methods, for calculating stresses at the vessel-nozzle junction. Many have asked how to model the thrust loads on the nozzle. WRC 368 addresses the internal pressure and the thrust loadings on the nozzle. In PVElite Version 4.2 we will implement WRC 368, as it can be a useful design aid. In this article, we examine various aspects of WRC 368 and how it affects the local stress calculations.

WRC 107 and WRC 297 provide the formulae for stresses resulting from external loading. WRC 107 has been discussed in two previous articles in June 1997, June 2000 newsletters. In this article, we will focus on stresses due to internal pressure.

Concepts

WRC 368 includes 2 loading components, the surface stress due to internal pressure and the pressure thrust load. Let's review the pressure thrust load.

Pressure Thrust

Pressure thrust is the force exerted on the vessel-nozzle junction due to the internal pressure. Figure 1 shows the arrangement of a typical vessel-nozzle junction.

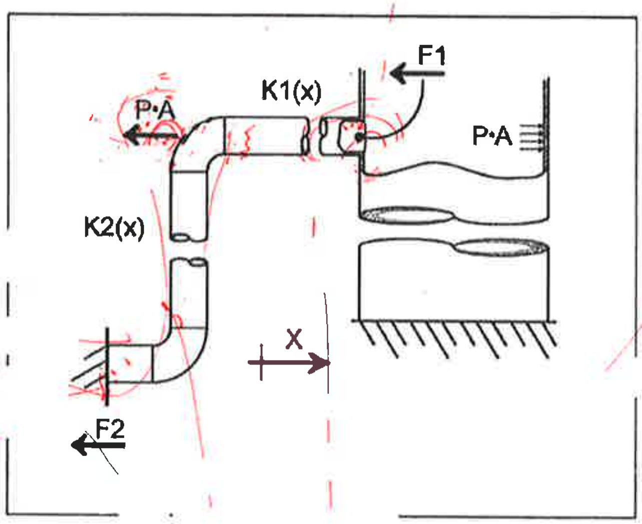
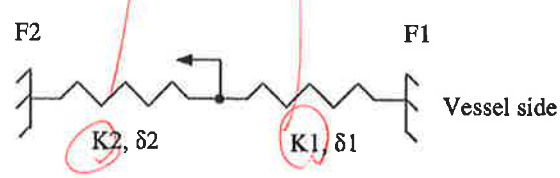


Figure 1

To illustrate further, let's assume the case of a nozzle attached to a vessel on one side and to the piping system on the other side. Let P be the internal pressure of the vessel and piping, and A be the inside area of the nozzle. Then the load of interest is P*A located on the elbow "upstream" from the nozzle, pointing away from the nozzle. The balancing force (P*A) acts on the vessel wall opposite to the nozzle and is shown in Figure 1. This balancing force is countered by the vessel support, which isolates it from the nozzle; hence it is not considered in this load evaluation.

The load on the vessel-nozzle junction will be a function of the stiffness between the vessel anchor and load (including any nozzle flexibilities) (Spring 1), and the stiffness of the system beyond the load (Spring 2). It can be visualized as two springs in series with the applied load between them.



The force F is in equilibrium with the two spring forces F1 and F2:

$$F = F1 + F2 \tag{1}$$

The spring stiffness K and the displacement δ can be related as:

$$K1 = F1 / \delta 1$$
$$K2 = F2 / \delta 2$$

So:

$$F = \delta 1 * K1 + \delta 2 * K2$$

Since, $\delta 1 = \delta 2$, let's denote it by δ:

So:

$$F = \delta * (K1 + K2)$$
$$\delta = F / (K1 + K2)$$

$K2 \to \infty$

Pressure thrust load on the vessel-nozzle junction: $F1 \to 0$

$$F1 = F * K1 / (K1 + K2) \quad (2)$$

If the piping system on the other side of the applied load (Spring 2) is stiff, for example due to an anchor, then pressure thrust will be absorbed by the anchor. Thus, the nozzle will experience very little direct axial stress. This can be seen from equation 2. Note that a greater K2 results in a lower thrust force F1. Therefore, in this case including all of the pressure thrust into analysis will be conservative.

If on the other hand the run of pipe denoted by spring 2 is flexible (maybe due to an expansion loop) then the nozzle will see more of the force due to pressure thrust. Therefore, we should add the appropriate portion of the pressure thrust.

There can be another extreme case; if the nozzle has a blind flange then it will experience the entire force due to the pressure thrust. We must include the whole pressure thrust load for this case.

Hence, the amount of pressure thrust acting on a nozzle depends on the structural response of the system to a pressure load. If appropriate pressure thrust loads are applied to the piping and are analyzed, the structural load at the nozzle due to pressure can be calculated. More research is warranted in this direction, to determine the amount of pressure thrust the vessel-nozzle junction experiences. Note: Except for the pressure effect on expansion joints, the CAESAR II program does not automatically include piping loads due to pressure. Instead, the longitudinal pressure stress is simply added to the piping stresses where applicable as a scalar.

If we cannot accurately determine the amount of pressure thrust, there is a method that analyzes the thrust load more accurately. Here we will review WRC 368 and compare it with other current methods. WRC 368 applies the full load due to pressure thrust (P*A).

Let's look at the various categories of stress caused by internal pressure and pressure thrust load.

Primary Stress
Primary stress is necessary to satisfy the equilibrium conditions with the external imposed loading such as P*A, M/Z. It may also be called load-controlled stress (ASME Code Case N-47-28). Primary stresses are not self-limiting in nature and can cause ductile rupture or a complete loss of load carrying capacity due to the plastic collapse of the structure upon single application of load (ASME). Primary stress can be further sub-categorized as:

- General Primary Membrane Stress (Pm)
  This is the average primary stress across a solid section. It excludes the effect of discontinuities and concentrations. An example is stress in a cylinder due to internal pressure given by Pd/2t.
- Local Primary Membrane Stress (Pl)
  This is the average stress across a solid section. It is caused by external edge resultants developed because of the global discontinuities. Examples include stresses developed at the nozzle hole or at the small end of a conical reducer.

Secondary Stress (Q)
Secondary stress is developed as result of imposed strain. Secondary stress is a global self-limiting stress. Bending stresses and the stresses due to thermal expansion come under this category.

Peak Stresses (F)
Peak stress is a localized self-limiting stress. It causes no objectionable distortion except that it may be a possible source of fatigue failure. Fatigue analysis for the vessel-attachment junction is explained in the June 2000 newsletter.

Nomenclature
Following nomenclature is used in this article:

R	:	Mean Vessel Radius
D	:	Mean Vessel Diameter
T	:	Vessel Thickness
d	:	Nozzle Diameter
t	:	Nozzle Thickness

WRC 368, an Introduction:

WRC-368, entitled "Stresses in Intersecting Cylinders Subjected to Pressure" was released in 1991. WRC 368 provides an approximate method of calculating the maximum stress intensities due to internal pressure at cylinder-nozzle intersections. It is based on the finite element analysis program developed by Prof. C.R. Steele, FAST2. The same program was used in the development of WRC 297.

The method for design of nozzles, subjected to pressure, is given in many pressure vessel codes. A typical method is the area-replacement method. This method assures that the general primary membrane stress near the opening remains below the level of stress before the hole was made. This method does not consider the local primary membrane stresses and bending stresses. The WRC 368 method provides the maximum value of membrane stress intensity (general and local, Pm+PL) and the membrane + bending stress intensity (Pm+PL+Q). Moreover, these stresses are calculated in both the shell and the nozzle. Therefore, WRC368 considers two additional criteria of failure, in addition to the case checked by the area-replacement method.

The FAST2 program, used for creating this Bulletin, applies the full pressure thrust force on the nozzle along with the internal pressure. Therefore, it can be deduced that WRC 368, which is based on FAST2 program, also includes the pressure thrust force on the nozzle. This was further confirmed by one of the authors of WRC 368. It is important because WRC 368 provides much better modeling of the pressure thrust load than the other current methods. Let's compare the analysis methods WRC 107, FEA and WRC 368.

Comparative Study:

Here we will compare the results from analysis performed using the following methods:

1. Pd/2t: This approach uses the general primary membrane stress equation (Pd/2t) for calculation of internal pressure stress. This method is used in the WRC 107/297 module in COADE's programs (CAESAR II, CodeCalc and PVElite), as WRC 107/297 only address external loads. For this approach we did not include the pressure thrust load, see Figure 2.
2. Pd/2t + full Pressure Thrust, Pd/2t + PT(107): This method uses the methodology of WRC 107. In addition to pressure, the whole thrust load (P*A) is applied as a load along the axis of the nozzle. Here we would check the box to include the pressure thrust load.

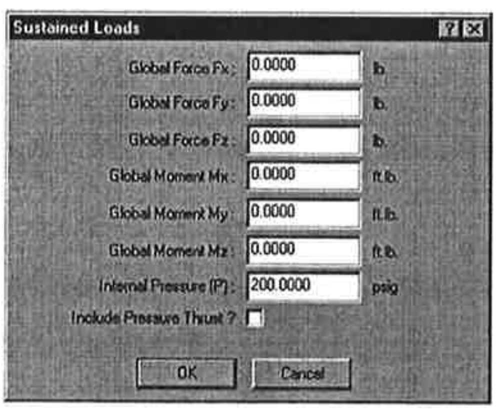


Figure 2

3. WRC 368: Here we used the WRC 368 feature implemented in CodeCalc/PVElite, to activate it click on the appropriate check box as shown in Figure 3. The Loadings include internal pressure and the full pressure thrust load on vessel-nozzle junction.

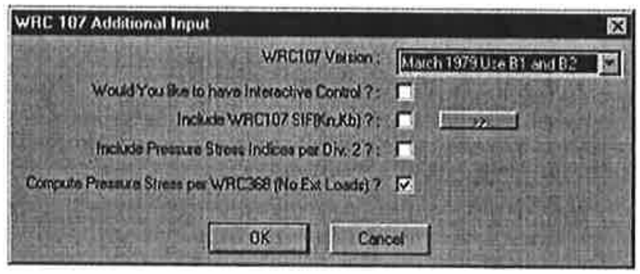


Figure 3

4. FEA: The NozPro finite element program, developed by Paulin Research Group, is used to analyze the models. This program also applies the whole pressure thrust load. Links to this program are conveniently provided in the WRC 107 module in CodeCalc/PVElite.

Internal Pressure only and No Pad:

First, we will do a comparison with internal pressure, no external loads and no reinforcement pad. However, the pressure thrust is an external load, it is considered here because it occurs when the system is pressurized.

Vessel:
Mean diameter: 70 inch
Thickness: 1 inch
Length: 220 inch

Nozzle:
Mean diameter: different runs at 14, 21, 28, 35 and 40 inch
Thickness: .875 inch
Length: 20 inch
Pressure: 200 psi

Let's check if these models are within the geometric limitation of WRC 107/368. The models with nozzle mean diameters of 21 inch to 40 inch exceed the curves used for calculating the bending stress due to radial load on the nozzle (in this case, the pressure thrust). This becomes more pronounced as the nozzle diameter increases. We will see later that this may have an effect on the accuracy of the bending stresses due to the thrust load.

The d/D ratio for the model with the mean nozzle diameter of 40 inches is 0.571, which exceeds the limitation of 0.5 in WRC 107/297/368.

Figure 4 displays the finite element mesh and the contour of the secondary stress, for the model with nozzle mean diameter of 14 inches.

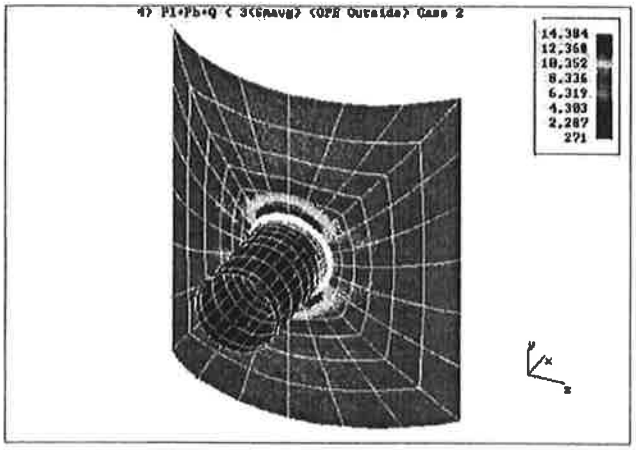


Figure 4

An important parameter in this evaluation is the d/D ratio (nozzle mean diameter/Vessel mean diameter). Therefore, to see its effect we varied the nozzle diameter from 14 to 40 inches, while keeping the rest of the geometry constant. The variation of the primary membrane stresses is shown in the Figure 5. The stresses from WRC 368 and from Pd/2t + PT(107) are close, the stresses from FEA taper off with the increasing d/D ratio.

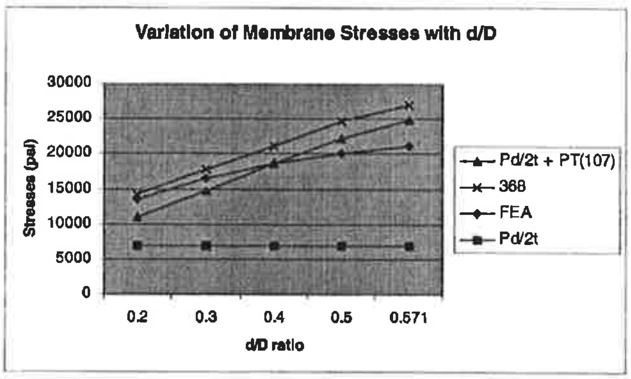


Figure 5

Figure 6 shows the variation of the membrane + bending stresses compared to the d/D ratio. Notice the increase in the stress values from the Pd/2t + PT(107) method with the increasing d/D ratio. If the allowable stress for this case is 60, 000 psi (3*S_{mavg}, for SA-516 70), the design fails miserably per Pd/2t + PT(107) method. However, it still passes when analyzed with FEA and WRC 368 methods!

The reason is simple, as the nozzle diameter increases; the thrust load (P*A) increases by the square of that amount and becomes a significant number. The tests used for preparing WRC 107 did not include internal pressure. Hence, the method Pd/2t + PT(107), does not properly address the pressure issues, especially for the bending stress. Another point to note is that for this method, the curve used for calculating the bending stress due to the thrust load was exceeded. In other words, there was no data available in WRC 107 for this case. Then program used the last value available on the curve, which introduces an inaccuracy. Hence, the increase in stress values from Pd/2t + PT(107) will also be affected by this.

The results from WRC 368 and FEA are relatively close. Indicating that, WRC 368 can be used as a design tool, if performing a finite element analysis is not an option.

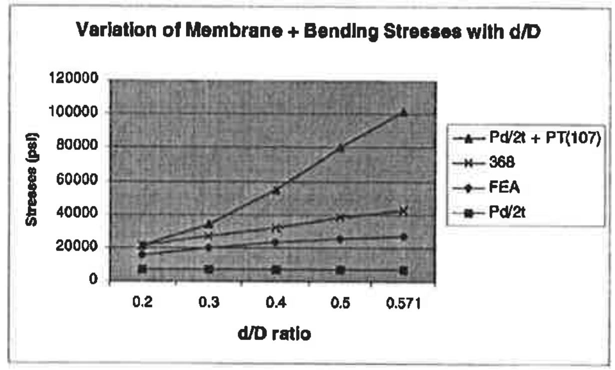


Figure 6

Stresses from the pd/2t method are much smaller than the other methods that additionally include the pressure thrust effect. Pressure thrust load can make a significant effect on stress level around the vessel-nozzle junction. Hence, it useful to check the system and estimate if any pressure thrust load exists.

Due to a more accurate analysis performed by FEA, this design still passes with the full pressure thrust load. We can also see that the accuracy of the WRC methods decrease with an increasing d/D ratio. The points with maximum membrane + bending stress per the FEA, are located in the longitudinal plane (shown in Figure 4), corresponding to the points A and B in the WRC 107 convention. However, WRC 107 reports areas of high stress near points C, D along the circumferential plane. That again suggests that WRC 107 is not appropriate for modeling the pressure loadings.

Reinforcement pad

WRC 107, 297 and 368 do not consider a reinforcement pad. WRC 368 recommends a rule of thumb that has been used successfully and provides somewhat accurate and generally conservative results.

If

Pad width $> 1.65 * \sqrt{RT}$ and $> \frac{d}{2}$

then the shell thickness can be increased by the amount of pad thickness. This ensures that the pad be at least as wide as the region of discontinuity stress around the hole. If the pad does not satisfy these limitations then it should be ignored in the analysis. When the pad is not considered because of this limitation, the results from WRC 368 can be significantly conservative.

Internal pressure and External loads

To get a complete analysis of the vessel-nozzle junction, the stresses from external loads and ones from internal pressure should be combined. We considered using WRC 368 pressure stresses with the 107/297 stresses due to external loads in the section VIII Div 2 stress summation. However, there are some obstacles to this approach. The main reason is that WRC 368 provides the maximum stress intensity, but lacks information about the location and the orientation. On the other hand, the equations given in WRC 107/297 calculate the stresses at different locations around the vessel-nozzle junction and assign proper signs and directions to the stress values.

It is not possible to accurately calculate the stress intensity value due to the combined loads, using WRC 368 along with WRC 107/297. However, WRC 368 recommends that an upper bound on the combined stress can be obtained by adding the absolute value of the maximum stress from external loads to the results from WRC 368. This resulting combined stress can be quite conservative depending upon the stress distribution, as the maximum stress due to external loads and pressure can occur at different locations. Moreover, the stresses from these 2 loading conditions can also act in opposite directions to reduce the combined effect.

Limitations of WRC 368

WRC 368 has geometric limitations similar to those traditionally applied to WRC 107 and 297:

- $10 < D/T < 1000$
- $4 < d/t < 1000$
- $0.1 < t/T < 3$
- $0.3 < Dt/dT < 6$
- $0.3 < d/\sqrt{Dt} < 6.5$
- Nozzle must be isolated (it may not be close to a discontinuity) – not within $2.5\sqrt{RT}$ on vessel and not within $2.5\sqrt{rt}$ on nozzle.
- Results are based on nozzles extending normal to the vessel, on the outside only.

WRC 368 only addresses cylinder-to-cylinder intersections loaded under internal pressure. When these limits are exceeded then the results will not be as accurate.

Conclusions

We have shown that for cylinder-nozzle junctions, under internal pressure only, WRC 368 is a better tool than the pd/2t + PT(107) method, assuming that FEA is most accurate. It provides a much more accurate modeling of the pressure thrust effect when the full thrust load acts on the vessel-nozzle junction, such as in case of a nozzle with a blind flange. Unfortunately, there is no option to control the amount of thrust load. Hence, WRC 368 will be conservative, in cases where only a portion of the thrust load acts on the nozzle. However, because of better accuracy than pd/2t + PT(107), the results may be more reasonable (as seen in the case above).

Utilizing WRC 368 along with WRC 107/297 is not very accurate for calculating the combined stress from pressure and external loads. This is because WRC 368 does not provide information about the location and the orientation of the stresses. However, if the stress analyst has an estimate of the pressure thrust, then a feasible option is to use the pd/2t + PT(107) method and instead of the full thrust load enter the estimated value in the radial load input (with proper signs). The analyst should also note that the results of WRC bulletins will be less accurate if the model exceeds the geometrical limitations or if the curves used for calculating the stresses are exceeded. If the analyst does not have an estimate of the thrust load, he or she can put the whole thrust load and watch-out for very high values of Membrane + Bending stresses. In those cases, WRC 368 can be used to check the pressure stress levels, or advanced analysis tools such as finite element method can be used to obtain accurate combined stress.

Overall, knowing the benefits and limitations of WRC 368, it can be a useful design aid.