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EFFECT OF PIPING LOADS ON VESSEL SUPPORT AND FOUNDATION DESIGN

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ABSTRACT

It is a common challenge for pressure vessel and foundation engineers to determine the effects of piping loads on the foundation and vessel support design and to find out the appropriate design method to be used. Pressure Vessel Codes specify loadings to be considered in the vessel design but limited guidance is provided on the application of piping loads when designing vessel supports. Consideration of piping loads in the design of vessel supports and foundation is left to the engineer's judgment.

Vessel supports are typically designed to withstand the operating weight of the vessel, seismic and wind loading. Pressure vessel literatures provide well-established methodologies in considering these loads in the design of vessel supports. Civil engineering literature, such as the American Society of Civil Engineers (ASCE) Wind Loads for Petrochemical and Other Industrial Facilities [11] or the ASCE Guidelines for Seismic Evaluation and Design of Petrochemical Facilities [17], provide well documented procedures and guidelines for evaluating wind and seismic loads. However, there is limited literature on how to account for external loads from attached piping.

Typical major project schedules have vessels and their supports/foundations designed well before the development of the piping design that provides calculated actual nozzle and piping loads on the vessel. This paper reviews the type of piping loads, how the piping loads are translated to the vessel support/foundation and provides a proposal for simplified approach analysis on how to apply these piping loads in the design of the vessel support/foundation. There might be cases where the piping loads will cancel out, but that may not always be the case. Ignoring and not considering nozzle loads in the support design/foundation may not be appropriate for all vessels.

The intent of this paper is to also make the Vessel Engineer aware of ways to reduce these loads and to encourage communication with Stress Engineers in regards to flexibilities and other factors used to calculate nozzle/piping loads. In most cases, the vessel shell and nozzles are considered as rigid anchors in the piping stress analysis. By using the proper flexibility at nozzle junctions and the global vessel flexibility, the effect of piping loads in the design of the vessel support can be reduced. There is little or no industry guideline on how to include loads due to the thermal expansion or contraction of piping. Since the thermal loads and calculation of flexibility are the least understood part of the analysis, this paper provides background including examples of how these loads and flexibilities can be calculated.

In short, the intent of the paper is to provide a better understanding of how piping loads are translated to the vessel support and to provide some design guidelines that are not readily available in current literature and are not clearly specified in the industry codes or standards.

INTRODUCTION

External loads on vessel nozzles are usually imposed by the piping system connected to the nozzle. ASME piping codes classify these loads as sustained, thermal or occasional loads. Sustained loads are due to pressure and weight of components in the piping system. Thermal loads arise when free thermal expansion of piping is restrained by anchors or equipment nozzles. Occasional loads are imposed on the piping system by events like wind, earthquake, fluid hammer or surge flow.

Piping loads affect the vessel locally and globally. The local effect occurs at the juncture of the vessel wall and nozzle or at the juncture of the vessel wall and pipe support. These loads produce local stresses in the vicinity of the attachment-to-shell juncture. On the other hand, piping loads also have a global effect on the entire vessel. The resultant piping loads eventually could make their way down to the base. The resultant piping loads must be resisted by the vessel support and foundation in order to achieve equilibrium. In an overly simple vessel with one or two nozzles, a simple free body diagram of the vessel will show that nozzle and piping forces are translated to the base of the vessel. The challenge arises when more nozzles each with different types of loads (real or fictitious) and all loads applied at different directions are introduced into the system. Calculating a resultant piping load then becomes difficult. Simulations of the above using AutoPIPE [2] are presented with a simple example in this paper.

In general, piping stress engineers do not consider vessel shell and nozzle flexibilities in their analysis. They would only consider it as a last resort. This very conservative practice of treating the vessel and nozzles as rigid anchors maybe advantageous from the safety point of view, but it usually results in a very uneconomical design of the vessel, foundation and piping system. Applying this conservative approach to all the equipment and piping systems significantly increases the total installed cost of the plant. If the flexibilities are ignored, the loads imposed by the piping system are considered fictitious. In other words, they are not actual or real loads. Ignoring nozzle/piping flexibility in the piping analysis could lead to an overestimation of the piping loads on the nozzle or vessel attachment. WRC 297 [3] mentioned that if nozzle/piping flexibility is not included in the analysis, piping loads may be overestimated by several orders of magnitude. Antaki [4] also reported that using the correct flexibility in static, thermal and dynamic analyses will significantly reduce predicted nozzle loads. Using actual loads in the vessel, piping and foundation analyses will result in an optimized plant design. In addition, it is not just the flexibilities of the vessel, but all the flexibilities of the system including the foundations, individual pipe supports, piping, and vessels that are required to calculate the actual loading.

It has become a common practice to ignore the global effect of nozzle loads on the vessel support and foundation. This seems particularly true in the case of tall/flexible towers or columns. Forces, unless they cancel out, will always make their way down to the support base and foundation. It can be assumed that existing equipment has not failed in operation from piping loads being ignored in the support/foundation design since there have been no reported failures. There could be many reasons why such existing equipment has not failed in operation such as:

1) Conservatism or high factor of safety used in the design of the vessel, foundation and piping system. The designs are usually per component basis, if the whole system was modeled together, the loads would be shared between all components.

2) The load combinations used for design are not actual loading. For instance many plants shut down during a high wind event, therefore, design wind loads are not combined with the thermal loads which are the dominant loads due to piping. The probability of the worst case wind and piping loads acting together is remote.

3) Most calculated piping loads are increased for design typically 10%-20%.

4) The flexibility of the foundations is rarely considered in the piping analysis or the vessel analysis, but experience with dynamic analysis shows this can have a dramatic impact on the magnitude of loads.

5) Additional piping loads were included in the design of the support/foundation. This is typically done very conservative (i.e. effective diameter increase, including some piping in vessel model, and /or adding piping/nozzle loading to the vessel model).

LITERATURE AND CODES

The existing literature and codes do not give clear guidance on how to translate the piping loads to the vessel support and foundation. For example, Bednar [5] identifies that wind load on the pipe shall be taken into account by assuming projected surface of the largest pipe in the top third of the column running to the ground. In addition to the wind loads, Bednar identifies, the piping loads shall be evaluated and they shall consist of the weight of the pipe supported by nozzles and the loads due to thermal expansion of the pipes. Bednar further pinpoints that it can be assumed that the total sum of the piping reactions of all side of the nozzles will have a small effect on the entire vessels and can be disregarded. In addition, Bednar states that the thermal thrust on the top nozzle can be considerable and shall be added to other loadings on the vessel. Bednar recommends that this thrust moment shall be calculated using the following formula $M_p=60D^3$ in lb-in. On the other hand, Megyesy [6] claims that "the vessel is not intended to serve as anchor points for piping and to avoid excessive loading in the vessel, the piping shall be adequately supported". Per Australian Standard AS 1210 [7], "the specified direction and maximum size of local loads for localized nozzle stress calculation might not be credible when combined with other maximum nozzle loads for the design of the total resulting piping loads acting on vessel supports. Engineering studies or a detailed assessment of piping load cases can provide a more credible total resultant piping load. In the absence of such assessment or any specified design loads, the supports should be designed for a minimum of 50% of the most conservative addition of all credible piping loads as applied to the vessel supports.” British and European Standard BS EN 13445-3 [8] declares: "Additional forces from attached piping, other than weight, wind, and earthquake loads shall be considered. It is the responsibility of the designer to decide to what extent additional forces from attached piping have to be taken into account for the static analysis of the columns since their influence depends on the whole behavior of the column and piping configuration". BS EN 13445-3 provides the following guidelines for incorporation of loads from piping: "Forces and moments on nozzles and supports on the column caused by external piping may act as internal and/or external loads. Internal forces are those that cause local loads only and have no influence on the global equilibrium because they are self-compensating". Further the standard identifies: "Horizontal and vertical reaction forces only shall be taken into account, bending moments should be neglected. In the piping analysis the local elasticity of the column wall should be taken into account. The global elasticity of the whole column may be taken into account provided that all essential pipes attached to the column are considered in the piping analysis". ASME Codes [9, 10] identify that piping loads shall be considered for support design but do not provide requirements on how to apply piping loads to the design of the supports.

TYPE OF PIPING/NOZZLE LOADS

Piping loads are the net forces and moments exerted on equipment nozzles, piping supports (and guides) attached to the vessels. The loads exerted on equipment are directly related to how the equipment and piping is supported. The layout and support of the piping will affect the forces and moments exerted on the equipment.

When considering piping loads in the vessel support and foundation design, the best option is to get the actual loads from the pipe stress engineer. However, actual loads may not always be available as the design of vessels and foundations are usually done before stress engineers analyze the piping system. Vessel and foundation engineers must use engineering judgment with reasonable assumptions in estimating the piping loads. With the advent of modern plant design software, engineers can view the 3D layout of the plant (Figure 1). By knowing the piping layout and routing, the engineer could estimate the location and direction of piping loads.

In order to simplify the analysis, loads can be divided into two categories: external and internal loads. External and internal loads act on the vessel when attached piping expands. Figure 2 shows a vertical vessel with attached pipes.

To distinguish the external loads from the internal loads, a control boundary is drawn around the vessel. External loads are acting at the pipe guide Ⓐ, at the side nozzle Ⓑ, and at the bottom nozzle Ⓒ. Thermal forces and moments developed inside the control boundary are internal loads. They are considered to have no global effect on the vessel support and foundation.

Further simplification is to divide loads based on the source such as wind loads, dead loads, seismic loads and thermal loads.

Figure 1. 3D plant layout

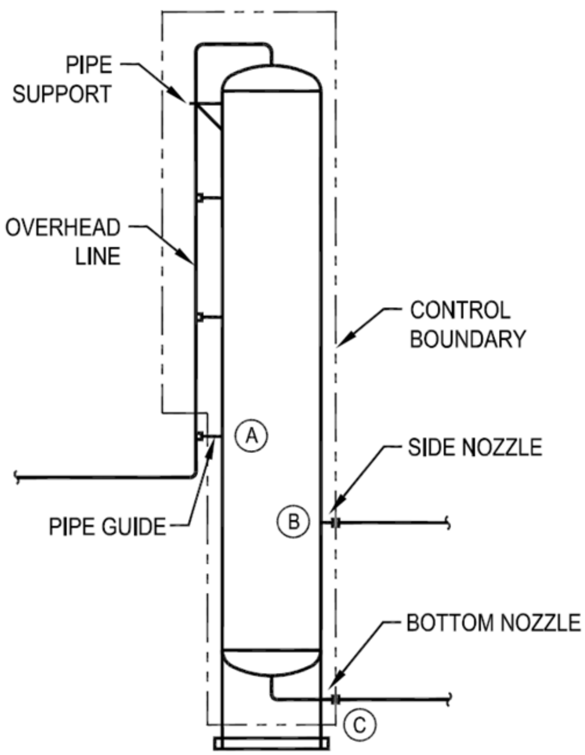

Figure 2. Vertical vessel with attached piping

Wind Loads

For the wind load on piping, vessel engineers usually include the tributary area of attached piping in wind load calculations. ASCE [11] provides guidelines and methodologies on how to account for the wind loads of items attached to vessel, including piping. Common industry practice is to increase effective outside diameter of the vessel by the diameter of the overhead line (including insulation) and use this increased effective diameter when calculating for the projected area exposed to the wind. This approach is usually sufficient to account for attached piping effect on the foundation due to the wind forces for static analysis. Of course, one shall look at the piping layout and make an engineering judgment if some corrections are required. For dynamic analysis of tall slender tower where length to diameter ratio is high (>15), a separate and more complex analysis shall be performed.

Dead Loads

To estimate portion of the piping dead loads that will be transferred to the vessel foundation, it is recommended to use the weight of the piping including insulation and liquid weight. One shall also consider eccentric vertical loads using the distance between the centerline of the piping and vessel. To avoid "double-dipping", vessel engineers should use their judgment and consider either the estimated piping weight or the vertical component of the piping forces in their design. To consider both could result in overdesign or under design, depending on which occasional load (wind or earthquake) governs.

Seismic Loads

Similar to dead loads, it is recommended to use the weight of the piping including insulation and liquid. In addition, one shall consider how the weight is distributed on the different elevations because this will have an impact on the final seismic effect on the foundation. Since this is a well known practice, this paper will not further elaborate the effect of the seismic loads.

Thermal Loads

Among the different classes of piping loads, the least understood is the thermal load. There is little or no industry guideline on how to include loads due to the thermal expansion or contraction of piping. This paper focuses on how to apply piping thermal loads to the vessel support/foundation design.

The externally applied loads considered by WRC [3, 12] in local stress analysis are the radial forces, the shear forces, the bending moments and the torsional moments. Among these external loads, the horizontal component of the radial and shear forces have the most significant effect on the vessel support and foundation. Horizontal piping forces are transferred to the vessel support and foundation as base shear forces and bending moments. External piping moments imposed on nozzles may be considered to act as a force couple as shown in Figure 1. A couple is a pair of forces that are equal in magnitude, but opposite in direction. Its effect is to rotate the nozzle. The moment of a couple is usually small in comparison to the moment of a force about the vessel base support. In other words, the resultant bending moment at the base due to horizontal piping forces is more significant than the piping moment.

Piping loads can also act on other vessel appurtenances besides the nozzle. On tall vertical towers, piping loads also act at pipe supports and pipe guides. In the same case as nozzle loads, the vertical load on a pipe support can be omitted if already accounted for. The horizontal forces on the pipe guides are the loads that need to be considered in vessel support and foundation design.

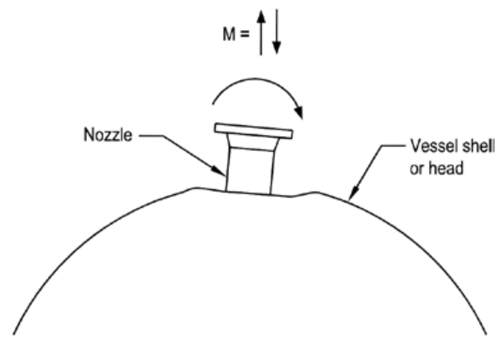

Figure 3. Moment as a force couple

Nozzle Flexibility/Stiffness

"All the world is a spring!" is an expression one might hear from pipe stress engineers. It describes how every element in the world has flexibility/stiffness and reacts like a spring. The elasticity of a spring is expressed by Hooke's law (Figure 4) which states that the displacement of a spring is proportional to the force acting on it. The most common form of Hooke's law is the spring equation:

$$F = k\,x \tag{1}$$

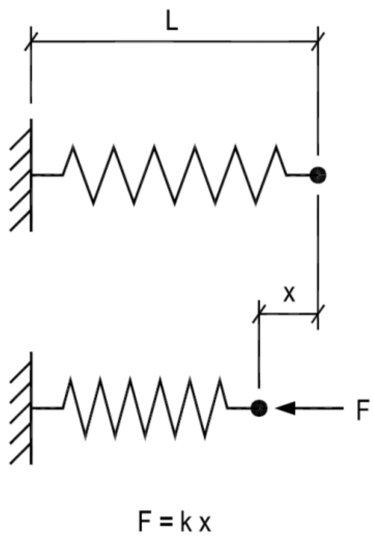

Figure 4. Hooke's Law

Hooke's law may also be expressed in terms of stress and strain. Within the elastic limit, Hooke's law states that stress is proportional to strain. The proportionality constant E is the well-known modulus of elasticity or Young's modulus.

$$\sigma = E\,\varepsilon \tag{2}$$

The stress in Equation 2 can also be expressed as a function of thermal force and pipe cross-sectional area; and the strain as a function of thermal expansion and pipe length:

$$\frac{F}{A} = E\,\frac{\delta}{L} \tag{3}$$

The displacement due to thermal expansion is expressed as:

$$\delta \quad = \alpha\,(T_2 - T_1)\,L \tag{4}$$

Substituting Equation 4 into Equation 3, the force can be expressed as:

$$F \quad = E\,\alpha\,A\,(T_2 - T_1) \tag{5}$$

It has been an industry practice to consider vessel nozzles as rigid anchors in pipe stress calculations. This conservative approach results in unrealistically high nozzle loads. Consider the simple layout in Figure 5, where a straight pipe is anchored to two vessels. Assume the vessel and piping material are carbon steel and calculate the thermal force developed in the pipe and vessel shell. The pipe is subjected to a final temperature of 150 °C from an initial temperature of 20 °C. The cross-sectional area of the pipe is 3601 mm^2. The modulus of elasticity is 202,350 MPa and the coefficient of thermal expansion is 12.4 x 10^{-6} mm/mm/°C.

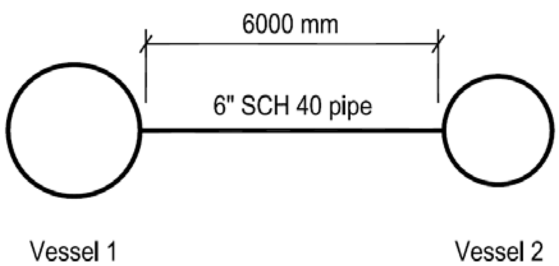

Figure 5. Vessels with stiff piping

Without vessel flexibility, we get the thermal force from Equation 5 as:

$$\begin{aligned} F \quad &= (202{,}350)\,(12.4 \times 10^{-6})\,(3{,}601)\,(150 - 20) \\ &= 1{,}174{,}604 \text{ N} \\ &= 1{,}175 \text{ kN} \end{aligned}$$

Now consider one vessel as rigid and the other vessel as flexible with a spring rate of 55,000 N/mm.

From Equation 4 we get the pipe expansion:

$$\begin{aligned} \delta \quad &= (12.4 \times 10^{-6})\,(150 - 20)\,(6{,}000) \\ &= 9.672 \text{ mm} \end{aligned}$$

Applying the vessel flexibility, the reduced force from Equation 1 is:

$$\begin{aligned} F' \quad &= (55{,}000)\,(9.672) \\ &= 531{,}960 \text{ N} \\ &= 532 \text{ kN} \end{aligned}$$

If both vessels were to be considered flexible, the force will reduce even further. The resulting forces in the preceding example are excessive even with the inclusion of vessel flexibility. In reality, a straight piping layout is rarely used. Expansion loops are usually provided to add flexibility to the piping system.

Similarly, bending moments can also be reduced by considering flexibility at the vessel shell-nozzle junction. Inclusion of rotational spring rates in the analysis will reduce the longitudinal, circumferential and torsional bending moments.

Piping engineers use specialized finite element analysis (FEA) software, such as Autopipe [2], to do piping stress calculations. It allows the engineer to build a 3D model of the piping system, add supports, temperature and pressure information, and see how the piping reacts. To try to reduce the nozzle loads, additional flexibility needs to be added to the piping system including at the nozzle-shell junction.

The flexibility of the nozzle-vessel junction is affected by the geometry of the vessel and the nozzle. Simplification of this geometry is very complicated. FEA programs such as Nozzle/PRO [13], are used to obtain the junction flexibility.

Guided Cantilever Method

One of the simplified methods for piping flexibility analysis is the guided cantilever method. A guided cantilever, as shown in Figure 6, is a beam restrained by a guide at the free end. Applying a force at the free end will subject the beam to bending and deflection, but no rotation.

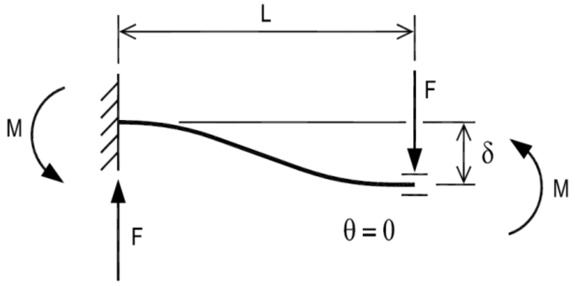

Figure 6. Guided Cantilever

From basic beam theory, the reaction force and moment of a guided cantilever are:

$$F = \frac{12\,E\,I}{L^3}\,\delta \tag{6}$$

$$M = \frac{F\,L}{2} = \frac{6\,E\,I}{L^2}\,\delta \tag{7}$$

Considering a pipe as a beam, the bending stress in the pipe can be expressed as:

$$\sigma = \frac{3\,E\,d_o}{L^2}\,\delta \tag{8}$$

Piping codes require that the computed displacement stress range in piping systems must not exceed the allowable displacement stress range S_A. For ASME B31.3 [14], the allowable stress range is set as:

$$S_A = f\,(1.25\,S_c + 0.25\,S_h) \tag{9}$$

The allowable stresses S_c and S_h are found in Table A-1M of ASME B31.3. For simplicity and to err on the safe side, the stress reduction factor f can be considered as 1.

Setting the bending stress in Equation 8 to be equal to the allowable stress range S_A, an expression for the required length L to accommodate the thermal expansion can be given as:

$$L = \sqrt{\frac{3\,E\,d_o\,\delta}{S_A}} \tag{10}$$

The guided cantilever method is generally conservative because it assumes that the piping system does not rotate. In reality, piping systems have rotational displacements. In the absence of actual loads from piping stress, the guided cantilever method can be used to determine the estimated thermal forces imposed by piping. Kellogg [15] mentioned that in capable hands and with ample allowance for their limitation, the guided cantilever method serve to provide a quick rough check for piping flexibility analysis. The same philosophy can be applied for designing vessel supports and foundations. With reasonable safety margins, the guided cantilever method can provide a rough estimate of piping loads to be used in the vessel support and foundation design.

Sample Calculations of Guided Cantilever Method

Sample Calculation 1:

Overhead Line of vertical vessel in Figure 7

Pipe material: ASTM A106, Gr. B
Design temperature: 250 °C
Cold modulus of elasticity: E = 202,350 MPa
Code allowable stress range: S_A = 274.5 MPa
Pipe size and thickness: NPS 16, STD wall
Pipe moment of inertia: I = 233,957,073 mm^4
Cross-sectional area of pipe: A = 11,876 mm^2
Length of Leg No. 0-1: $L_{0\text{-}1}$ = 18,000 mm
Length of Leg No. 1-2: $L_{1\text{-}2}$ = 30,000 mm
Length of Leg No. 2-3: $L_{2\text{-}3}$ = 8,000 mm
Mean coefficient of thermal expansion:
α = 13.0 x 10^{-6} mm/mm/°C
Height of pipe guide from skirt base: h_1 = 40,000 mm

The thermal expansion of Leg No. 0-1 is:

$\delta_{0\text{-}1}$ = (13.0 x 10^{-6}) (18,000) (250 – 20) = 54 mm

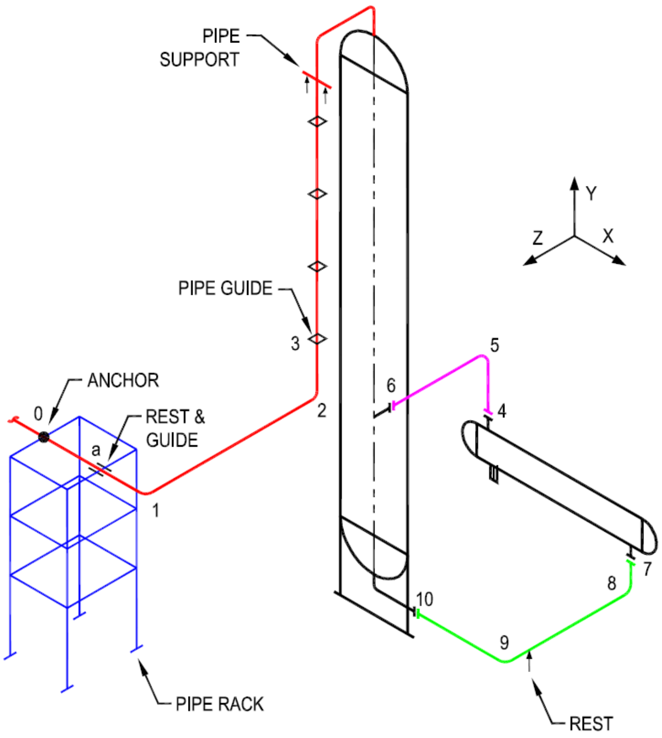

Figure 7. Isometric Layout

If piping flexibility is not considered, the thermal load is excessively high. A simple method of analyzing piping systems with multiple bends is to consider imaginary anchors. By dividing the piping system with real and imaginary anchors, the system can be broken up into simpler subsystems which can be evaluated using the guided cantilever method.

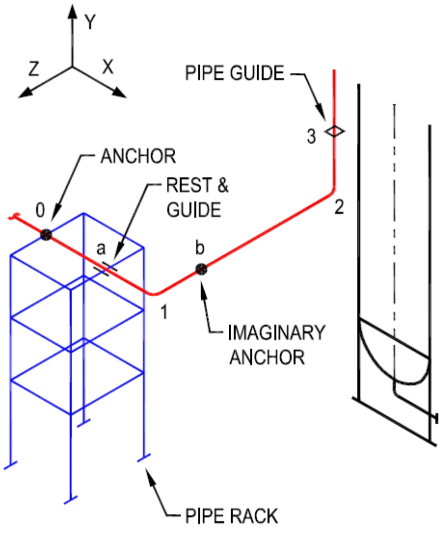

Figure 8. Overhead Line

From Equation 10, the minimum required length to absorb the thermal expansion of Leg No. 0-1 is:

$$L = \sqrt{\frac{3\,(202{,}350)(406.4)(54)}{(274.5)}}$$

$$L = 6{,}967 \text{ mm}$$

To err on the safe side, locate the imaginary anchor at this minimum length and designate as node *b*. From Figure 8, we get the lengths:

$L_{1\text{-}b}$ = 6,967 mm
$L_{b\text{-}2}$ = 23,033 mm

From Equation 6, the force along the x-direction at node 1 can be determined. By statics, this is also the force at the vessel pipe guide (node 3):

$$F_{1(X)} = F_{3(X)} = \frac{12\,(202{,}350)\,(233{,}957{,}073)}{(6{,}967)^3} \quad (54)$$

= 90715 N
= 91 kN

The moment of the force at the base of the skirt is:

$M_{3(Z)} = -F_{3(X)}\, h_1$ = -(90715) (40,000)

= -3,628,600,000 N-mm
= -3,629 kN-m

The force in the z-direction is determined in a similar way. The thermal expansion of Leg No. b-2 is:
$\delta_{b\text{-}2}$ = (13.0 x 10^{-6}) (23,033) (250 – 20) = 69 mm

From Equation 6, we get the force along the z-direction at the vessel pipe guide (node 3):

$$F_{3(Z)} = \frac{-12\,(202{,}350)\,(233{,}957{,}073)}{(8{,}000)^3} \quad (69)$$

= -76,560 N
= -77 kN

The base moment is:

$M_{3(X)} = F_{3(Z)}\, h_1$ = (-76,560) (40,000)

= -3,062,400,000 N-mm
= -3,062 kN-m

(Note: If $L_{2\text{-}3}$ is not known at the start of design, it can be approximated by using Equation 10)

The Appendix attached to this paper provides sample calculations for the side and bottom nozzles of the vertical vessel in Figure 7.

Summation and Resultant of Forces and Moments

Table 1 summarizes the forces and moments in the sample calculations. It also provides the resultant force and moment at the base support.

Table 1. Summary of Forces and Moments

	Forces (kN)		Moments (kN-m)	
	F_X	F_Z	M_X	M_Z
Sample Calculation 1	91	-77	-3062	-3629
Sample Calculation 2	-	115	3438	-
Sample Calculation 3	-141	7	35	704
Summation Σ	-50	45	411	-2925

Resultant Force: $\sqrt{Fx^2 + Fz^2}$ = 67 kN

Resultant Moment: $\sqrt{Mx^2 + Mz^2}$ = 2,954 kN-m

Vessel Flexibility/Stiffness

A vertical vessel is a cantilever beam with one end fixed and the other end free. Loads due to expansion such as thermal loading from piping can be reduced depending on the flexibility of the vessel and the amount of displacement of the thermal load. To model this correctly, a displacement rather than a load would need to be used. A vessel with a higher modulus of elasticity (less flexible) will transfer a higher load to the foundation. To take advantage of the overall vessel flexibility to reduce foundation loads, all displacement loads would need to be taken in account on the vessel. Due to the nature of the vessel structure geometry, location and direction of displacements on the vessel, the determination of foundation loading from piping displacements and sustained loads can be a time consuming and complicated task depending on how many nozzle loads are interacting on the vessel. To ultimately determine the overall foundation loading from nozzle sustained and thermal loading, and take the vessel flexibility into account, all geometric and loading parameters, including the vessel, piping and joining structures would need to be taken into account. An engineering software package would be required to accurately determine the overall effects of the vessel flexibility.

AutoPIPE Model and Analysis

A simulation was performed to demonstrate how the loads at the nozzle junction are decreased once flexibility is introduced into the system.

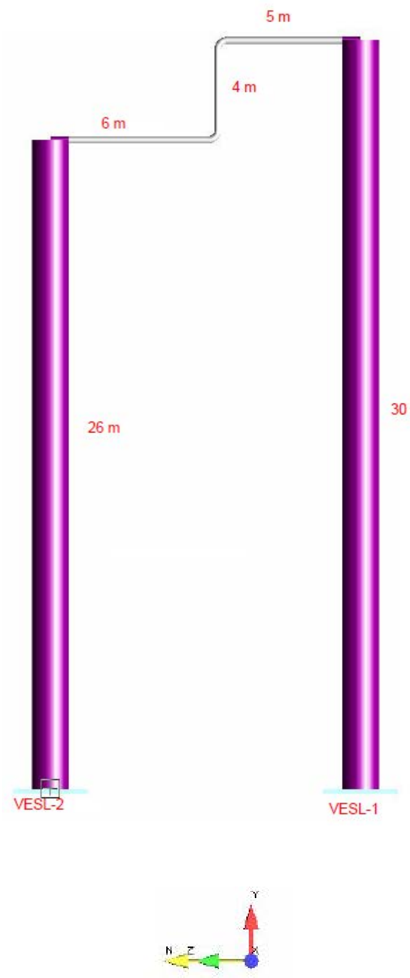

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Figure 9. Simple piping system

The simulation is based on a vessel VESL-1 attached to a very simple piping system (Figure 9). Note VESL-2 is kept the same for both simulations as part of the piping system. The vessel VESL-1 was simulated at a temperature of 500 °C.

For the results, Tables 2 & 3 show that the nozzle loads applied at the shell/nozzle junction do indeed decrease once flexibilities are applied to the nozzle junction and reduce even further once the vessel flexibility is introduced into the model. It also shows that when looking at an extremely simple piping system like the one being simulated, nozzle loads are transferred entirely to the foundation.

Table 2. Thermal loads for case when VESL-1 is rigid

	X		Y		Z		Mx		My		Mz	
	Nozzle Rigid	Nozzle Flex	Nozzle Rigid	Nozzle Flex	Nozzle Rigid	Nozzle Flex	Nozzle Rigid	Nozzle Flex	Nozzle Rigid	Nozzle Flex	Nozzle Rigid	Nozzle Flex
VESL-1	(N)		(N)		(N)		(N.m)		(N.m)		(N.m)	
Nozzle Thermal Load	0	0	-14,894	-11,603	26,977	25,720	22,187	4,609	0	0	0	0
Load Translated to Foundation	0	0	14,984	11,603	-26,977	-25,720	-842,751	-784,898	0	0	0	0

Table 3. Thermal loads for case when VESL-1 is flexible

	X		Y		Z		Mx		My		Mz	
	Nozzle Rigid	Nozzle Flex	Nozzle Rigid	Nozzle Flex	Nozzle Rigid	Nozzle Flex	Nozzle Rigid	Nozzle Flex	Nozzle Rigid	Nozzle Flex	Nozzle Rigid	Nozzle Flex
VESL-1	(N)		(N)		(N)		(N.m)		(N.m)		(N.m)	
Nozzle Thermal Load	0	0	-6,574	6,075	14,455	14,165	3,086	751	0	0	0	0
Load Translated to Foundation	0	0	6,574	6,075	-14,455	-14,165	-441,659	-430,263	0	0	0	0

Foundation Flexibility

Vessels are typically supported with a matt foundation with piles. As stated in the Introduction, no foundation is completely rigid and if the flexibility of the foundation is included in the piping analysis the loads would be reduced significantly. Throughout this paper and for most piping and vessel analysis, the foundation is assumed rigid (fixed). This allows for a conservative simplification in the vessel and piping modeling. As with a modal seismic analysis which includes the interaction of components with their foundations, the loads can be reduced, especially the thermal loads which are displacement controlled (stiffer the system, the higher the loads). A good structural program can be used to model the foundation and derive the flexibility, which then can be applied as a boundary in the piping and vessel models.

SIMPLIFICATION OF THE ANALYSIS

As explained in this paper, to calculate the effect of the piping loads on support and foundation design can be extremely complex and time consuming. Considering that, some codes offer solutions for the simplification of this analysis. In addition, engineers use their judgment to decide which piping loads would have a significant effect on the support and foundation design. Engineering judgment needs to consider the vessel dimensions (such as thickness, diameter and height), nozzle sizes and magnitude of applied loads. Common industry practice is to do parallel engineering and in the initial stages of a plant design the piping loads are not yet defined. However, for thick and rigid (i.e. reactors) vessels with considerable nozzle sizes, an engineer should insist on the piping stress engineers and project to allocate time and effort for performing piping stress analyses for critical pipe lines and obtain actual loads at the time when support and foundation designed. Since such vessels have a small number of nozzles, this effort is well spent and will prevent faulty designs. Below are some examples of possible simplification of the analysis.

Forces and Moments

Only effect of the thermal loads shall be evaluated. The rest of the loads shall be analyzed as explained in previous section of this paper. According to Moss and Basic [16], the effect of tensional loads on the local stress is negligible and the same can be used to the effect of this load on the foundation. The authors of this paper believe this is a fair statement and suggest to be implemented. According to BS EN 13445-3 [8] horizontal and vertical reaction forces shall be taken into account and bending moments at nozzles should be neglected. Authors deem that this approach is reasonable because the effect of nozzle bending moments are much lower than the effect of vertical and especially horizontal forces. However, one needs to look at each particular vessel where this approach may not be applicable. For example, for a vessel with a large horizontal nozzle with high bending moments (especially if the nozzle is on the lower elevation) effect of nozzle bending moments shall be taken into account.

Vessel and nozzle Rigidity

It is reasonable to assume that for rigid vessels, nozzle loads are translated to the foundation without significant reduction due to nozzle and shell flexibility. However, at the initial stages of a project, nozzle orientation may not be known. Applying nozzle loads in the same direction is conservative but not reasonable. As stated above, authors believe that for rigid vessels with a few nozzles a quick analysis of the piping model is appropriate to determine if the piping loads will add up or cancel each other out. Australian AS 1210 [7] approach states that in the absence of detail information about nozzle loads or any specified design loads, the support should be designed for a minimum of 50% of the most conservative addition of the all credible piping loads as applied to the vessel supports.

It is not clear to the authors the background of this recommendation, but deem that this approach may be suitable for short rigid vertical vessels and horizontal vessels. For the rest of the vessels this method is too conservative (especially for tall towers). Considering complexity of the analysis, application of the percentage to the total loads may be a good practical solution. However, to get the right percentage, a detail study utilizing sample vessels from past projects with different vessel dimensions in combination with actual loads shall be done to determine different percentages for different types of vessels.

CONCLUSION

Many classes of piping loads as described in this paper are applied to a vessel. This paper in particular focused on thermal loads since they are the least understood and there is little or no industry guideline on how to include these on the vessel support design.

In order to determine whether these loads have a global effect on the design of the foundation, flexibility of the vessel as well as nature and direction of the loads needs to be taken into account. It was shown that when flexibility is introduced into the system, nozzle/piping loads are indeed reduced.

Since at an initial stage of a project the direction and nature of the loads are usually unknown and it is not practical to apply flexibilities prior to the design of the vessel support, this paper provides a recommended simplified approach analysis to account for piping/nozzle loads into the vessel support/foundation design. Just ignoring nozzle/piping loads is not appropriate. The recommended approach might even be too conservative for a variety of the vessels (especially tall towers). A study using the vessel geometry and mass should be performed for different vessels. Authors are currently researching using a structural program RISA [1] which models beam elements to fully model the piping and the vessel as a better alternative to determine support/foundation loads for tall flexible towers. This study shall take into account local and global vessel flexibility. However, this study is not a part of this paper and may be a topic for a future paper.

NOMENCLATURE

A = cross-sectional area of pipe, mm^2
c = distance from neutral axis to outermost fiber, mm
d_i = inside diameter of pipe, mm
d_o = outside diameter of pipe, mm
E = cold modulus of elasticity, MPa
F = force, N
f = stress range factor
h = height from base
I = moment of inertia, mm^4
k = axial displacement spring rate, N/mm
k_θ = rotational spring rate, mm-N/deg
M = external overturning moment, N-m
L = relaxed length of spring or pipe leg, mm
S_A = allowable displacement stress range, MPa
S_c = allowable stress in the cold condition, MPa
S_h = allowable stress in the hot condition, MPa
T_1 = initial temperature, °C
T_2 = final temperature, °C
x = displacement of spring, mm
α = mean coefficient of thermal expansion, mm/mm/°C
ε = strain, MPa
δ = deflection due to thermal expansion, mm
θ = rotation, degrees
σ = stress, MPa

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APPENDIX

Sample Calculation 2:

Side Nozzle of vertical vessel in Figure 7 & 10

Material: ASTM A106, Gr. B
Design temperature: 250 °C
Cold modulus of elasticity: E = 202,350 MPa
Code allowable stress range: S_A = 274.5 MPa
Pipe size and thickness: NPS 24, STD wall
Pipe moment of inertia: I = 808,445,744 mm^4
Cross-sectional area of pipe: A = 17,956 mm^2
Length of Leg No. 4-5: $L_{4\text{-}5}$ = 10,000 mm
Length of Leg No. 5-6: $L_{5\text{-}6}$ = 20,000 mm
Mean coefficient of thermal expansion:
α = 13.0 x 10^{-6} mm/mm/°C
Axial translational stiffness at nozzle/shell junction:
k_B = 70,000 N/mm
Height of nozzle from skirt base: h_2 = 30,000 mm

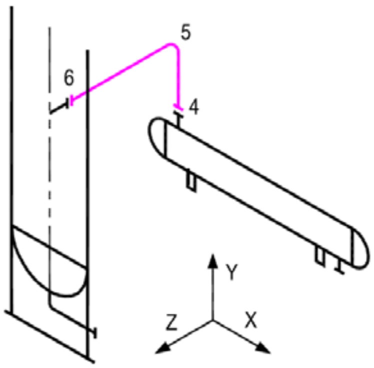

Figure 10. Side nozzle

The direction of thermal expansion of Leg No. 4-5 is opposite that of gravity so the force due to the pipe expansion of this vertical leg will not be considered in the vessel support design.

The thermal expansion of Leg No. 5-6 is:

$\delta_{5\text{-}6}$ = (13.0 x 10^{-6}) (20,000) (250 – 20) = 60 mm

From Equation 10, the minimum required length to absorb the thermal expansion of Leg No. 5-6 is:

$$L = \sqrt{\frac{3\,(202{,}350)(609.6)(60)}{(274.5)}}$$

= 8,994 mm

$L_{4\text{-}5}$ is greater than the minimum so Leg No. 4-5 is considered flexible. Without considering the flexibility at the nozzle/shell junction, the axial force on the side nozzle of the vertical vessel is:

$$\frac{12\,(202{,}350)\,(808{,}445{,}744)}{(10{,}000)^3}\,(60) = 117{,}784\ \mathrm{N}$$

The force can be reduced by considering the flexibility at the juncture of the shell and nozzle. The flexibility at the nozzle/shell junction and the flexibility of pipe Leg No. 4-5 can be considered as springs in series as shown in Figure 11. The general formula for the equivalent spring rate of multiple springs in series is:

$$\frac{1}{k_{eq}} = \frac{1}{k_1} + \frac{1}{k_2} + \cdots \frac{1}{k_n} \tag{11}$$

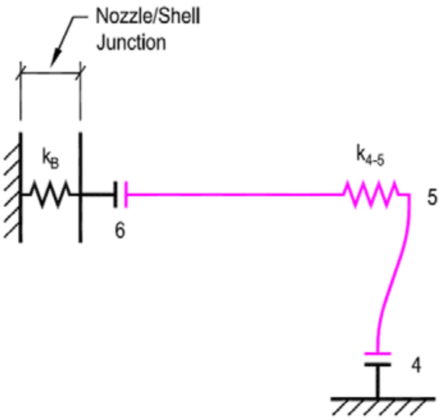

Figure 11. Springs in series

From Equations 1 and 6, the flexibility of a guided cantilever is derived as:

$$k = \frac{12\ \mathrm{E\ I}}{L^3} \tag{12}$$

The flexibility of Leg No. 4-5 is:

$$k_{4-5} = \frac{12\,(202{,}350)(808{,}445{,}744)}{(10{,}000)^3} = 1{,}963\ \mathrm{N/mm}$$

From Equation 11, the equivalent flexibility of the nozzle and piping system is:

$$\frac{1}{k_{eq}} = \frac{1}{70{,}000} + \frac{1}{1963}$$

$$k_{eq} = \frac{(70{,}000)(1{,}963)}{70{,}000 + 1{,}963} = 1{,}910 \text{ N/mm}$$

The thermal force at the side nozzle is:

$$F_{6(Z)} = k_{eq}\ \delta_{5-6} = (1{,}910)(60)$$
= 114,600 N
= 115 kN

The base moment is:

$M_{6(X)} = F_{6(Z)}\, h_2$ = (114,600) (30,000)
= 3,438,000,000 N-mm
= 3,438 kN-m

Sample Calculation 3:

Bottom nozzle of vertical vessel in Figure 7 & 12

Material: ASTM A106, Gr. B
Design temperature: 250 °C
Cold modulus of elasticity: E = 202,350 MPa
Code allowable stress range: S_A = 274.5 MPa
Pipe size and thickness: NPS 20, STD wall
Pipe moment of inertia: I = 463,461,390 mm^4
Cross-sectional area of pipe: A = 14,916 mm^2
Length of Leg No. 8-9: $L_{8\text{-}9}$= 22,000 mm
Length of Leg No. 9-10: $L_{9\text{-}10}$ = 15,000 mm
Mean coefficient of thermal expansion:
α = 13.0 x 10^{-6} mm/mm/°C
Height of nozzle from skirt base: h_3 = 5,000 mm

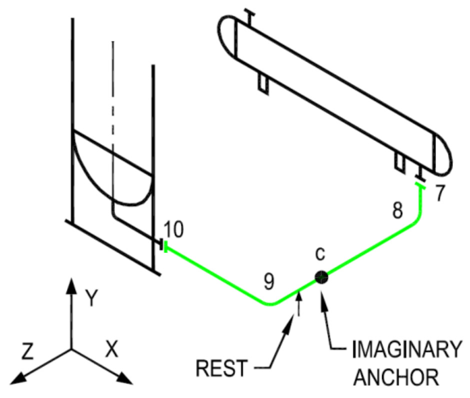

Figure 12. Bottom nozzle

The flexibility of the nozzle/head junction can be considered in the analysis. However, for this sample calculation, the bottom nozzle and head are assumed to be rigid for simplicity in analyzing the piping load.

The thermal expansion of Leg No. 9-10 is:

$\delta_{9\text{-}10}$ = (13.0 x 10^{-6}) (15,000) (250 – 20) = 45 mm

The minimum required length to absorb the thermal expansion is:

$$L = \sqrt{\frac{3\,(202{,}350)(508)(45)}{(274.5)}}$$

$$L = 7{,}110 \text{ mm}$$

Locate the imaginary anchor at the minimum length and designate as node *c*. From Figure 12, we get the lengths:

$L_{8\text{-}c}$ = 14,890 mm
$L_{c\text{-}9}$ = 7,110 mm

From Equation 6, the force along the x-direction at node 10 is:

$$F_{10(X)} = \frac{-12\,(202{,}350)\,(463{,}461{,}390)}{(7{,}110)^3}\,(45)$$

= -140,897 N
= -141 kN

The moment of the force at the base of the skirt is:

$M_{10(Z)} = F_{10(X)}\, h_3$ = (140,897) (5,000)

= 704,485,000 N-mm
= 704 kN-m

The thermal expansion from node *c* to node 9 is:

$\delta_{c\text{-}9}$ = (13.0 x 10^{-6}) (7,110) (250 – 20) = 21 mm

From Equation 6, the force along the z-direction at node 10 is:

$$F_{10(Z)} = \frac{12\,(202{,}350)\,(463{,}461{,}390)}{(15{,}000)^3}\,(21)$$

= 7,002 N
= 7 kN

The moment of the force at the base of the skirt is:

$M_{10(X)} = F_{10(Z)}\, h_3$ = (7,002) (5,000)

= 35,010,000 N-mm
= 35 kN-m