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INVESTIGATION OF MODIFIED NB-3600 BRANCH PIPING STRESS ANALYSIS

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ABSTRACT

The current rules in the ASME Code Section III, NB-3600, for the design of Class 1 piping branch connections provide stress indices for pressure and bending moment loading to calculate interaction stresses at the connection with contributions from the run pipe and branch pipe loads. The stresses at the connection are calculated by applying factors to the far field pressure and bending stresses calculated for the run pipe and for the branch pipe. When a branch nozzle is provided at the intersection with a wall thickness exceeding that of the branch pipe, the current Code rules do not allow any credit for the locally thicker wall in calculating stresses due to branch pipe bending, resulting in a very conservative assessment.

This technical paper aims to demonstrate that an adequately conservative stress analysis is achieved by a modified approach where the branch pipe bending stresses that are combined with run pipe stresses in the piping equations are calculated based on the nozzle thickness at the juncture, as opposed to the wall thickness of the branch pipe remote from the juncture. A comparison is performed of a 3D finite element analysis of a typical branch connection and an appropriately modified calculation of the Code branch pipe stress.

INTRODUCTION

Class 1 standard piping branch connections are designed according to ASME Code Section III, NB-3643.2 [2], and stress indices for bending moment (NB-3650) analysis are calculated according to NB-3683.8. Two possible designs that are depicted in NB-3600 are shown in Figure 1 In both cases, the rules of NB-3683.1(d) for the evaluation of branch connections and tees require that the bending stress at the juncture from loading on the attached branch pipe is calculated based on the nominal diameter and wall thickness of the "attached branch pipe". This stress gets combined (absolute sum) with the stress in the run pipe from which the branch pipe emerges.

Since the location where branch pipe and run pipe stresses interact is at the piping juncture, it seems appropriate to calculate the combined stresses based on branch pipe dimensions for the design shown on the right in Figure 1 (NB-3643.3(a)-1 (d)). When the branch pipe is attached with a reinforced nozzle as shown on the left in Figure 1 (NB-3643.3(a)-1 (a)), it seems highly conservative to calculate the branch pipe stress based on the dimensions of the, possibly much thinner, attached piping. The branch pipe bending stress calculated in this way is typically much higher than the stress calculated based on the dimensions of the nozzle body that is actually attached to the run pipe. Note also that this means that no matter how thick and how long the nozzle body is, the bending stress is always calculated in NB-3683.8 based on the size of the branch pipe.

The conservatism in calculating the branch stress in the Code for connections with a nozzle was pointed out in two papers by Faraz and Gurdal from the 2015 [3] and 2016 PVP Conferences [4]. These papers compare the Code method of calculating stress and Finite Element analyses and show the conservatism of the stresses calculated in the Code when a reinforced nozzle design is used.

Beside the conservatism of the stress analysis method for branch nozzle piping connections that is presently in NB-3600, the method also misrepresents trends that would be expected from design changes, i.e. that the stress would be reduced if a larger nozzle is used. While it is acceptable for design formulas to be simplified and conservative, it is highly desirable that design changes that alleviate an overstressed situation are appropriately reflected in simplified formulas. The present paper explores whether a simple "fix" of the Code rule as proposed by Faraz and Gurdal [3] is confirmed by the present example, and whether limitations need to be imposed on an amended Code rule for piping branch nozzle connections.

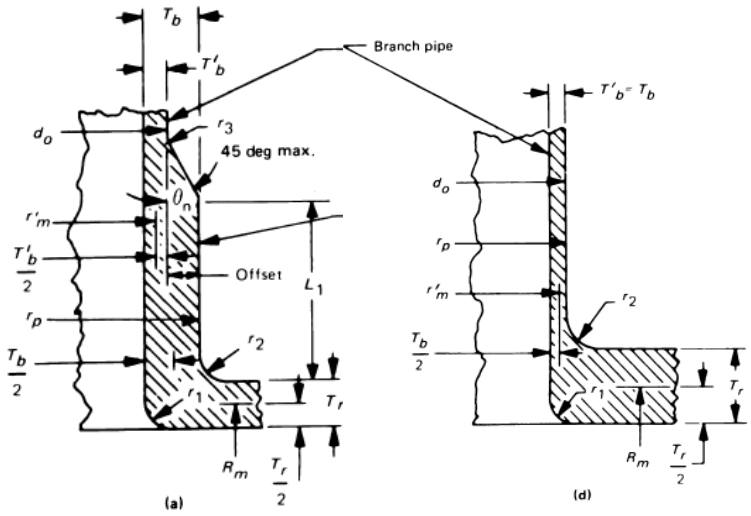

Figure 1: Some Pipe Branch Connection Designs According to Figure NB-3643.3(a)-1

PROPOSED MODIFICATION OF THE CODE RULES FOR BRANCH CONNECTIONS

It was noted above that it seems unrealistic to calculate stresses and stress indices at the juncture between run pipe and reinforced nozzle region based on remotely attached branch piping that in reality should have no effect on the stress concentration at the juncture. Therefore, the proposal that was put forward in Reference [3] and [4] is explored here. Following these references, both the nominal stress and the stress indices are calculated based on the dimensions of the reinforced nozzle.

The run pipe section modulus is unchanged from the Code [2], while the branch section modulus at the juncture is now defined as

$$Z_b = \pi (0.5\, d_m)^2 T_b \tag{1}$$

where

d_m = nominal mean diameter of reinforced or unreinforced branch , in (mm),
T_b = wall thickness of branch connection reinforced nozzle, in (mm).

The definitions of the stress indices for pressure loading remain unchanged from the Code [2] as well. The stress indices for run pipe moment loading in [2] are modified as

$$B_{2r} = \max\left[1.0, 0.75\left(\frac{d_m}{2\,T_b}\right)^{0.3}\right], \tag{2}$$

$$C_{2r} = \max\left[1.25, \left(\frac{d_m}{2\,T_b}\right)^{0.3}, 1.875\left(1 - \min\left(0.5, 0.5\left(\frac{T_b}{T_r}\right)\left(\frac{T_b}{d_i}\right)^{0.5}\right)\right)\right] \tag{3}$$

$$K_{2r} = 1.6. \tag{4}$$

T_r = nominal wall thickness of run pipe, in (mm)

For branch pipe moment loading, only the expression for the Primary plus Secondary stress index needs to be modified as follows:

$$C_{2b} = \max\left[1.5, 1.5\left(\frac{R_m}{T_r}\right)^{2/3}\left(\frac{d_m}{2\,R_m}\right)^{1/2}\left(\frac{T_b}{T_r}\right)\left(\frac{d_m}{2\,r_p}\right)\right]. \tag{5}$$

d_i = internal diameter of reinforced or unreinforced branch, in (mm)
r_p = outside radius of reinforced nozzle or branch connection, in (mm)

These stress indices will be used in the comparisons between modified Code rules and finite element results below. It is assumed that the evaluation of the branch pipe near the nozzle, as well as of the thickness transition from nozzle to branch pipe, is performed separately and independently from that of the juncture.

ANALYZED GEOMETRY

The characteristic non-dimensional geometric parameters characterizing the branch to run pipe intersection are given in Table 1. An additional parameter, the length of the reinforced nozzle, will be defined later. The transition from the reinforced nozzle to the branch pipe is by a 1:3 taper. Note that, for functional reasons of the attached instrument line, the branch opening in the inside of the run pipe is specified as sharp, i.e. has no measurable radius.

Table 1 Geometry of Pipe Intersection

Dimensionless Parameter	Value	Meaning
R_m / T_r	5.333	Ratio of wall thickness to mean Diameter of Run
d_m / D_m	0.120	Ratio of Branch Reinforcement to Run mean diameters
T_b / T_r	0.388	Ratio of Branch Reinforcement to Run wall thickness
r_2 / T_b	0.858	Ratio of Branch fillet radius to Reinforcement wall thickness
T'_b / T_b	0.371	Ratio of Branch pipe to Reinforcement wall thickness

MATERIAL PROPERTIES

It is assumed that the piping material is a typical carbon steel; with S_m = 17500 psi. This value is used only to provide a rough target for the piping stresses. A typical limit for primary membrane plus bending stress would be 1.5 S_m = 26300 psi, while for primary plus secondary stresses it would be 3 S_m = 52500 psi.

FINITE ELEMENT MODEL

The finite element program used for the analysis is ANSYS [1]. The finite element model for the connection is shown in Figure 3. The mesh consists of 20-noded hexagons everywhere. A mesh convergence study for the model has been conducted to determine the level of refinement of the mesh for accurately capturing the reported stresses.

To determine the stresses, one end (downstream) of the run pipe is fixed and the bending moment load for the run is applied at the other end. The branch bending load is applied at the end of the branch pipe. Pressure is applied to the internal surfaces and the corresponding pressure blow-off axial stresses are applied at the free ends of branch and run pipe.

Pressure loading, run bending loading and branch bending loading were analyzed as separate load cases and then combined.

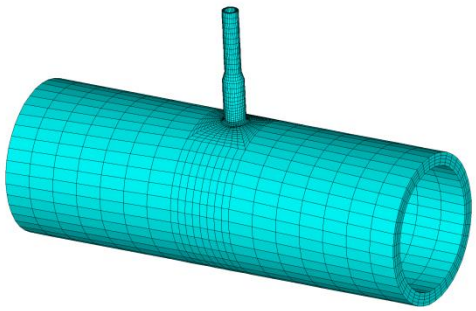

Figure 2 Finite Element Model of Branch Pipe Intersection

Linearized stresses at the piping intersection were extracted at stress classification lines, and the membrane and membrane plus bending stresses were obtained. In total, 33 classification lines in the run pipe wall and 33 classification lines through the branch pipe wall over half of the circumference of the intersection element were evaluated. Figure 3 shows the locations of the lines, which were evenly distributed at an angular spacing of 5.625º. The lines were chosen with consideration of the principles formulated by Hechmer and Hollinger [5].

Total Stresses were extracted directly as the highest stresses from the finite element model.

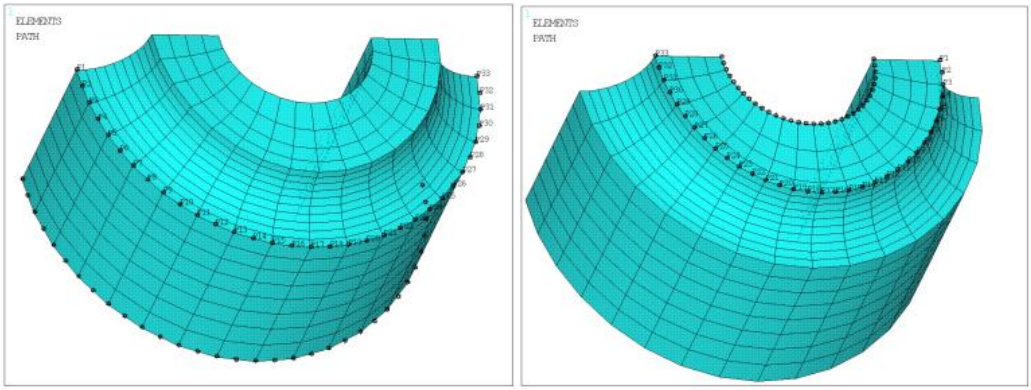

Figure 3 Stress Classification Lines at Intersection, Left: Run Location, Right: Branch Location

BASE ANALYSIS CASE

The dimensionless length of the reinforced nozzle, L_1, for the base case is $L_1 / \sqrt{T_b\, d_m/2}$ = 6.69. This does not include the 1:3 tapered transition to the branch pipe. The applied loading consists of internal pressure, p, run bending specified in terms of the remote run bending stress, σ_{br}, and branch bending specified in terms of the remote branch bending stress, σ_{br}, as follows:

p = 1600 psi, σ_{br} = 12,000 psi, σ_{bb} = 12,000 psi

The loads are applied individually and then in a combined load case:

Load Case 1: Pressure Load Case 2: Run Bending

Load Case 3:Branch Bending Load Case 4: Combined Loading

It is instructive to plot the linearized (Membrane and Membrane Plus Bending) stresses along the circumference of the half intersection volume shown in Figure 3 (left). The plots for the different load cases are shown in Figure 4. The maximum linearized stresses due to internal pressure (upper left plot in Figure 4) occur on the plane of symmetry. This is expected from the notion of stress concentration around a hole where the largest stress component is the hoop stress.

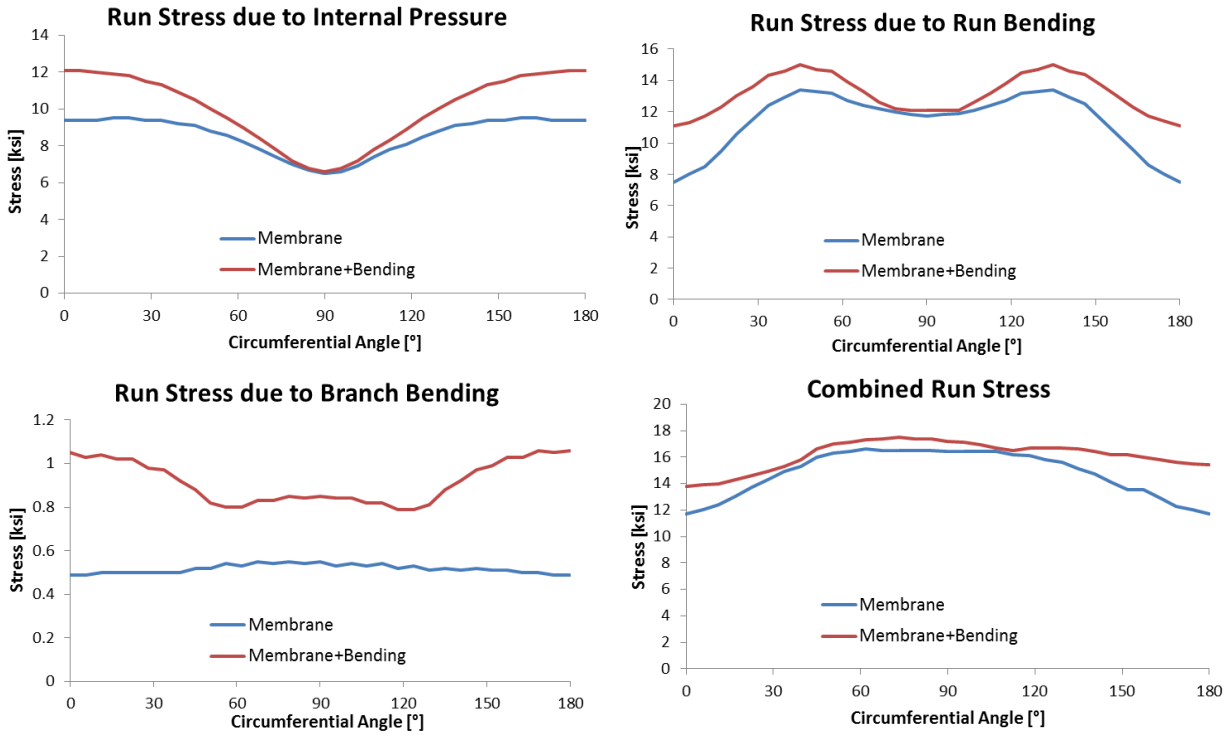

Figure 4 Membrane and Membrane Plus Bending Stress at the 33 Classification Lines in the Run Pipe (Figure 3) for the 4 Load Cases

As seen from the upper right plot in Figure 4, the highest Membrane and Membrane Plus Bending stresses under run pipe bending in the plane of the juncture occurs neither in the symmetry plane nor at 90° around the nozzle from the symmetry plane. Instead, the maximum for the present geometry is at about 50° from the symmetry plane. This can be rationalized by considering the axial stress flow due to bending going around the inside diameter of the branch ID, and thus well inside of the classification line circle.

Since branch pipe bending is also occurring in the plane of symmetry, it is not a surprise that the stresses in the wall of the run pipe have their maximum on the plane of symmetry as shown by the lower left plot in Figure 4. The magnitude of these stresses is low.

The maximum linearized stresses for combined loading (load case 4) show an unsymmetric pattern. Since the maximum stresses of the individual loads are due to different stress components, and occur at different locations around the nozzle, the combined stress is much lower than the sum of the individual maxima.

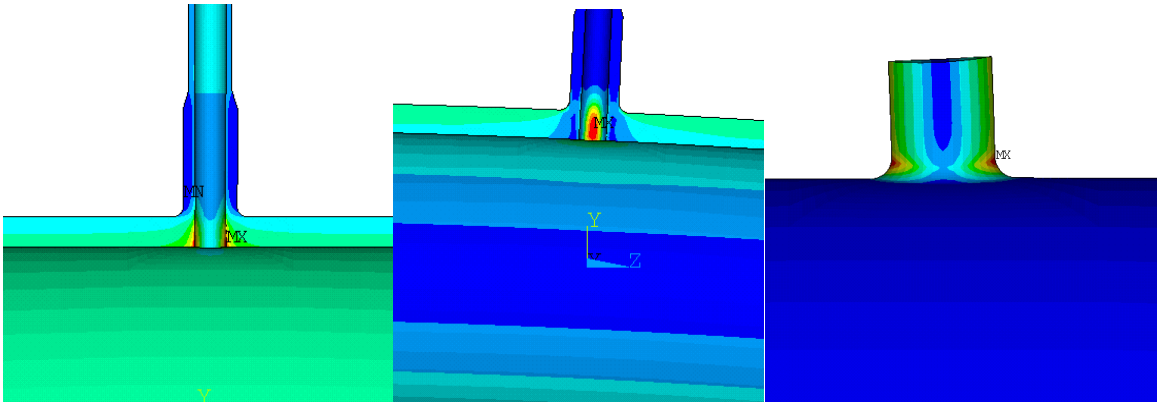

Figure 5 Locations of Maximum Total Stress for a) Pressure, b) In-Plane Run Bending, c) In-Plane Branch Bending

Figure 5 shows the stress contours near the location of maximum total stress for Load Case 1 to 3 with individual loadings. The highest stress with internal pressure occurs inside the run pipe bore on the plane of symmetry (due to hoop stress), while for in-plane run bending it occurs inside the run pipe bore perpendicular to the plane of symmetry (due to axial stress). The highest stress due to in-plane branch bending occurs on the outside of the branch nozzle on the plane of symmetry.

The highest stresses at each location are summarized in Table 2. Except for pure branch bending, the highest stresses occur in the run pipe.

Table 2 Highest Stresses Obtained from Finite Element Analysis for Base Case

Loading	Juncture Location	Stress [ksi]			
		Nominal	Membrane	Mem+ Bend	Total
Pressure	Run	9.33	9.5	12.1	25.4
	Branch	5.56 (pipe) 3.43 (nozzle)	4.32	8.44	
Run Bending	Run	12.0	13.4	15.0	31.7
	Branch	--	5.50	11.7	
Branch Bending	Run	--	0.55	1.06	6.7
	Branch	12.0 (pipe) 3.82 (nozzle)	2.89	5.11	
Combined	Run	--	16.6	17.5	37.1
	Branch	--	7.8	15.1	

VARIATION OF LOADING DIRECTION

The loading direction of applied bending was varied to determine which combination would result in maximum stresses. For the run pipe, inspection of the problem leads to the conclusion that bending in the symmetry plane subjects the nozzle to the highest stress. Keeping the direction of the run bending moment constant, the direction of the branch moment was rotated from bending in the symmetry plane to bending perpendicular to the symmetry plane. The angle between the symmetry plane and the branch pipe bending plane is denoted by ϕ, which varies between 0° (bending in the symmetry plane) to 90° (bending perpendicular to the symmetry plane). The far field bending stress was kept constant as the branch pipe bending plane was rotated.

The resulting combined stresses as a function of ϕ are shown in Figure 6. The effect of rotating the branch pipe moment is overall very small. The highest combined membrane+bending stresses (Load Case 4) are obtained for the branch bending in the plane perpendicular to the symmetry plane. The highest juncture stresses due to branch bending are obtained in the branch nozzle, and the greatest membrane+bending stress for this load case is obtained for bending in the symmetry plane. Membrane stresses are practically unaffected by the variation of branch bending direction.

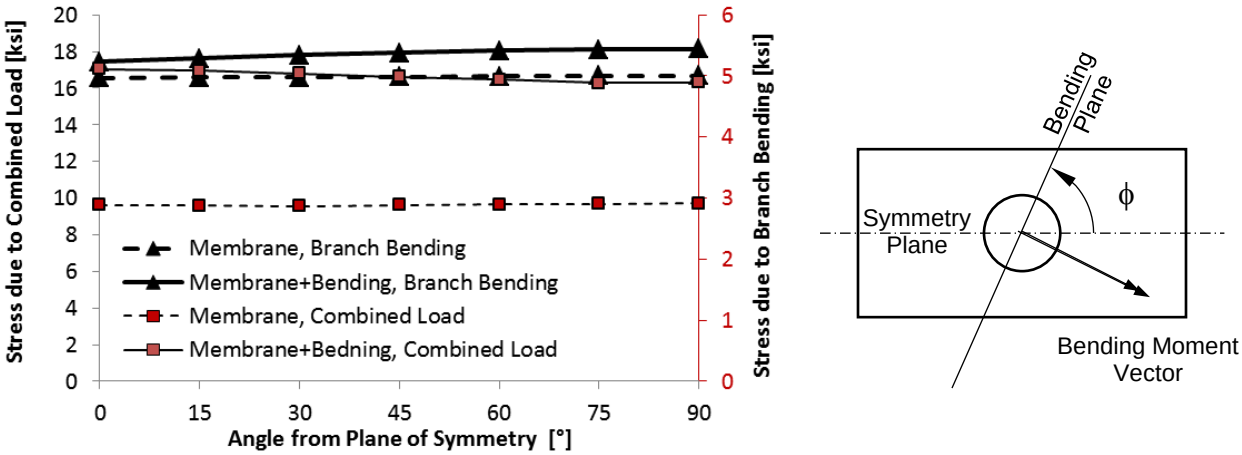

Figure 6 Variation of FE Linearized Stress with Direction of Branch Bending

VARIATION OF REINFORCEMENT LENGTH

The length of the reinforced nozzle, L_1, was varied from the value of the base case/ Dimensionless lengths of $L_1/\sqrt{T_b d_m/2}$ = 6.69 (base case), $L_1/\sqrt{T_b d_m/2}$ = 3.35, $L_1/\sqrt{T_b d_m/2}$ = 1.67 , $L_1/\sqrt{T_b d_m/2}$ = 1.00 and $L_1/\sqrt{T_b d_m/2}$ = 0.80 were analysed. The smallest length is practically identical with the length of reinforcement perpendicular to the run pipe wall, L_N, which according to NB-3643.3 [2] is equal to L_N = 0.5 r_2 + 0.5$\sqrt{T_b\, d_m/2}$ = 0.835 $\sqrt{T_b d_m/2}$ where

r_2 = fillet radius at OD of branch to run transition, in (mm).

The variation of membrane and membrane plus bending stress intensity with the length of the reinforced nozzle is plotted in Figure 7. The dependence of the level of stress on the nozzle length is almost imperceptible. Only the stresses due to branch bending, which are largest at the base of the nozzle, change slightly with nozzle length, the membrane stresses trending slightly up and the membrane plus bending stresses trending slightly down with shorter nozzle length. Similarly, the maximum total stresses for all load cases vary by less than 1%, except for case of branch pipe loading, where the highest stress at the fillet is about 10% lower for the shortest nozzle length than for the longest nozzle.

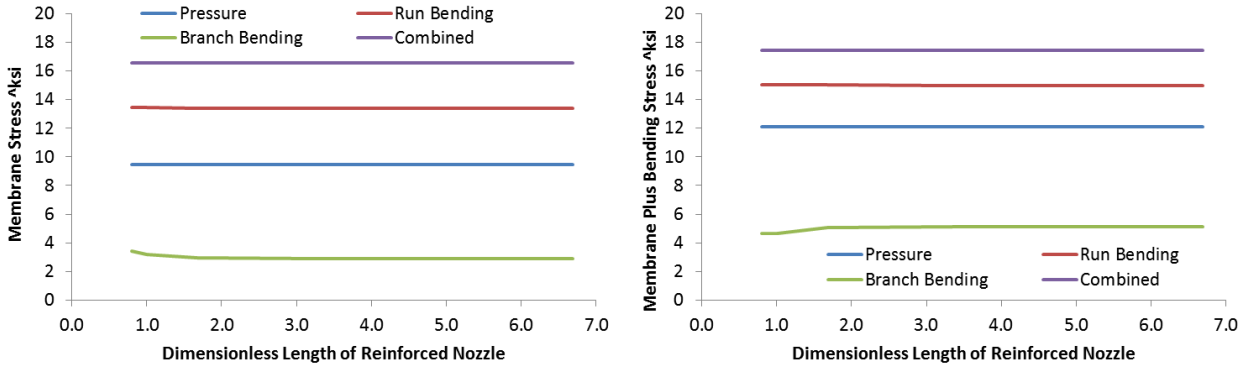

Figure 7 Effect of Reinforced Nozzle Length on Membrane and Membrane Plus Bending Stress

In summary, the stresses at the juncture remain very stable for all investigated lengths of the reinforced nozzle, down to the ASME Code limit of reinforcement perpendicular to the vessel wall. The lack of variation of the stresses suggests that an even smaller value of minimum length could be used.

COMPARISONS OF MODIFIED CODE FORMULAS AND FE ANALYSIS

For the comparison with the piping equations, the ASME Code design equations were used assuming the terms involving temperature are negligible. This is thought to be acceptable and adequate for the purpose of the present comparison. The finite element results were obtained at the piping juncture as membrane, membrane plus bending and total stresses, where the membrane stresses were considered to correspond to the NB-3650 eq. (9), the membrane plus bending stresses to eq. (10) (or eq. (12)) and the total stresses to eq. (11). This equivalency is based on the stress classification beside a nozzle in NB-3200 where in general terms the local membrane stress is classified as primary, the bending stress near the juncture is classified as secondary and the stress maximum shown by a sufficiently refined finite element analysis is the total stress.

In the following the NB-3600 equations with the modifications by eq, (1) to (5) will be called "modified Code equations". The complete set of stress indices for the modified Code equations is given by $B_1 = 0.5$, $B_{2r} = 1.0$, $B_{2b} = 1.0$, $C_1 = 1.45$, $C_{2r} = 1.64$, $C_{2b} = 1.5$, $K_{2r} = 1.6$, $K_{2b} = 2.0$.

Results for individually applied loads (Load Case 1, 2 and 3) are compared in Table 3. Generally, the agreement between the finite element results and the modified NB-3600 Code equations is good. Only for branch bending is there a greater difference, the modified Code equations being conservative. Note that the branch indices are at the lowest possible (fixed) value given by the Code.

Table 3 Comparison of NB-3600 Equations and Finite Element Results – Individual Loads

Load Case	Pressure (1)			Run Bending (2)			Branch Bending (3)		
	Eq. (9) ksi	Eq. (10) ksi	Eq. (11) ksi	Eq. (9) ksi	Eq. (10) ksi	Eq. (11) ksi	Eq. (9) ksi	Eq. (10) ksi	Eq. (11) ksi
FE	4.7	12.1	25.4	13.4	15.0	31.7	2.9	5.1	6.7
Modified 2017 Code	4.7	13.5	27.1	12.0	19.6	31.4	4.1	6.1	12.3

The base Load Case combining pressure, run bending and branch bending loads is Load Case 4. The finite element results and modified Code equations are compared in Table 4. The modified Code equations are now very significantly conservative. The reason for this is that the stress maximum for each individual loading occurs at a different location on the juncture. The combined stress is therefore not much higher than the highest stress due to individual loads. In the Code equations, on the other hand, the

highest stresses all add up without any reduction. The modified Code equations provide very adequate conservatism for combined loading.

For further comparison the stresses from the present Code equations are also shown. The stresses are very conservative, more than 1.7 times up to almost three times the finite element result. This is mainly because the nominal branch pipe stresses are calculated based on the geometry of the attached pipe remote from the juncture.

Table 4 Comparison of NB-3600 Equations and Finite Element Results – Combined Loading

Load Case	Combined Load – Base (4)		
	Eq. (9) ksi	Eq. (10) ksi	Eq. (11) ksi
FE	16.6	17.5	37.1
Modified 2017 Code	20.6	39.0	70.2
2017 Code	28.7	51.2	94.5

Three more load cases are generated to explore different regimes of the load parameter. Load Case 5 has increased pressure such that the NB-3640 limit is reached. The branch pipe bending stress is also increased compared with the base load case. Load Case 6 has the lower pressure of the base case, but increased run and branch bending. Load Case 7 has the same bending as Load Case 6, but no pressure to be comparable with ASME eq. (12). The results are given in Table 5.

Load Case 5: p = 3000 psi, σ_{br} = 12 ksi (run), σ_{bb} = 16 ksi (branch)
Load Case 6: p = 1600 psi, σ_{br} = 17 ksi (run), σ_{bb} = 26 ksi (branch)
Load Case 7: p = 0 psi, σ_{br} = 17 ksi (run), σ_{bb} = 26 ksi (branch)

Table 5 Comparison of NB-3600 Equations and Finite Element Results – Additional Combined Load Cases

Load Case	5			6			7
	Eq. (9) ksi	Eq. (10) ksi	Eq. (11) ksi	Eq. (9) ksi	Eq. (10) ksi	Eq. (11) ksi	Eq. (12) ksi
FE	20.9	22.9	42.9	21.7	23.9	51.5	21.7
Mod. 2017 Code	26.0	52.8	97.8	30.1	54.0	96.9	40.5
2017 Code	36.8	69.0	130.2	47.7	80.3	149.6	66.8

For Load Case 5 the juncture reaches the applicable stress limit for Design conditions, eq. (9) according to the modified Code equations. For Load Case 6, the modified eq. (10) is approximately at 3 S_m. . In all cases, the modified Code equations are seen to be quite conservative relative to the FE results. The original Code equations are very consevrative and could be considered excessively so.

RECOMMENDATIONS FOR AMENDED CODE RULES

The existing Code rules in NB-3600 [2] are confirmed to give excessively conservative stresses, as noted in the papers of Faraz and Gurdal, [3] and [4]. With reinforcement at the juncture, the conservatism occurs because the nominal branch pipe stresses are calculated based on the geometry of the attached pipe remote from the juncture. When the branch pipe stresses are high, the only way to satisfy the piping design rules may be to increase the wall thickness of the entire branch pipe, which is counter-productive if the branch bending stresses are due to displacement controlled loads.

The modified Code equations still give conservative results because the pressure and moment stresses that are combined in the Code equation occur at different locations on the juncture. Additional confidence that the modified Code equations would provide sufficient conservatism comes from the

variation of the reinforced nozzle length. It was shown that the stresses change negligibly down to the NB-3643 length of reinforcement perpendicular to the run pipe wall.

It would therefore be justifiable to implement the modified Code equations as follows,

- Calculate the nominal branch bending stress at the juncture based on dimensions of the reinforcement (nozzle),
- Calculate the stress indices at the juncture based on the dimensions of the reinforcement (nozzle).

An adequate restriction would be that the reinforcement should extend up to the NB-3643 length of reinforcement perpendicular to the run pipe wall, but a smaller value could also be considered.

CONCLUSIONS

The results of a three-dimensional elastic finite element (FE) analysis of a prototypical piping branch connection were compared with the results of a proposed modification of the ASME Code NB-3600 analysis for branch connections. The effect of loading direction was investigated with the FE model. For bending of the pipe run, a bending direction in the plane of the branch connection results in the largest branch stresses. For bending of the branch pipe, the effect of bending direction is generally small. For other than branch bending dominant cases, slightly higher stresses were found for branch bending in the direction perpendicular to the symmetry plane of the joint.

It is concluded that the present investigation supports the proposal by Faraz and Gurdal [3][4] to modify the Code rules in NB-3680 as follows,

- Calculate the nominal branch bending stress at the juncture based on dimensions of the reinforcement (nozzle),
- Calculate the stress indices at the juncture based on the dimensions of the reinforcement (nozzle).

Based on the consideration that the stress distribution at the juncture is influenced by the geometry in the vicinity, not further away, this recommendation is reasonable. It is also rational in that it allows an optimization of the juncture design, with reinforcement providing the proper effect on the calculated stress at the juncture.

The present study investigated a nozzle length down to the length of reinforcement perpendicular to the vessel wall. When varying the length of the branch reinforcement (nozzle) from about 7 times the characteristic shell length down to slightly less than the length of reinforcement perpendicular to the vessel wall per NB-3643, it was found that the stresses at the juncture remain almost unchanged, providing confidence that proper conservatism is achieved by the modified Code equations., The small dependency of stress on nozzle length would also support a shorter minimum required nozzle length.

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