STRESS INTENSIFICATION FACTOR OF BEND

COMPARING ASME NB-3000 WITH STOOMWEZEN D1101

Date: Di 22-Jan-2002
Time: 19:20:58

Project: IMPROVEMENTS OF FUTURE DEVELOPMENTS

Jobnr: PV2002

ABSTRACT:

This Case Study is the result of technical discussions I had with Wim Guijt in 1995 and 1996. The original studies were in Dutch and are translated here for general use and understanding. The aim of this study is directed towards comparing stress intensification factors in elbows subjected to pure in-plane bending moments using the analytical method as described in ASME Section III Subsection NB 3685 and the general method as described in RToD D1101 and ANSI B31.1. The later is generaly adopted by most pipe stress analysis programmes.

The difference between the two codes consists in the fact that ASME uses bends with smooth surfaces and thin walls whereas the other codes use bends made of welded pipe. One would expect that according Markl the bending stress calculated according to D1101 would be 2 times the bending stress calculated according ASME NB-3000. Moreover Markle proofed that the weld has high influence on the fatigue live of the bend.

However this Case study shows the opposite:
The calculated local bending stresses according to NB-3000 are 2 times higher than the calculated cross sectional bending stresses according D1101.
Experiments by a great number of people have prooven the figures calculated by NB-3000 to be in-line with their findings. This would mean that the stresses calculated according to D1101 under-estimate the occuring stresses.

REFERENCES:

Stress intensification factor SP95039 Paul W.H. Voorhaar 1995
Dienst voor het Stoomwezen RToD D1101
NEN 3650 Transportleiding systemen 1992
ASME Section III div. 1 table NB3685.1-2

Stress Intensification factors in Elbows Subjected to
In-plane Bending Moments at Elevated temperatures F.G. Cesari 1986

Stress Indices and Stress Intensification Factors of
Pressure Vessel and Piping Components R.W. Schneider / E.C. Rodabaugh 1981

Fatigue Tests of Piping Components A.R.C. Markl 1952

0	PVo	Zo 20-Jan-2002	First Issue			
REV	BY	DATE	DESCRIPTION	CHECKED	PROJECT APPROVAL	THIRD PARTY APPROVAL

STATUS CODE	DOCUMENT NUMBER	REVISION	STATUS
A Preliminary for information only B For review C Authorized for construction	**WEBSITE VOORHAAR**	**0**	**B**

In-Plane Bending of Curved Circular Tubes

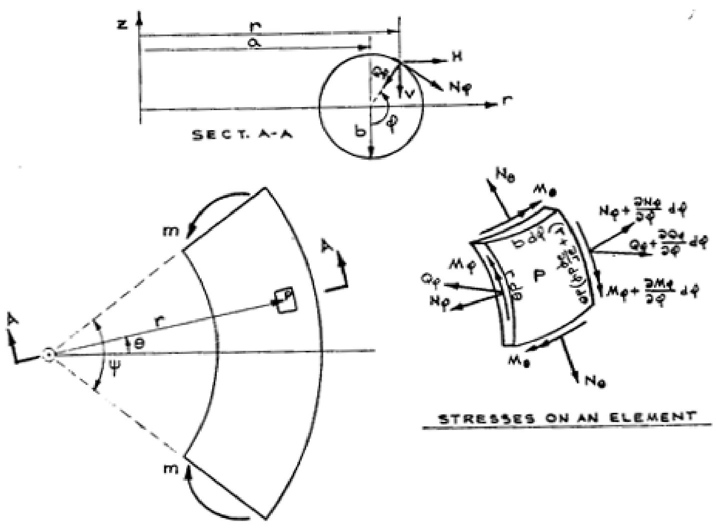


Fig. 1 Curved circular tube

This case study was made to show the difference between the stress indices of ASME NB-3000 and Rules for Pressure Vessels RToD D1101. It shows that the local bending stresses calculated according to ASME NB-3000 are a factor 2 higher than the bending stresses calculated with RToD D1101.

Input Data

$D_o := 914.4 \cdot mm$ Outside diameter

$t := 6.35 \cdot mm$ Wallthickness

$Allowance := 0 \cdot mm$ Allowance

$t_m := t - Allowance$ Netto wallthickness

$R := 5 \cdot D_o$ Elbow Radius $R = 4572 \cdot mm$

$E := 204000 \cdot \frac{N}{mm^2}$ E modulus

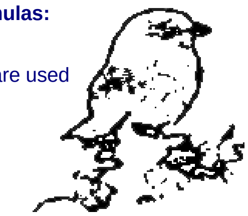
$\mu := 0.30$ Poisson ratio

Design pressure range:

$$P^T = (0 \quad 0.16 \quad 0.32 \quad 0.48 \quad 0.64 \quad 0.8 \quad 0.96 \quad 1.12 \quad 1.28 \quad 1.44 \quad 1.6) \cdot \frac{N}{mm^2}$$

$r := \frac{D_o - t_m}{2}$ Mean pipe radius $r = 454.025 \cdot mm$

From Table NB-3685.1-2 from ASME Section III, Division 1 NB-3000 we derive the following formulas:



$$\lambda := \frac{t_m \cdot R}{r^2 \cdot \sqrt{1-\mu^2}} \qquad \lambda = 0.148$$

Equations are valid for $\lambda \geq 2$ only. But equations are used in the absence of more applicable ones.

$$\tau_q := \frac{P_q \cdot R^2}{E \cdot r \cdot t_m} \qquad \tau_1 = 0.0057$$

$$X_{1_q} := 5 + 6 \cdot \lambda^2 + 24 \cdot \tau_q \qquad X_{1_1} = 5.267$$

$$X_{2_q} := 17 + 600 \cdot \lambda^2 + 480 \cdot \tau_q \qquad X_{2_1} = 32.808$$

$$X_{3_q} := X_{1_q} \cdot X_{2_q} - 6.25 \qquad X_{3_1} = 166.557$$

$$X_{4_q} := \left(1 - \mu^2\right) \cdot \left(X_{3_q} - 4.5 \cdot X_{2_q}\right) \qquad X_{4_1} = 17.219$$

$$\phi_i := (-90 + i) \cdot deg \qquad \phi_{90} = 0 \cdot deg \qquad \phi_0 = -90 \cdot deg$$

Inplane stress indices according to ASME section III div.1 table NB3685.1-2

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$$i_{x_{i,q}} := \sin\left(\phi_i\right) + \frac{\left(1.5 \cdot X_{2_q} - 18.75\right) \cdot \sin\left(3 \cdot \phi_i\right) + 11.25 \cdot \sin\left(5 \cdot \phi_i\right)}{X_{4_q}}$$

$$i_{y_{i,q}} := \lambda \cdot \frac{9 \cdot X_{2_q} \cdot \cos\left(2 \cdot \phi_i\right) + 225 \cdot \cos\left(4 \cdot \phi_i\right)}{X_{4_q}}$$

Stress intensification factor with rerounding according to Stoomwezen D1101

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$$X_i := 3.25 \cdot \frac{r^{\frac{11}{6}} \cdot R^{\frac{2}{3}}}{t^{\frac{5}{2}}} \qquad h := \frac{t \cdot R}{r^2} \qquad i_{D1101} := \frac{0.9}{h^{\frac{2}{3}}}$$

$$i_{D1101_q} := \left(\frac{i_{D1101}}{1 + \frac{P_q}{E} \cdot X_i}\right)$$

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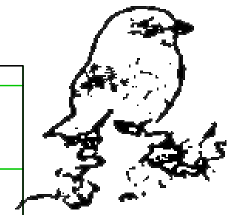

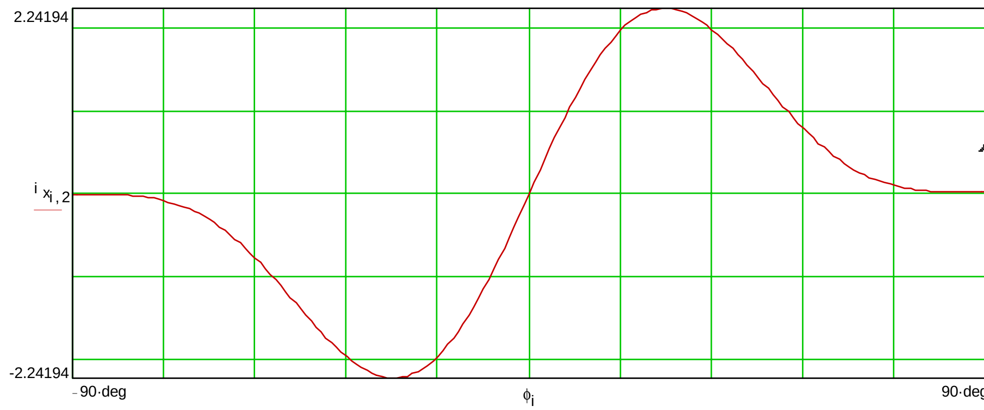

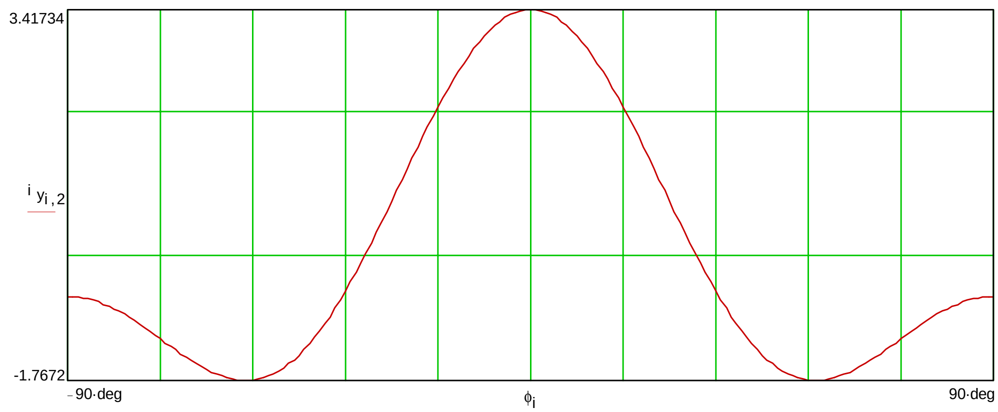

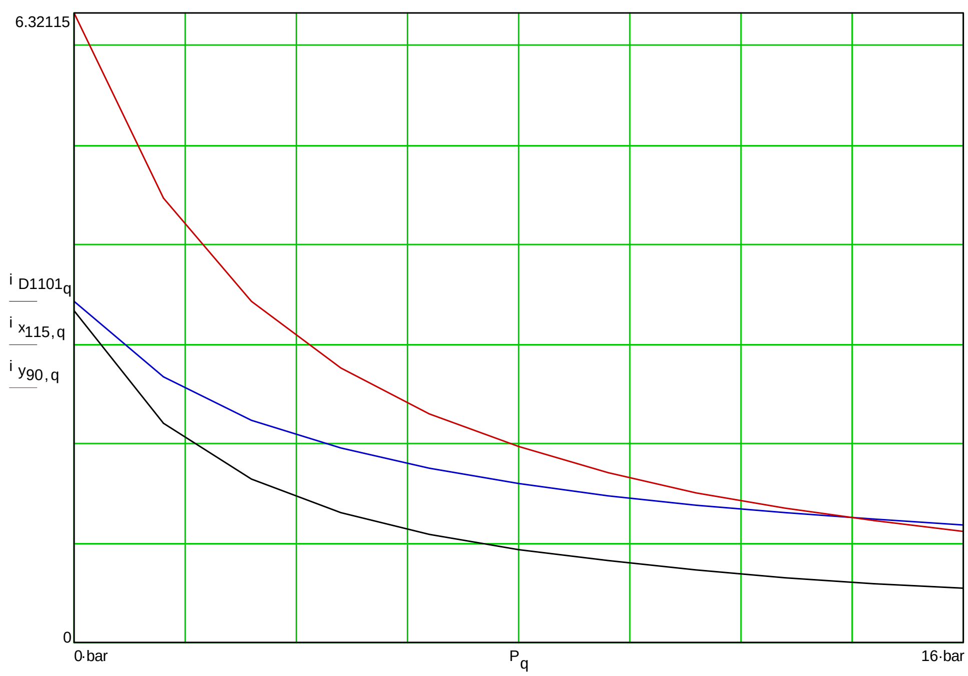

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According to ASME Section III div.1 table NB3685.1-2 we calculate the stress indices, the maximum of these will be taken to compair with Stoomwezen RToD D1101

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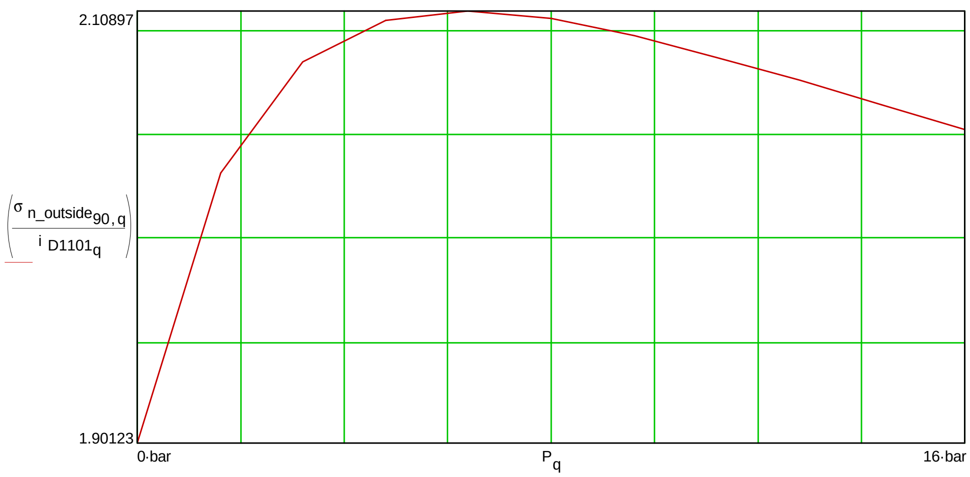
i_{NB3000} is maximum of:

$\sigma_{n\_outside} := \mu \cdot i_x + i_y$ $\quad$ $\max(\sigma_{n\_outside}) = 6.439$ $\quad$ $i\_max\_outside := \max(\sigma_{n\_outside})$

$\sigma_{n\_mid} := \mu \cdot i_x$ $\quad$ $\max(\sigma_{n\_mid}) = 1.026$ $\quad$ $i\_max\_mid := \max(\sigma_{n\_mid})$

$\sigma_{n\_inside} := \mu \cdot i_x - i_y$ $\quad$ $\max(\sigma_{n\_inside}) = 3.601$ $\quad$ $i\_max\_inside := \max(\sigma_{n\_inside})$

Ratio of stress indice of NB 3000 and RToD D1100 shows resemblance with factor of 2



$i_{D1101} := \max(i_{D1101})$ $\quad$ $i_{D1101} = 3.325$ $\quad$ Maximum intensification according D1101

$i_{NB3000} = 6.439$ $\quad$ Maximum intensification according NB3000

$\frac{i_{NB3000}}{i_{D1101}} = 1.937$

Assuming that the location of the highest stresses according to D1101 is at the same location as according NB3000 the ratio is close to 2.

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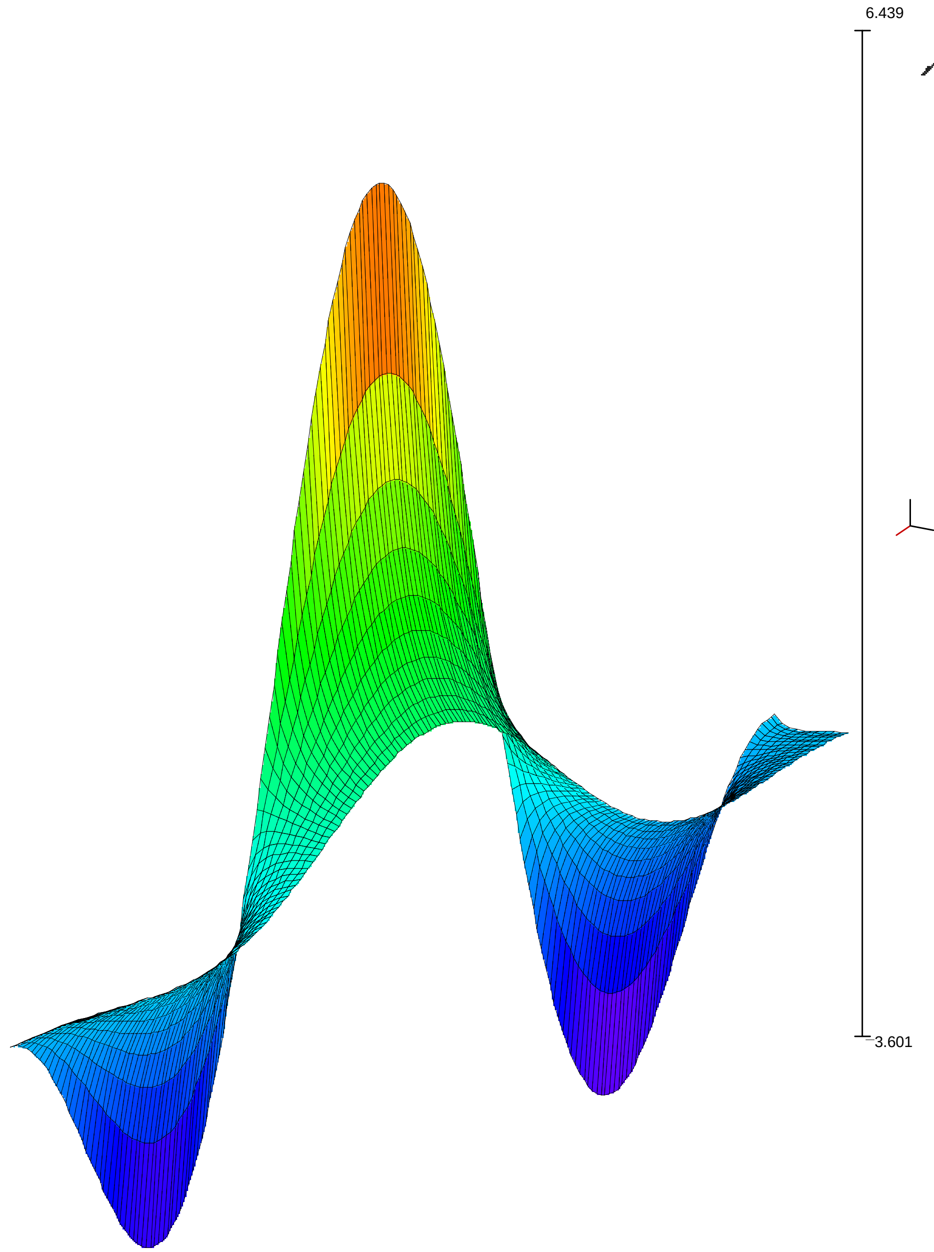
Stress intensification of bending stress for the whole pressure range and from 0 to 90 deg



Stress_Intensification_NB3000