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Considerations Pertaining to Exerted Piping Loads on Pressure Vessel Nozzles

Walther Stikvoort
Consultant Pressure Equipment Integrity, Wagnerlaan 37, 9402 SH, Assen, The Netherlands

ABSTRACT: This technical note is intended to make the engineer involved in the design of pressure vessels with connected piping aware of the possible options that yield the most advantageous design from an economic perspective. Pressure vessels are connected to external piping by a nozzle welded to the vessel wall and a flange connection. Due to expansion or contraction of the piping system connected to the pressure vessel, which is caused by the difference in installation and operating temperatures, the piping system exerts forces and moments on the nozzle of the pressure vessel. This means that nozzle loads therefore impose an important constraint in the design of pressure equipment. It is important that such loads do not overstress the vessel-nozzle intersection. This technical note is limited to the interface of the pressure vessel piping system's imposed external loads.

KEYWORDS: Pressure vessels, Piping system, Nozzle loads, Vessel-Nozzle intersection, Imposed external loads

I. INTRODUCTION

Generally, external nozzle loads are provided from one of two sources. The first is from “standard nozzle load tables” [1][9][11]. These are usually characterized by the size of the nozzle rather than its function (e.g. inlet, outlet, vent, drain, etc.). These loads also tend to be round numbers and are often the same magnitude regardless of direction. The magnitude of the standard loadings can have an appreciable effect on the cost of a vessel. The published table values specified can result in additional plate thickness in the shell or head or additional external reinforcement. The second source of external nozzle loads is from a piping system stress analysis. These loads will vary depending on the nozzle size, location and load case.

II. METHODS

Nozzle load tables tend to be conservative since they are generic for each nozzle size regardless of the function. These published values tend to use the largest anticipated loads for any nozzle of a particular size. This imposes an additional problem since the vessel shell and/or heads can vary in plate thickness. It’s sort of a one-size-fits-all scenario where the standard nozzle load table is used for all vessels regardless of the type of pressure vessel, purpose of the nozzle, location of the vessel, and even project. This is where the conservatism comes in. In order to encompass all of the possible permutations, the magnitude of the nozzle loads will be conservative.

External nozzle loads derived from a pipe stress analysis is more accurate than those from standard nozzle load tables. This is due to the fact that each connection is unique and specific to each nozzle. However the magnitude of the nozzle loads from the pipe stress analysis, may be conservative depending upon the boundary conditions defined. It is not uncommon for the stress engineer to completely constrain the ends of the piping model and defining these as anchor points with complete rigidity. While a pressure vessel can be considered relatively stiff, it is not infinitely rigid at the nozzle - vessel junction. Completely constraining the model of the piping system where it connects to the pressure vessel will result in external nozzle loads that may be higher than what actually exists.

The purchaser of the vessel will need to be advised as to the financial impact for incorporating conservative tabulated loads. Companies are not apt to change standard nozzle load tables. Often these tabulated values are mandated without knowledge of how they were derived. Without knowing these details, there is a reluctance to reduce the external loads.

The external nozzle loads at the nozzle can be reduced by defining the stiffness at the nozzle-vessel interface. Sometimes the analyst will estimate the stiffness based on experience using a rough order of magnitude value. Other times the pressure vessel engineer can provide values for the stiffness of the pressure vessel at the nozzles’ locations for the pipe stress engineer to use. Both of these methods will produce more accurate loads at the nozzles.

Most computer programs allow defining imposed displacements and stiffness coefficient at the nozzle-shell intersection. Some programs also provide tools for calculating the stiffness coefficients.

The procedures used are usually derived from the following publications:

(a) PD 5500 Appendix G & Enquiry Case 5500/137 (b) (2021).

(b) WRC 297 "Local Stresses In Cylindrical Shells Due To External Loadings On Nozzles-Supplement to WRC Bulletin No. 107 (Revision I)"(1987).

(c) WRC 107 "Local Stresses In Spherical And Cylindrical Shells Due To External Loadings"(1965).

(d) WRC 537 " Precision Equations and Enhanced Diagrams for Local Stresses in Spherical and Cylindrical Shells Due to External Loadings for Implementation of WRC Bulletin 107 (2020).

(e) Flexibility analysis of the vessel-piping interface by: Martin M Schwartz; International Journal of Pressure Vessels and Piping 81(2004) p.181-189

(f) Pipe Stress Engineering by : Liang-Chuan Peng and Tsen-Loong Peng; ISBN:9780791802854, Publisher: ASME Press. Clause 8.3.2.(2009)

PD 5500 provides procedures for both spherical and cylindrical vessels, whereas the WRC 297 bulletin deals only with nozzles in cylindrical shells. The stiffness of the nozzle heavily influences the stresses in the piping system and also the forces and moments acting on the nozzle itself. Defining the nozzle as a fixed end for the piping can be unnecessarily conservative or can also render non-conservative results for piping stresses.

Reference [12] contains recommendations and supporting data for rules to compute local stresses in nozzles in shells and formed heads due to external loads and pressure.

III. INNOVATIVE APPROACH

Since it would be desirable to realize an empirical and also realistic load set as a function of nozzle outer radius, r, shell outer radius, R, shell thickness, t and allowable design stress, f, a completely innovative method has been developed which is based on reference [2] and has subsequently led to an article in Hydrocarbon Processing [10] in which the method is incorporated in a protocol. Characteristic in this approach are the following expressions [6] as shown in Table 1:

Table 1

Expressions that must be met	
Nozzle on Cylindrical shell element	**Nozzle on spherical shell element**
$\frac{3\,r\,.\,F + 1.5\,M_L + 1.15\sqrt{\frac{r}{t}}.\,M_C}{K} \leq 1.0$	$\frac{r\,.\,F + 2\,M_M}{K} \leq 1.0$
Auxiliary factor	
$K = \frac{\pi\,(r\,.\,t)^2\left(3 - \frac{p_d}{MAWP}\right)f}{\sqrt{R.t}}$	$K = \frac{3.6\,(r\,.t)^2\left(3 - \frac{p_d}{MAWP}\right)f}{\sqrt{R.t}}$

An explanation of the relevant parameters applied in table 1 are:

F	Radial force (N)	R	Mean radius of shell or sphere(mm)
M_L	Longitudinal moment (Nmm)	t	Wall thickness of shell or sphere (mm)
M_C	Circumferential moment (Nmm)	p_d	Internal design pressure (MPa)
M_M	Meridional moment (Nmm)	MAWP	Max. Allowable Working Pressure (MPa)
r	Outside radius of nozzle neck (mm)	f	Design stress (MPa)

Figure 1 shows the forces and moments acting on the nozzle. The shear forces and torsional moments are not considered here because they have minor effect on the stresses around the nozzle - shell junction.

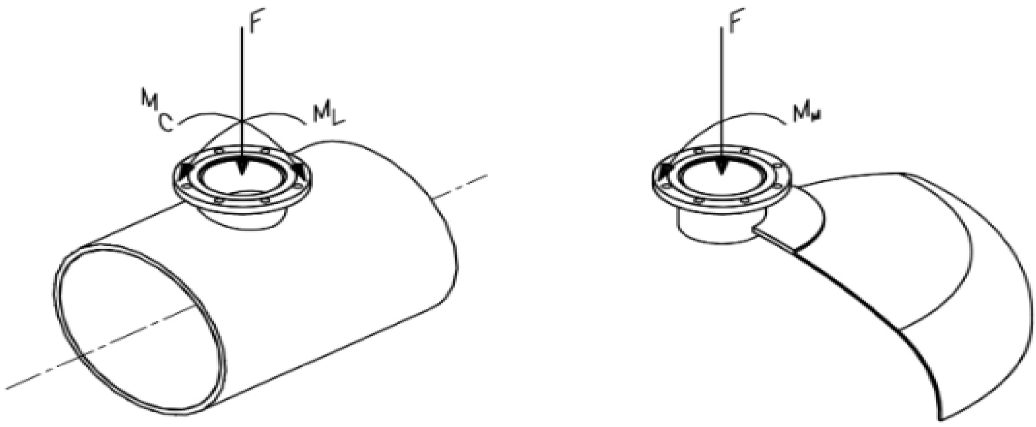

FIGURE 1: NOZZLE ON CYLINDRICAL SHELL AND SPHERICAL SHELL WITH LOADS ACTING ON IT

The internal design pressure is only considered initially when designing the nozzle. The allowable external loads(s) for each nozzle are provided to the pipe stress engineer. It will be the responsibility of the pipe stress engineer to ensure that the piping reactions remain below the limiting values provided by the vessel designer. The vessel design engineer is able to determine the nozzle flexibility. These values can be made available to the pipe stress engineer for incorporation into the pipe stress model. This will yield more realistic results.

IV. DISCUSSION AND CONCLUSIONS

Characteristic of this completely different approach as outlined in [10] is that the pressure vessel designed for internal pressure forms the starting point for calculating the limiting nozzle loads that can be performed by the vessel design engineer. This differs from the traditional approach where a prescribed set of nozzle loads determines the nozzle design. Or in most cases at a later stage, when the piping reactions are known from the formal pipe stress analysis. Experience has shown that a pressure vessel designed exclusively for internal pressure has sufficient piping imposed loading ability. In a practical sense, this means that adjustments to the pipe routing or support arrangement are almost never necessary to stay within the nozzle load limits. It has now been sufficiently demonstrated that this approach is very worthwhile to apply in practice. Finally, I would like to emphasize again that the stiffness at the vessel-nozzle interface has a significant influence on the stresses in the piping as well as forces and moments imposed on the connection to the adjacent pressure equipment.

In order to gain a broader insight into this issue, it is recommended to study the references [3] thru [8] and [10] more closely.

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