OPENING DESIGN METHODS

COMPARISION OF CODE RULES WITH FEA RESULTS

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ABSTRACT

Design of openings in shell and heads is one of the most important parts of the mechanical design of pressure vessels and due to its importance ASME code has devoted more than one section to this issue. Each section has its own approach, theoretical bases and of course limitations. Some of them can be used alternatively and for some problems requirements of more than one section has to be fulfilled. The fact that multiple set of rules for design of openings exist in ASME has made it a little confusing for designers to figure out which section should be applied to each specific problem. In this paper we have tried to provide designers with a deeper insight toward use of ASME code for design of openings by opening up the theoretical background of some sections, solving illustrative examples and comparison of FEA results with CODE results.

In the first chapter all ASME sections which are related to design of opening has been listed and application and limitation of each one is specified, furthermore it has been clarified when some sections should be used instead of or in addition to other sections.

In the second chapter the area replacement method of division 1 and the pressure area method of division 2 being the two most important design methods have been investigated. The theoretical basis of pressure area method according to **Jawad** has been explained. An example is solved by FEA, to verify the assumptions of analytical formulation.

In the third chapter an example is solved with three methods: Pressure area (Div.2), Area Compensation (Div.1) and FEA and results are summarized in tables and curves, to facilitate comparison of methods.

Finally, calculation procedures of area compensation and pressure area method are depicted in two flowcharts which are given in appendix 1 and 2.

1. OPENING DESIGN METHOD AND THEIR LIMITATIONS IN ASME

In table 1 all parts of ASME which may be used for design of nozzle reinforcement are listed and limitation of each one is specified. Some sections can be used instead of each other and in some cases more than one Section should be applied in order to properly design an opening.

For example for openings which fall in to limits of UG-36 and under internal pressure Rules of UG-37 to UG-42 or appendix 1-10 or appendix 1-9 may be used.

But for nozzles exceeding limits of UG-36, supplemental rules of appendix 1-7 shall be satisfied in addition to the rules of UG-37 to UG-42. Alternatively, openings in cylindrical or conical shells exceeding UG-36 limits may be designed for internal pressure, using only the rules of 1-10.

Openings in vessels not subjected to rapid fluctuations in pressure do not require reinforcement if requirement of UG-36(c)(3) is satisfied.

Appendix 1-10 is based on pressure area method which can be used for all nozzles on cylinders and cones including large openings. It is an alternative method to UG-37 procedure which results in a more accurate and economical design. But it is not applicable to external pressure cases and also does not specify any provisions for weld design. So when appendix 1-10 is applied welds should be designed according to ASME DIV.2 part 4-5-14.

ASME DIV.2 part 4-5 is similar to the method of appendix 1-10 but covers both internal and external pressure cases.

2. DESIGN METHODS

The traditional area-replacement method has been used in ASME pressure vessel and piping codes for many years. The area-replacement concept requires that the metal removed to make an opening be replaced by an equal area of reinforcement within a prescribed region around the opening. This concept is still used in the ASME B&PV Code, Section VIII, Division1, paragraph UG-37 for the design of reinforcement at openings in shells and formed heads. However, a substantial amount of information accumulated in recent years indicates that the area replacement method may lead to excessive conservatism in some applications.

REFERENCE	SHELL ID or thk.	NOZZLE ID	PRESSURE	RATIO	MATERIAL	PARENT ATTACHMENT
UG-36 to UG-43 are applicable	<=60" (1500mm)	D_i /2 <20" (500mm)	P_{in} , $P_{ext.}$			
	>60" (1500mm)	D_i /3< 40" (1000mm)	P_{in} , $P_{ext.}$			
Appendix 1-7 (Large opening)	=>60" (1500mm)	=>40" (1000mm) And >3.4 (Rt)^1/2	P_{in} , $P_{ext.}$	Rn/R<0.7		Radial Nozzle in a Cylindrical & Conical Shell. Half alfa ≤30˚
UG-36(c)(3) (Small opening)	Thk. <=3⁄8 in. (10 mm)	31⁄2 in. (89 mm)	P_{in} , $P_{ext.}$			
	Thk.>3⁄8 in. (10 mm);	23⁄8 in. (60 mm)				
Appendix 1-9		Integrally reinforced type of nozzles <=24" (600mm)	P_{in}	YS/TS<0.8 opening diameter: dm vessel diameter: Dm vessel thickness: t For (dm/Dm) > 0.5 (Dm/t) ≤ 100 For (dm/Dm) ≤ 0.5 (Dm/t) ≤ 250	UCS-23 UHA-23	Radial Nozzle in a Cylindrical & Conical Shell
Appendix 1-10			P_{in}			Radial & hillside nozzles in a Cylindrical & Conical Shell..
SEC VIII DIV.2 PART 4.5			P_{in} , $P_{ext.}$	IDs/Ts<= 400mm the ratio of the diameter along the major axis to the diameter along the minor axis of the finished nozzle opening shall be less than or equal to 1.5.		

Table 1) All parts of ASME which may be used for design of nozzle reinforcement

In order to overcome the over conservatism of the area-replacement method, the pressure-area method was recently introduced in the ASME B&PV Code, Section VIII, Division 2, Part4, paragraph 4.5.The pressure-area method is based on ensuring that the resistive internal force provided by the material is greater than or equal to the reactive load from the applied internal pressure. In this paper the basic theory behind the pressure-area method that is incorporated in the ASME B&PV Code, Section VIII, Division 2 is presented. The nozzle rules of ASME B&PV Code, Section VIII, Division 2, Part 4, paragraph 4.5 along with a commentary providing background and insight to the rules is provided.

2.1) ASME SEC. VIII DIV.1: AREA COMPENSATION

In ASME code area replacement method is used. Opening is reinforced by providing material near the hole in excess of the minimum thickness required for the individual components considered as a shell without any opening. The area of additional material is required to be at least equal to the area removed by making the hole in a shell of minimum thickness. The common nozzle configuration used in the ASME Section VIII Division 1 Code is illustrated in Figure 2.1.1. Design procedure is depicted in appendix 1.

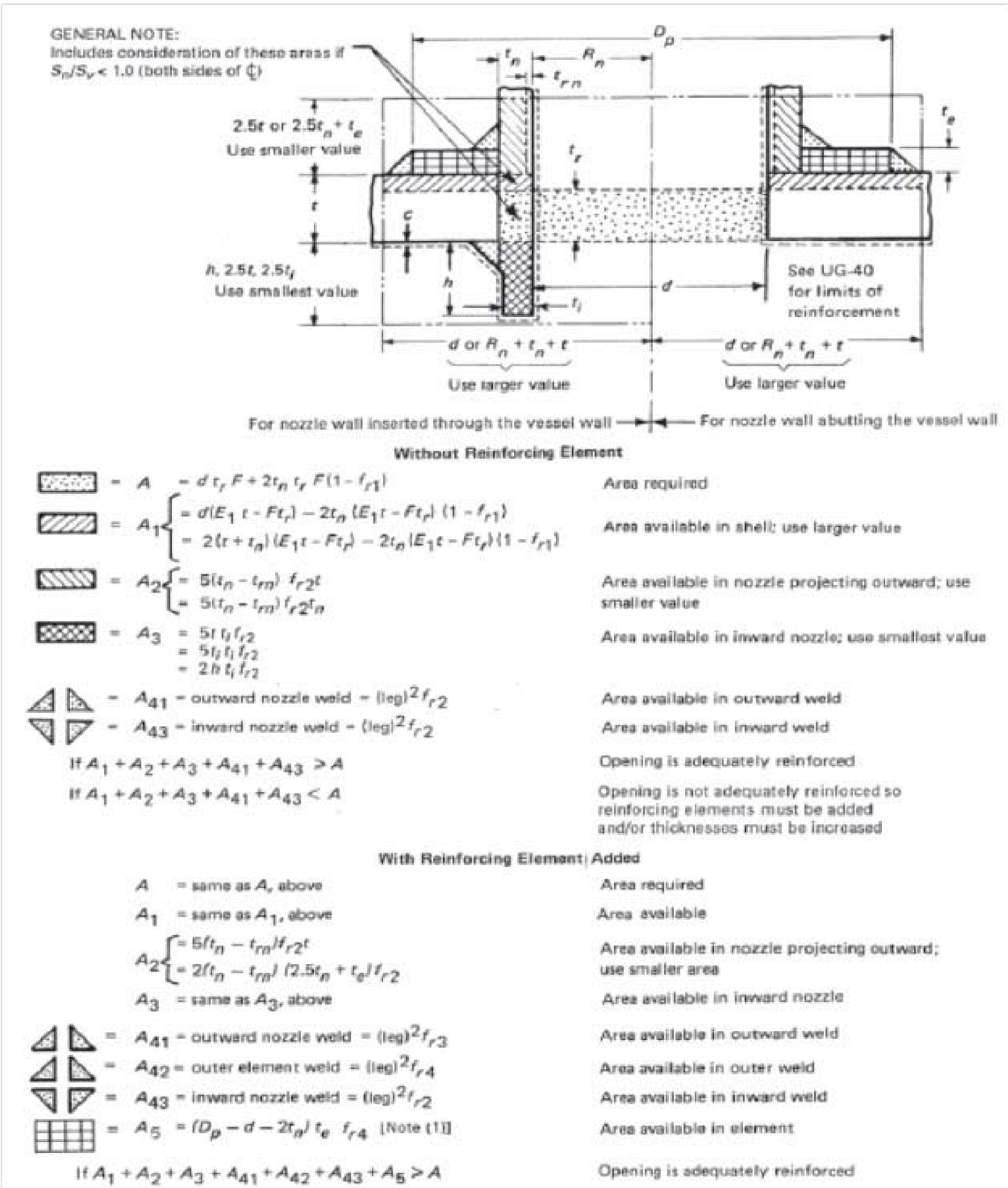


Figure 2.1.1) NOMENCLATURE AND FORMULAS FOR REINFORCED OPENINGS

2.2) DESIGN OF PLATE AND SHELL STRUCTURES: PRESSURE AREA THEORY

The pressure area analysis is based on the concept that the pressure contained in a given area within a shell must be resisted by the metal close to that area. Referring to fig. 2.2.1(a), the total force in the shaded area of the cylinder is (r)(P)(L) while the force supported by the available metal is (L)(t)(σ). Equating these two expressions results in t = pr/σ which is the equation for the required thickness of a cylindrical shell. Similarly for spherical shells, Fig. 2.2.1(b), Gives

$$(R\phi)\ (P)(R)\left({}^{1}/{}_{2}\right) = (R\phi)(t)(\sigma)$$

$$T = {}^{PR}/{}_{2\sigma}$$

Referring to Fig. 2.2.2a, it is seen that pressure area A is contained by the cylinder wall and pressure area B is contained by the nozzle wall. However, pressure area C is not contained by any material. Thus we must add material, M, at the junction. The area of material M is given by

$$(P)(R)(r) = (\sigma)(M)$$
$$M = (P)(R)(r)/\sigma$$

For a spherical shell, the required area, M, from Fig. 2.2.2b is,

$$(\mathrm{P})(\mathrm{R})(\mathrm{r})\left({}^{1}/{}_{2}\right) = (\sigma)(\mathrm{M})$$
$$M = ({}^{1}/{}_{2})(P)(R)(r)/\sigma$$

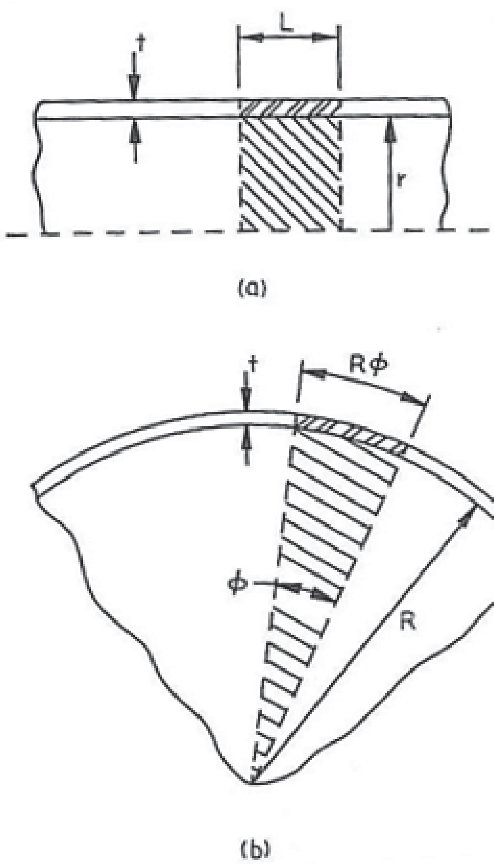


Figure 2.2.1) PRESSURE AREA INTERACTION

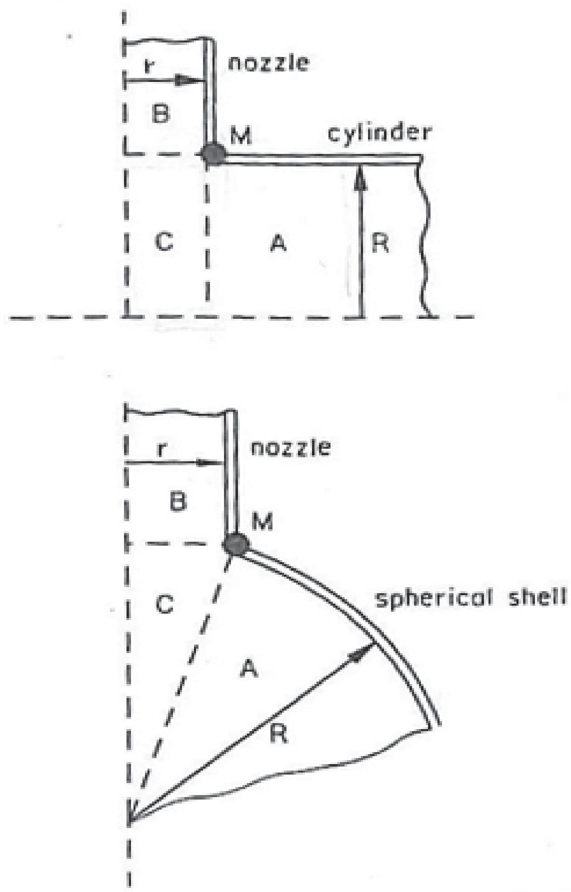


Figure 2.2.2) NOZZLE JUNCTIONS

The required area is added either to the shell, nozzle, or as a reinforcing pad as shown in Fig. 2.2.3.

The pressure area method can also be applied for junctions between components as shown in Fig. 2.2.4. Referring to Fig.2.2.4a, the spherical shell must contain the pressure within area ABC. The cylindrical shell contains the pressure within area AOCD. At point A where the spherical and cylindrical shells intersect, the pressure area to be contained at point A is given by AOC. However, because area AOC is used both in the ABC area for sphere and AOCD for the cylinder, and because it can be used only once, this area must be subtracted from the total calculated pressure in order to maintain equilibrium. In other words, this area causes compressive stress at point A. the area required is given by:

$$A = (r)\left(\sqrt{R^2 - r^2}\right)\left({}^{1}/_{2}\right)(P)/\sigma$$

Where σ, is the allowable compressive stress.

In Fig. 2.2.4b, pressure area A is contained by the cylindrical shell and area C by the spherical shell. Area B is contained by the transition shell. The transition shell is in tension because area B is used neither in the area A nor area B calculations.

In Fig. 2.2.4c, pressure area A is contained by the cone and area B by the cylinder. The transition shell between the cone and the cylinder contains pressure area C which is in tension and area D which is in compression.

Summation of areas C and D will determine the state of stress in the transition shell.

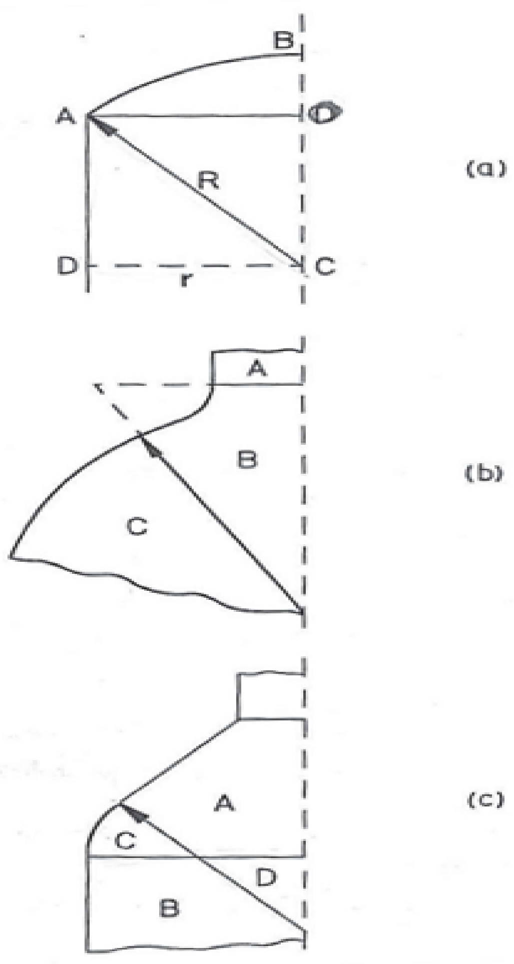


Figure 2.2.3) NOZZLE REINFORCEMENT

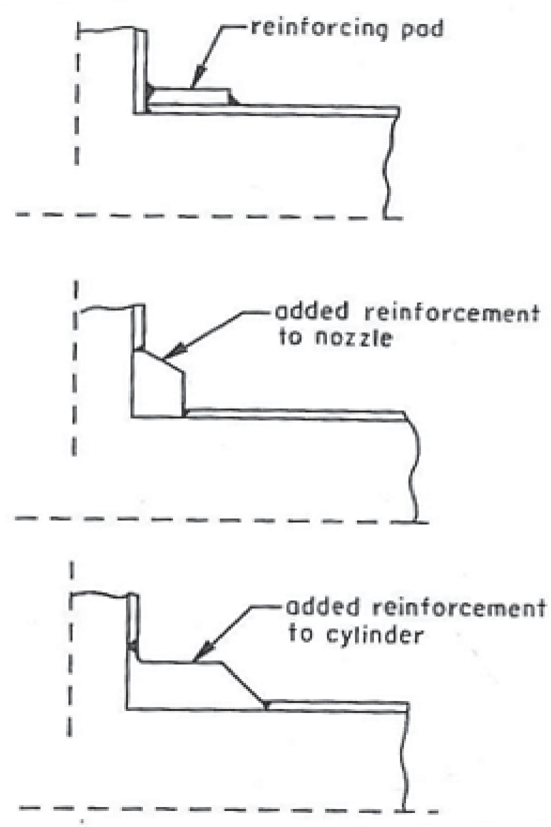


Figure 2.2.4) VARIOUS SHELL JUNCTIONS

Consequently, this part results in below formulas which are the fundamental formulas for ASME SEC. VIII DIV.2:

$$\sigma_{avg} = \frac{(PA_n + PA_s + PA_m)}{A_T}$$

$$\sigma_{circ} = \frac{PR_s}{t}$$

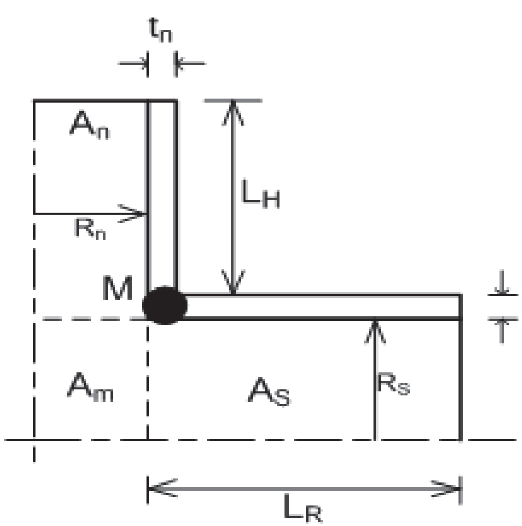
Where;

$$A_n = R_n L_H$$

$$A_s = R_s L_R$$

$$A_m = R_n R_s$$

$$A_T = A_1 + A_2 = L_R \mathrm{t} + L_H t_n$$



2.3) ASME SEC. VIII DIV.2: PRESSURE AREA

Based on this division to design a radial nozzle in a cylindrical shell subject to pressure loading, the average local primary membrane stress and the general primary membrane shall be determined as shown below:

$$\sigma_{avg} = \frac{(f_N + f_S + f_Y)}{A_T}$$

$$\sigma_{circ} = \frac{PR_{xs}}{t_{eff}}$$

Where :

f_N = $\mathrm{PR_{xn}}(\mathrm{L_H} - \mathrm{t})$: force from internal pressure in the nozzle outside of the vessel

f_S = $\mathrm{PR_{xs}}(\mathrm{L_R} + \mathrm{t_n})$: force from internal pressure in the shell

f_Y = $\mathrm{PR_{xs}}\,\mathrm{R_{xn}}$: discontinuity force from pressure.

A_T : total area within the assumed limits of reinforcement.

R_{xs} : shell radius for force calculation.

R_{xn} : nozzle radius for force calculation.

R_{nc}: radius of the nozzle opening in the vessel along the long chord, for radial nozzles $R_{nc} = R_n$

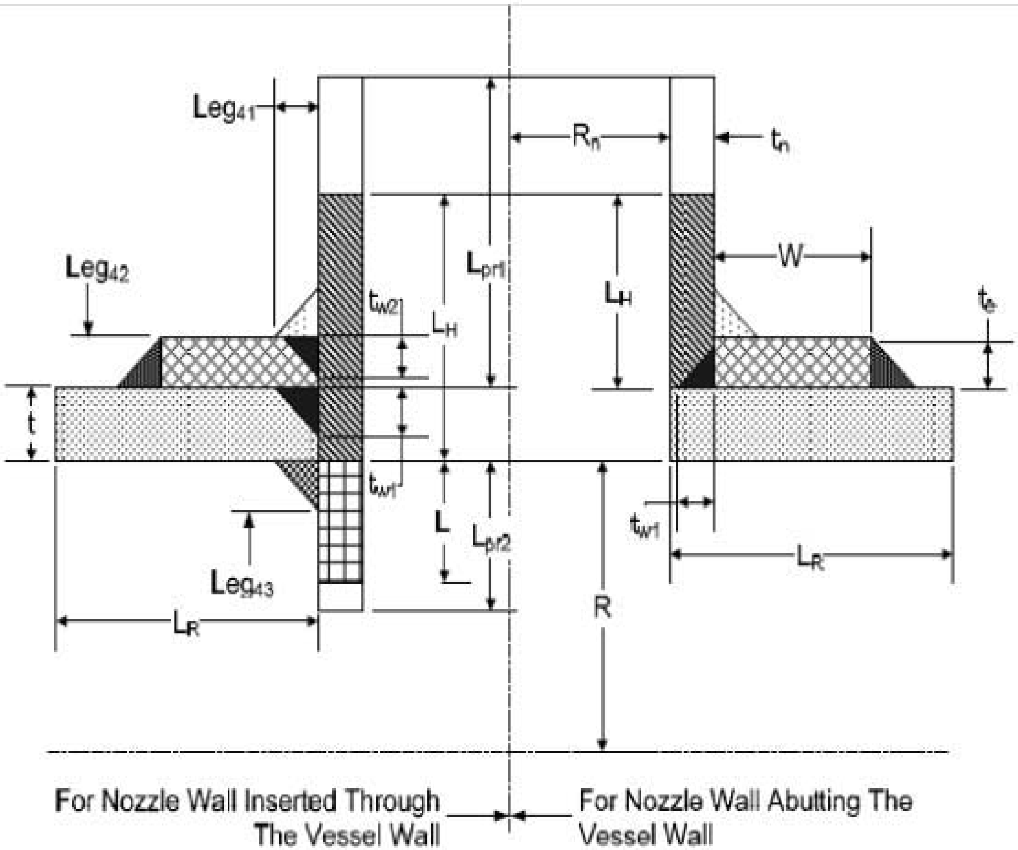
t_{eff} : effective thickness used in the calculation of pressure stress near the nozzle opening.

$$R_{xs} = \frac{t_{eff}}{\ln\left(\frac{R_{eff} + t_{eff}}{R_{eff}}\right)} \cong R_s$$

$$R_{xn} = \frac{t_n}{\ln\left(\frac{R_n + t_n}{R_n}\right)} \cong R_n$$

The denominator can be expanded as below :

$$\ln\left(\frac{R_n + t_n}{R_n}\right) = Ln\left(1 + \frac{t_n}{R_n}\right)$$
$$= \frac{t_n}{R_n} - \frac{\left(\frac{t_n}{R_n}\right)^2}{2} + \frac{\left(\frac{t_n}{R_n}\right)^3}{3} \ldots$$
$$\cong \frac{t_n}{R_n} \qquad \frac{t_n}{R_n} \leq 1$$



Determine the contributing areas as applicable where L_R, L_H, and L are determined from the procedures in paragraphs 4.5.6 and 4.5.11.

= A_1 = Area contributed by shell

= A_2 = Area contributed by nozzle projecting outward

= A_3 = Area contributed by nozzle projecting inward

= A_{41} = Area contributed by outward weld

= A_{42} = Area contributed by pad to vessel weld

= A_{43} = Area contributed by inward weld

= A_5 = Area contributed by reinforcing pad

A_T = Total area contributed

Notes: 1. Do not include any area that falls outside of the limits defined by L_H, L_R, and L_I. For example, if $W \geq L_R$, then $W = L_R$ and $A_{42} = 0.0$.

2. In accordance paragraph 4.1.4.1, all dimensions are in the corroded condition.

Figure 2.3.1) NOMENCLATURE FOR REINFORCED OPENINGS

$$t_{eff} = t(\frac{tL_R + A_5 f_{rp}}{tL_R})$$

$$f_{rn} = \frac{S_n}{S} \qquad f_{rp} = \frac{S_p}{S}$$

Then total available area should be determined:

$$A_T = A_1 + f_{rn}(A_2 + A_3) + A_{41} + A_{42} + A_{43} + f_{rp}A_5$$

Where:

$$A1 = (tL_R).correction\ factor$$

$$1 < correction\ factor = max\left[(\frac{\lambda}{5})^{0.85}, 1.0\right] < 2.1$$

$$\lambda = min\left[\left(\frac{(2R_n + t_n}{\sqrt{(D_i + t_{eff})t_{eff}}}\right), 12.0\right]$$

L_R : effective length of the vessel wall

For integrally reinforced nozzles:

$$L_R = \min[\sqrt{R_{eff}t}, 2R_n]$$

For nozzles with reinforcing pads:

$$L_R = \min[L_{R1}, L_{R2}, L_{R3}]$$

D_i : inside diameter of a shell or head

$$R_{eff} = 0.5D_i$$

$$L_{R1} = \sqrt{R_{eff}t} + W$$

$$L_{R2} = \sqrt{(R_{eff} + t)(t + t_e)}$$

$$L_{R3} = 2R_n$$

for variable thickness openings Where, $L_H > L_{pr3} + t$

$$A_2 = t_n(L_{pr3} + t) + 0.78\left(\frac{t_{n2}^2}{t_n}\right)\sqrt{R_n}t_{n2}$$

for variable thickness openings Where, $LH < L_{pr3} + t$

Or, for uniform thickness openings

$$A_2 = t_nL_H$$

L_H : effective length of nozzle wall outside the vessel

= min [LH1 , LH2 , LH3]

$$L_{H1} = t + t_e + \sqrt{R_nt_n}$$

$L_{H2} = L_{pr1} + t$ for nozzles inserted through the vessel wall

$L_{H2} = L_{pr1}$ for nozzles abutting the vessel wall

$L_{H3} = 8(t + t_e)$

$$A_3 = t_nL_I$$

L_I : effective length of nozzle wall inside the vessel = min [LI1 , LI2 , LI3]

$$L_{I1} = \sqrt{R_nt_n}$$

$$L_{I2} = L_{pr2}$$

$$L_{I3} = 8(t + t_e)$$

$$A_{41} = 0.5L_{41}^2$$

$$A_{42} = 0.5L_{42}^2$$

$$A_{43} = 0.5L_{43}^2$$

A_5 =min [A_{5a} , A_{5b}]

$A_{5a} = Wt_e$

$A_{5b} = L_Rt_e$ for nozzle inserted through the vessel wall

$A_{5b} = (L_R - t_n)t_e$ for nozzles abutting the vessel wall

Determine the maximum local primary membrane stress at the nozzle intersection:

$$P_L = max\left[(2\sigma_{avg} - \sigma_{circ}), \sigma_{circ}\right]$$

PL is determined from the calculated average membrane stress using a linear stress distribution. The assumed stress distribution is shown in Figure 2.3.2. The stress increases from the hoop stress in the shell, at a distance of LR from the nozzle, to a maximum value in the shell equal to PL near the opening. In order to verify the assumed stress distribution an example has been solved with FEA. A cylindrical shell with a nozzle opening has been considered and hoop stress distribution for five different nozzle thicknesses has been obtained with FEA.

The cylindrical shell inside diameter is equal to 2000mm, its metal thickness is equal to 12mm and it is subjected to 1Mpa internal pressure. Five different nozzle thicknesses has been considered for each of which the membrane hoop stress has been evaluated at five points located at 0, 40, 80, 120 and 160mm from the nozzle edge. The shell and the nozzle has been modeled in workbench and meshed with 3d solid elements (Figure 2.3.4) stress linearization has been performed at specified locations to obtain membrane stresses. The results are plotted in figure 2.3.3 and tabulated in table 2.3.1. Each curve in figure 2.3.3 shows the stress distribution corresponding to a nozzle thickness. As it is evident in figure 2.3.3 for all nozzle thicknesses, membrane hoop stress rises or falls in a linear trend. This confirms the assumption of linear stress distribution that was made for calculation of the maximum local stress (Pl) from the average stress (σ_{avg}).

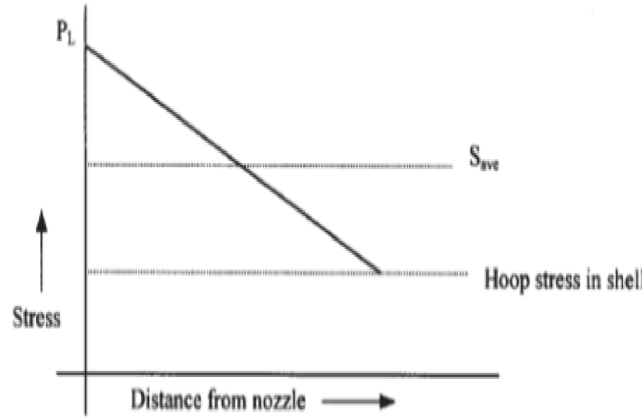


Figure 2.3.2) ASSUMED STRESS DISTRIBUTION AT SHELLDISCONTINUITY

Another interesting fact which is shown in curves of figure 2.3.3 is the nozzle wall reinforcement counteracting effect on stress concentration. As the nozzle wall thickness increases from 30mm to 90mm the maximum local stress decreases. For nozzle thickness 30mm and 40mm the stress concentration effect is dominant, for nozzle thickness, 50mm, the stress concentration and the reinforcement effects are in equilibrium and for nozzle thickness 70 and 90 the reinforcement effect results in local stresses that are even less than the circumferential hoop stress far away from the nozzle edge.

Stress contours are shown in figure 2.3.4, for all five nozzle thicknesses.

distance from nozzle edge	case	DP 1	DP 2	DP 3	DP 4	DP 5
	Nozzle Thk.(mm)	30	40	50	70	90
0	Membrane Stress (Mpa)	127.89	99.68	81.80	62.51	54.25
40		105.69	90.05	80.70	70.34	65.35
80		87.71	82.95	80.07	77.18	74.92
120		84.77	82.47	81.03	79.99	78.96
160		81.80	81.78	81.82	82.35	82.09

Table 2.3.1) MEMBRANE STRESS IN SHELL FOR DIFFERENT NOZZLE THICKNESS

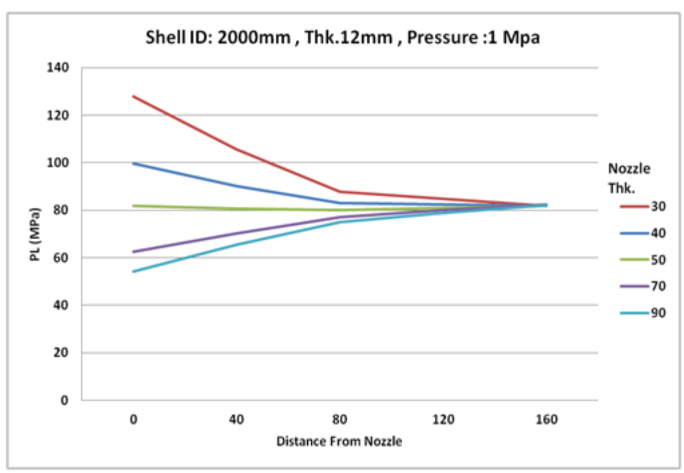
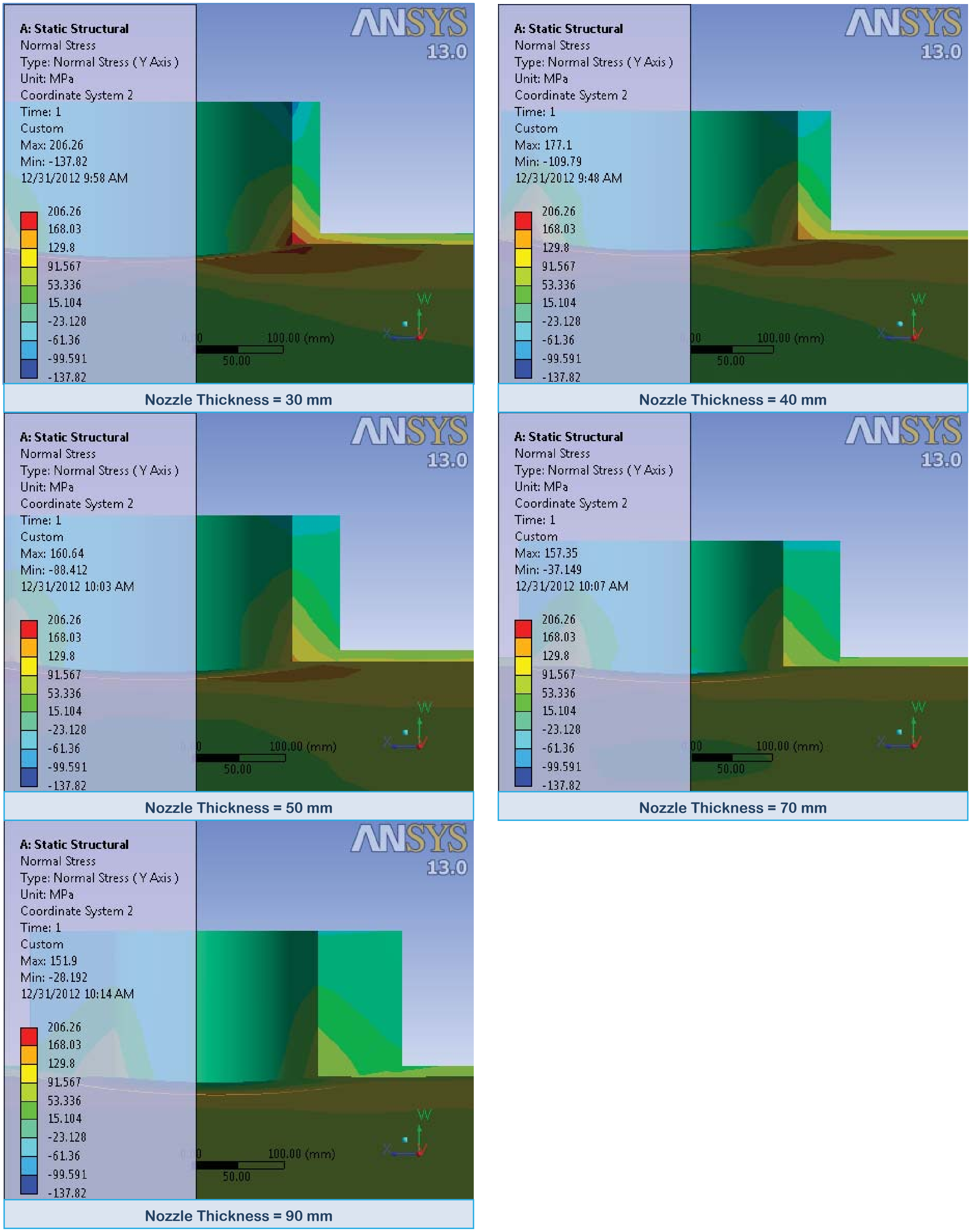


Figure 2.3.3) MEMBRANE STRESS IN SHELL FOR DIFFERENT NOZZLE THICKNESS



Nozzle Thickness = 30 mm

Nozzle Thickness = 40 mm

Nozzle Thickness = 50 mm

Nozzle Thickness = 70 mm

Nozzle Thickness = 90 mm

Figure 2.3.4) Stress counters showing the stress distribution in cylindrical shell with different nozzle thicknesses

The calculated maximum local primary membrane stress should satisfy below equation:

$$P_L \leq S_{allow}$$

Where;

$S_{allow} = 1.5SE$ For internal pressure

$S_{allow} = F_{ha}$ For external pressure

And F_{ha} is allowable stress for external pressure and is adjusted as follow:

$$F_{ha} = \frac{F_{he}}{FS}\frac{E_t}{E_y}$$

Where;

E_t : Tangent Modulus

The method for calculating the Tangent Modulus is to use the External Pressure charts in Section II, Part D, Subpart 3. The appropriate chart for the material under consideration is assigned in the column designated External Pressure Chart Number given in Tables 1A or 1B. The tangent modulus, Et , is equal to 2B/A , where A is the strain given on the abscissa and B is the stress value on the ordinate of the chart.

F_{he} : elastic buckling stress

$$F_{he} = \frac{1.6C_h E_y t}{D_o}$$

$$M_x = \frac{L}{\sqrt{R_o t}}$$

$C_h = 0.55(\frac{t}{D_o})$ For $M_x \geq 2(\frac{D_o}{t})^{0.94}$

$C_h = 1.12M_x^{-1.058}$ For $13 < M_x < 2(\frac{D_o}{t})^{0.94}$

$C_h = \frac{0.92}{M_x - 0.579}$ For $1.5 < Mx \leq 13$

Ch = 1 For $Mx \leq 1.5$

F_{ic} : buckling stress

$F_{ic} = S_y$ For $\frac{F_{he}}{S_y} \geq 2.439$

$F_{ic} = 0.7S_y(\frac{F_{he}}{S_y})^{0.4}$ For $0.552 < \frac{F_{he}}{S_y} < 2.439$

$F_{ic} = F_{he}$ For $\frac{F_{he}}{S_y} \leq 0.552$

FS: Design factor

FS = 2.0 For $F_{ic} \leq 0.55\, S_y$

$FS = 2.407 - 0.741(\frac{F_{ic}}{S_y})$ For $0.55S_y < F_{ic} < S_y$

FS = 1.667 For $F_{ic} = S_y$

Finally, the maximum allowable working pressure of the nozzle shall be determined:

In determining the maximum allowable working pressure, reverse the procedures in σ_{avg} and σ_{circ} above, and use the smaller of the pressures obtained.

$$P_{max1} = \frac{S_{allow}}{\frac{2A_p}{A_T} - \frac{R_{xs}}{t_{eff}}}$$

$$P_{max2} = S\left[\frac{t}{R_{xs}}\right]$$

$$P_{max} = min\,[P_{max1}, P_{max2}]$$

Where,

$$A_p = f_{xn}(L_H - t) + f_{xs}(L_R + t_n + R_{nc})$$

Design procedure is depicted in appendix 1.

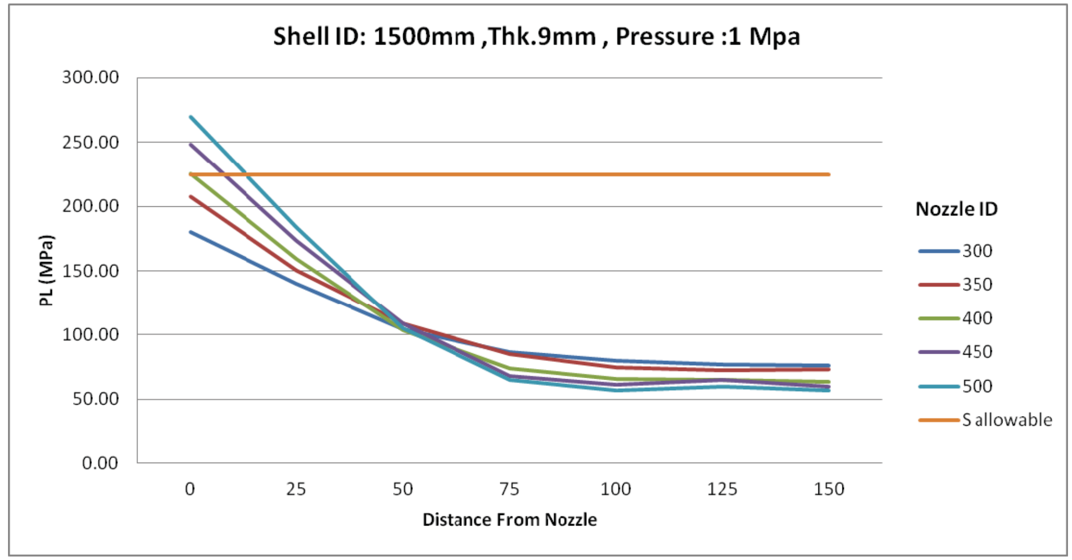
3. SUMMARY OF RESULTS

Afterward, Stress calculated by ASME section VIII DIV.2 for two different shell diameters is compared with the result of finite element analysis and required area of ASME section VIII, DIV.1.

The allowable stress is equal to 1.5SE.

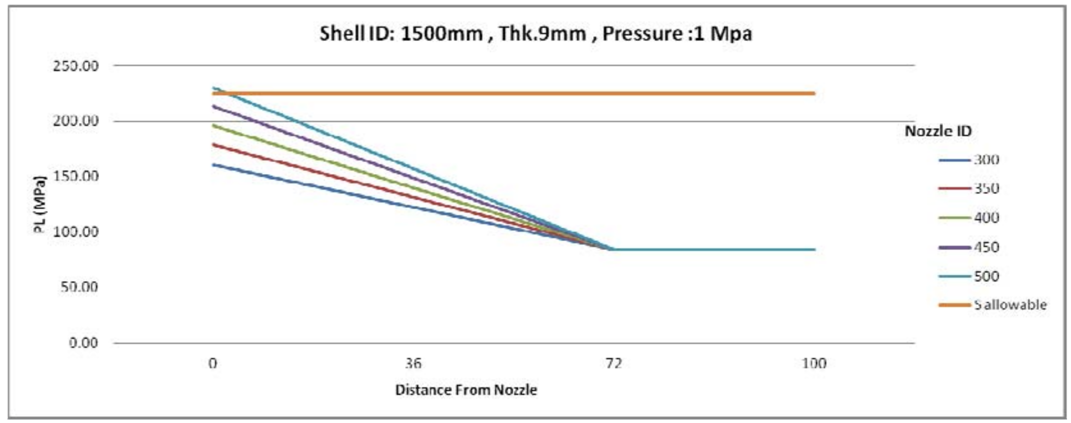
FEA METHOD

distance from nozzle edge	case	DP 1	DP 2	DP 3	DP 4	DP 5	allowable Stress (Mpa)
	Shell ID (mm)	1500	1500	1500	1500	1500	
	NozzleID (mm)	300	350	400	450	500	
0	Membrane Stress (Mpa)	**179.73**	**207.17**	**225.18**	**247.99**	**269.50**	**224.25**
25		140.30	150.40	159.59	173.10	183.38	**224.25**
50		104.71	108.99	103.90	108.86	104.77	**224.25**
75		86.42	85.21	73.77	68.18	65.36	**224.25**
100		79.73	75.00	66.12	61.46	56.78	**224.25**
125		76.96	72.83	64.99	64.94	59.85	**224.25**
150		76.33	73.55	63.51	59.79	56.55	**224.25**
RESULT		pass	pass	fail	fail	fail	



ASME SEC VIII_DIV.II

distance from nozzle edge	case	DP 1	DP 2	DP 3	DP 4	DP 5	Allowable Stress (Mpa)
	Shell ID(mm)	1500	1500	1500	1500	1500	
	Nozzle ID(mm)	300	350	400	450	500	
0	Membrane Stress (Mpa)	**160.90**	**178.70**	**196.10**	**213.20**	**230.00**	**224.25**
36		122.36	131.25	139.98	148.51	156.9	**224.25**
72		83.84	83.84	83.84	83.84	83.84	**224.25**
100		83.84	83.84	83.84	83.84	83.84	**224.25**
RESULT		pass	pass	pass	pass	fail	



ASME SEC VIII_DIV.I

case	DP 1	DP 2	DP 3	DP 4	DP 5
Shell ID(mm)	1500	1500	1500	1500	1500
Nozzle ID(mm)	300	350	400	450	500
Area Available(mm^2)	**1824.1**	**2013.8**	**2205.5**	**2396..2**	**1596.0**
Area Required (mm^2)	1511	1761.8	2015.1	2267.2	1679.6
RESULT	pass	pass	pass	pass	fail

Figure 3.1) STRESS IN SHELL (FEA METHOD, ASME DIV.2) AND AREA CALCULATION DIV.1

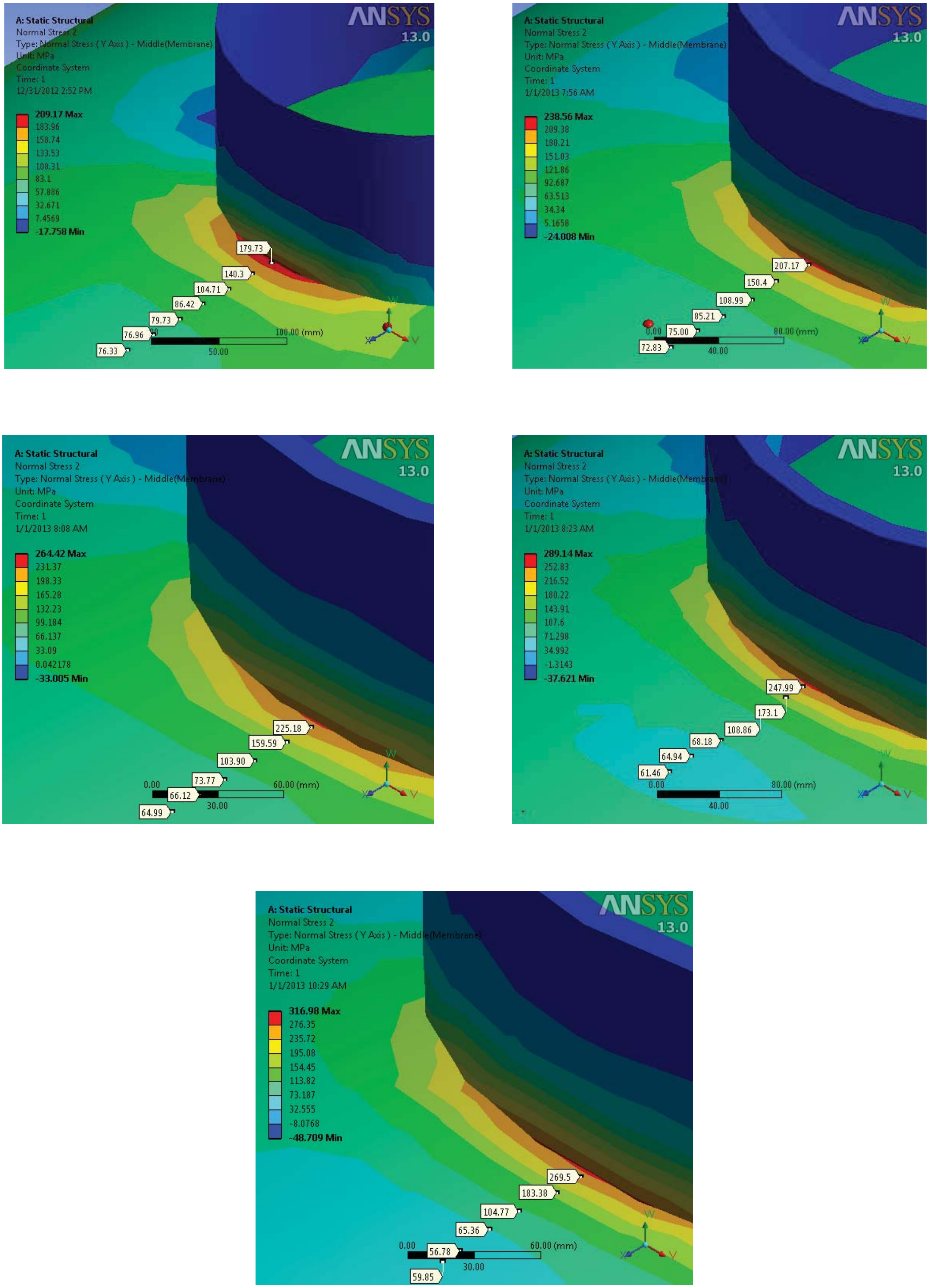
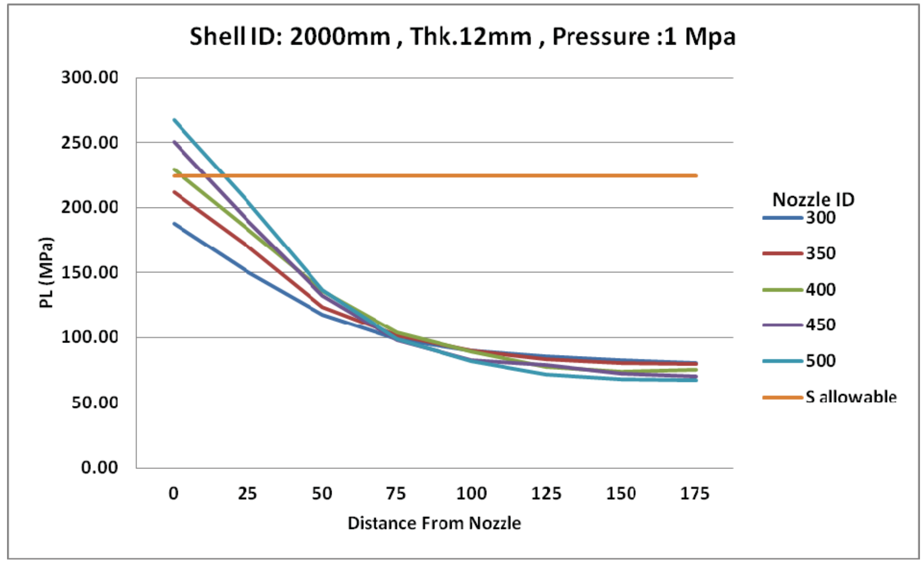


Figure 3.2) STRESS IN SHELL (FEA METHOD)

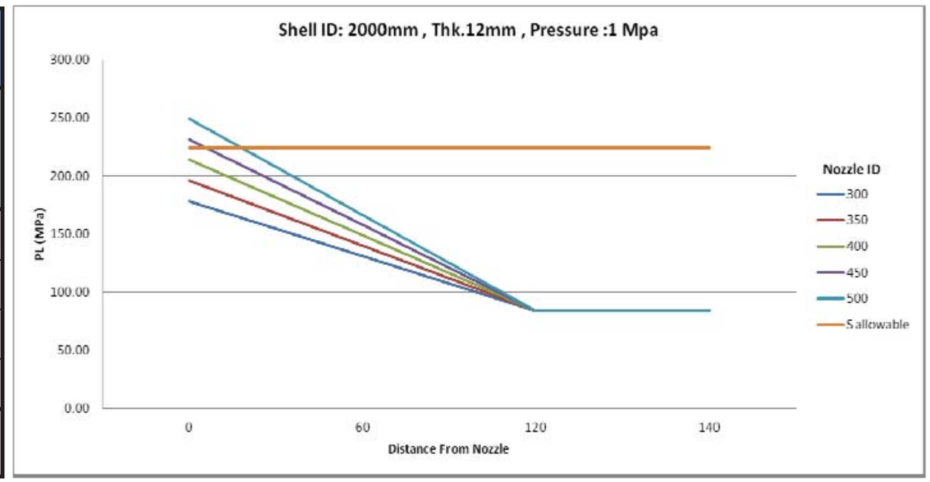
FEA METHOD

distance from nozzle edge	case	DP 1	DP 2	DP 3	DP 4	DP 5	allowable Stress (Mpa)
	Shell ID(mm)	2000	2000	2000	2000	2000	
	Nozzle ID (mm)	300	350	400	450	500	
0	Membrane Stress (Mpa)	**187.70**	**212.26**	**229.46**	**250.69**	**267.20**	**224.25**
25		150.48	170.33	183.63	189.82	204.47	**224.25**
50		117.87	123.85	136.32	132.29	136.99	**224.25**
75		98.98	101.68	103.71	97.89	98.65	**224.25**
100		90.02	89.85	89.56	83.04	81.84	**224.25**
125		85.34	83.11	77.32	79.33	72.09	**224.25**
150		82.58	80.59	74.16	72.65	68.47	**224.25**
175		80.35	79.50	75.08	70.24	67.43	**224.25**
RESULT		pass	pass	fail	fail	fail	



ASME SEC VIII_DIV.II

distance from nozzle edge	case	DP 1	DP 2	DP 3	DP 4	DP 5	allowable Stress (Mpa)
	Shell ID(mm)	2000	2000	2000	2000	2000	
	Nozzle ID(mm)	300	350	400	450	500	
0	Membrane Stress (Mpa)	**177.90**	**196.10**	**214.00**	**231.60**	**249.10**	**224.25**
60		130.86	139.95	148.90	157.74	166.46	**224.25**
120		83.84	83.84	83.84	83.84	83.84	**224.25**
140		83.84	83.84	83.84	83.84	83.84	**224.25**
RESULT		pass	pass	pass	fail	fail	



ASME SEC VIII_DIV.I

case	DP 1	DP 2	DP 3	DP 4	DP 5
Shell ID(mm)	2000	2000	2000	2000	2000
Nozzle ID(mm)	300	350	400	450	500
Area Available(mm^2)	**2204**	**2666.7**	**2921**	**3175.3**	**3429.6**
Area Required (mm^2)	2184.9	2350.8	2686.9	3022.9	3359
RESULT	pass	pass	pass	pass	pass

Figure 3.3) STRESS IN SHELL (FEA METHOD, ASME DIV.2) AND AREA CALCULATION DIV.1

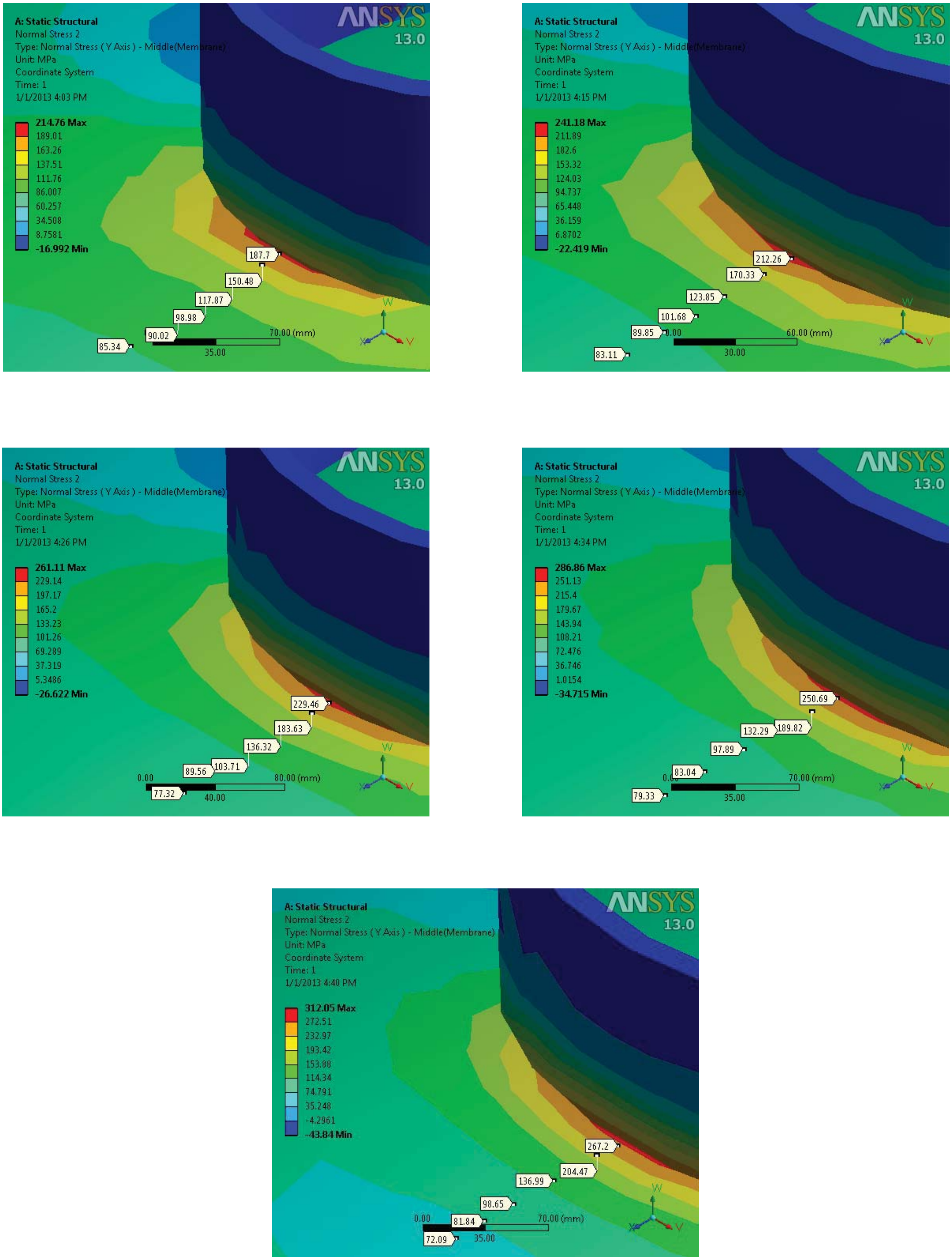
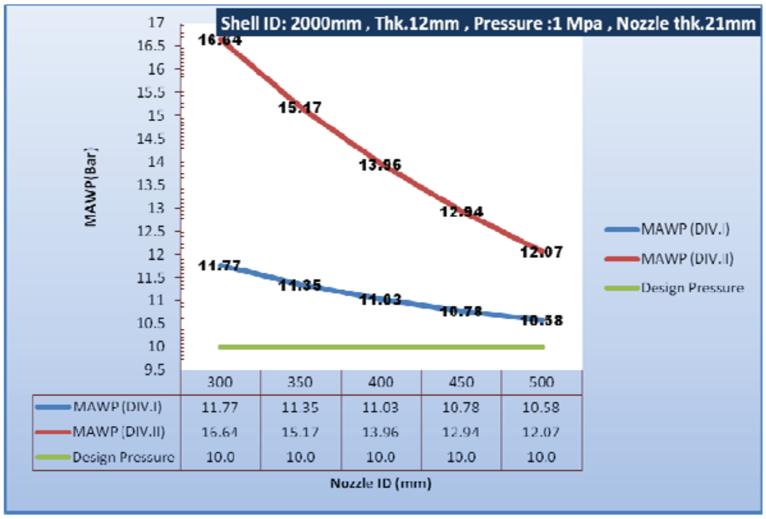
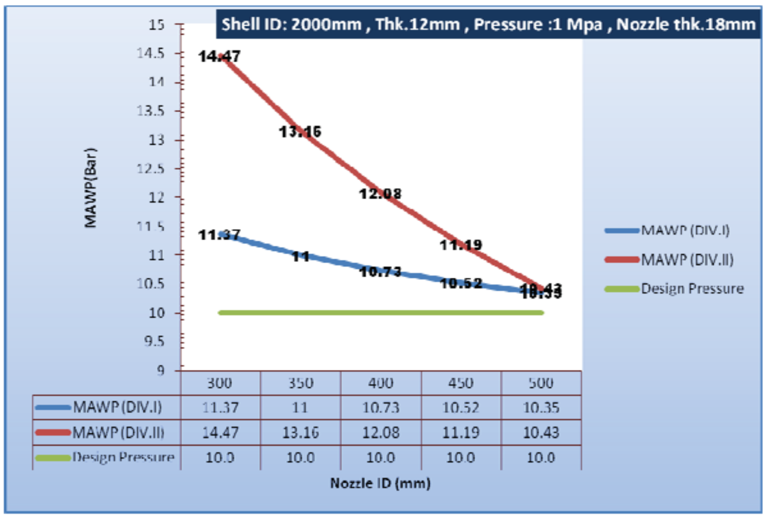


Figure 3.4) STRESS IN SHELL BASED ON FEA METHOD





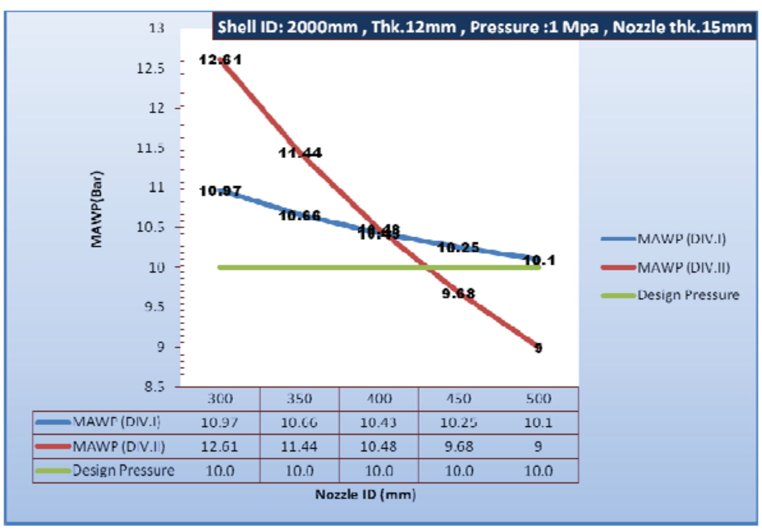
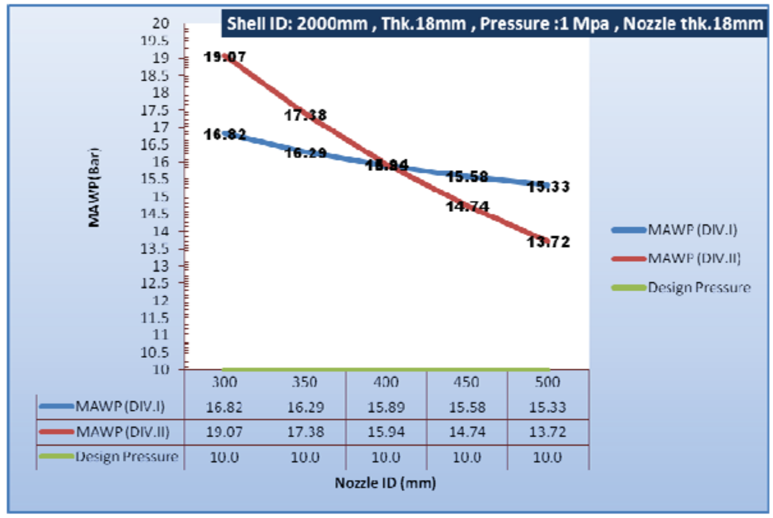
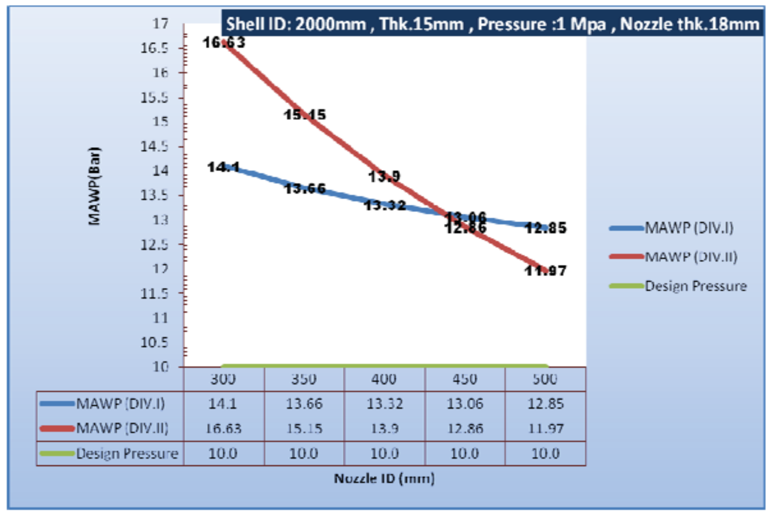


Figure 3.5) MAWP RESULTS (DIV. I via. Div. II)
SHELL ID: FIX, NOZZLE THK.: VARIANT





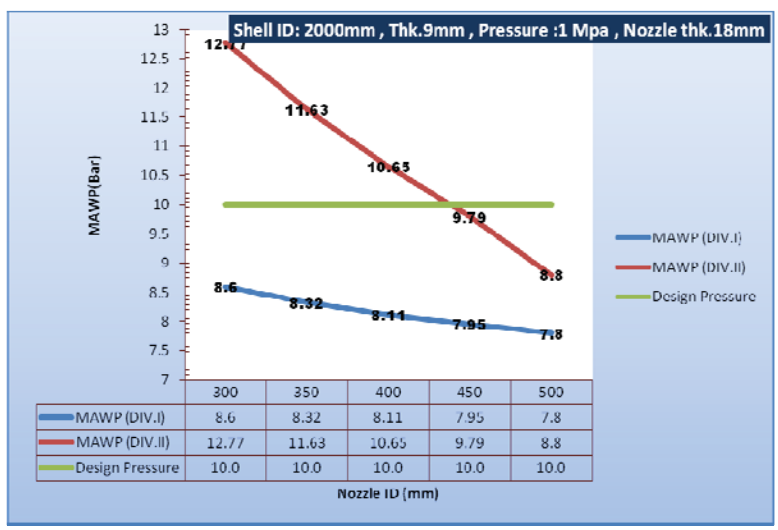
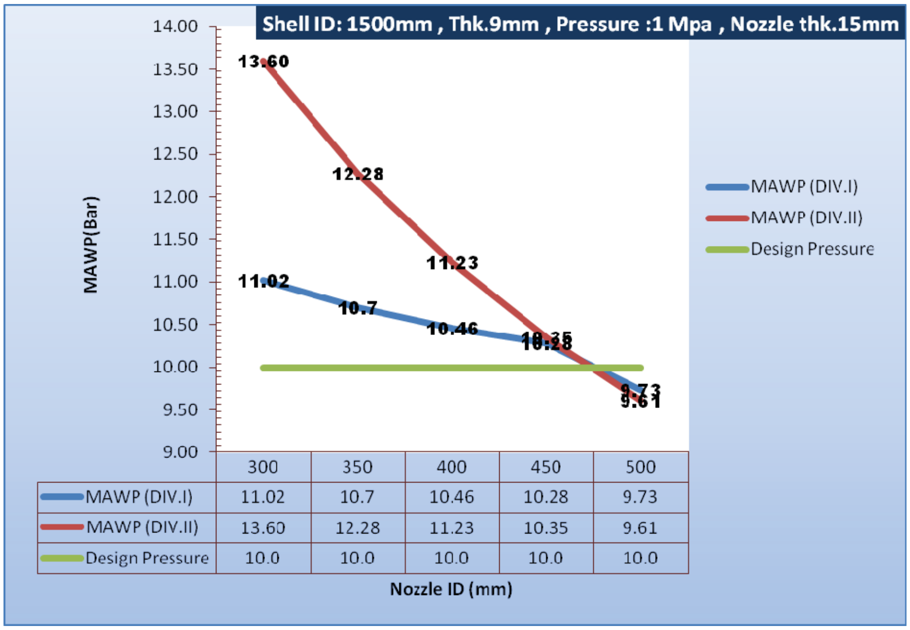
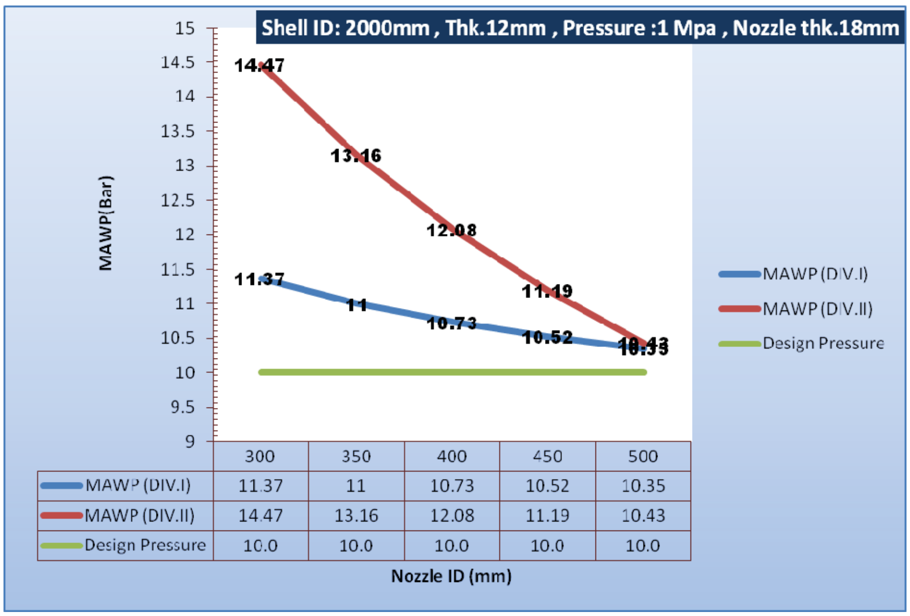


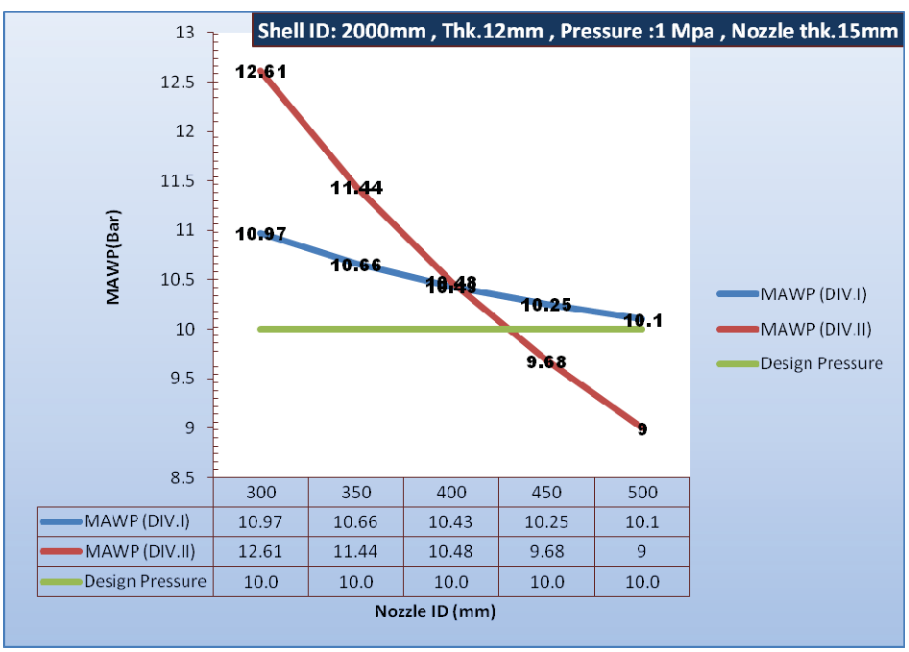
Figure 3.6) MAWP RESULTS (DIV. I via. Div. II)
SHELL ID: VARIANT, NOZZLE THK.: FIX



	300	350	400	450	500
MAWP (DIV.I)	11.02	10.7	10.46	10.28	9.73
MAWP (DIV.II)	13.60	12.28	11.23	10.35	9.61
Design Pressure	10.0	10.0	10.0	10.0	10.0



	300	350	400	450	500
MAWP (DIV.I)	11.37	11	10.73	10.52	10.35
MAWP (DIV.II)	14.47	13.16	12.08	11.19	10.43
Design Pressure	10.0	10.0	10.0	10.0	10.0



	300	350	400	450	500
MAWP (DIV.I)	10.97	10.66	10.43	10.25	10.1
MAWP (DIV.II)	12.61	11.44	10.48	9.68	9
Design Pressure	10.0	10.0	10.0	10.0	10.0

Figure 3.7) MAWP RESULTS (DIV.I via. Div.II) various shell ID and Nozzle Thk.

APPENDIX 1

OPENING DESIGN PROCEDURES

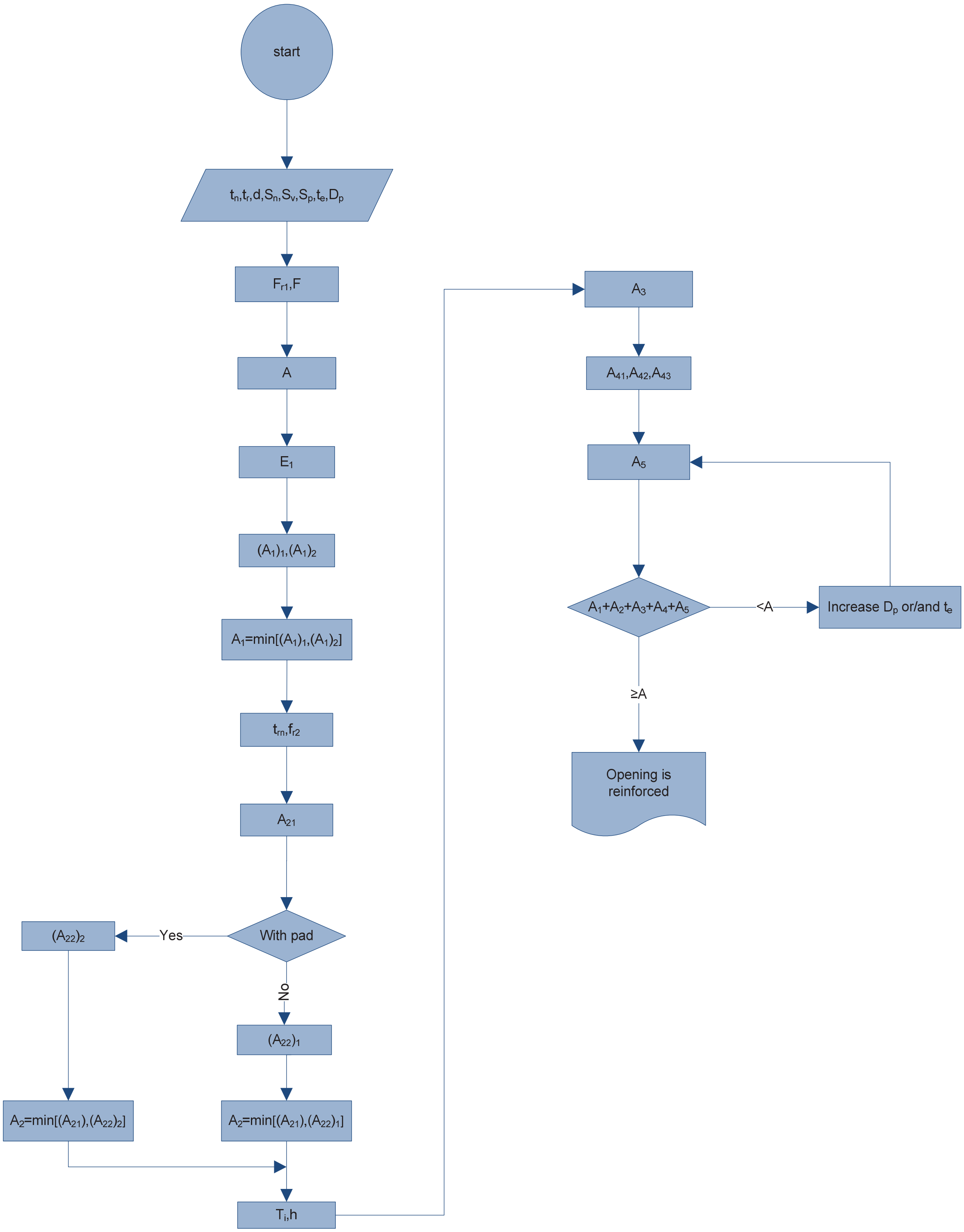


Figure A1.1) CHECKING OPENING AND REINFORCEMENT REQUIREMENT BASED ON ASME SEC VIII DIV.1

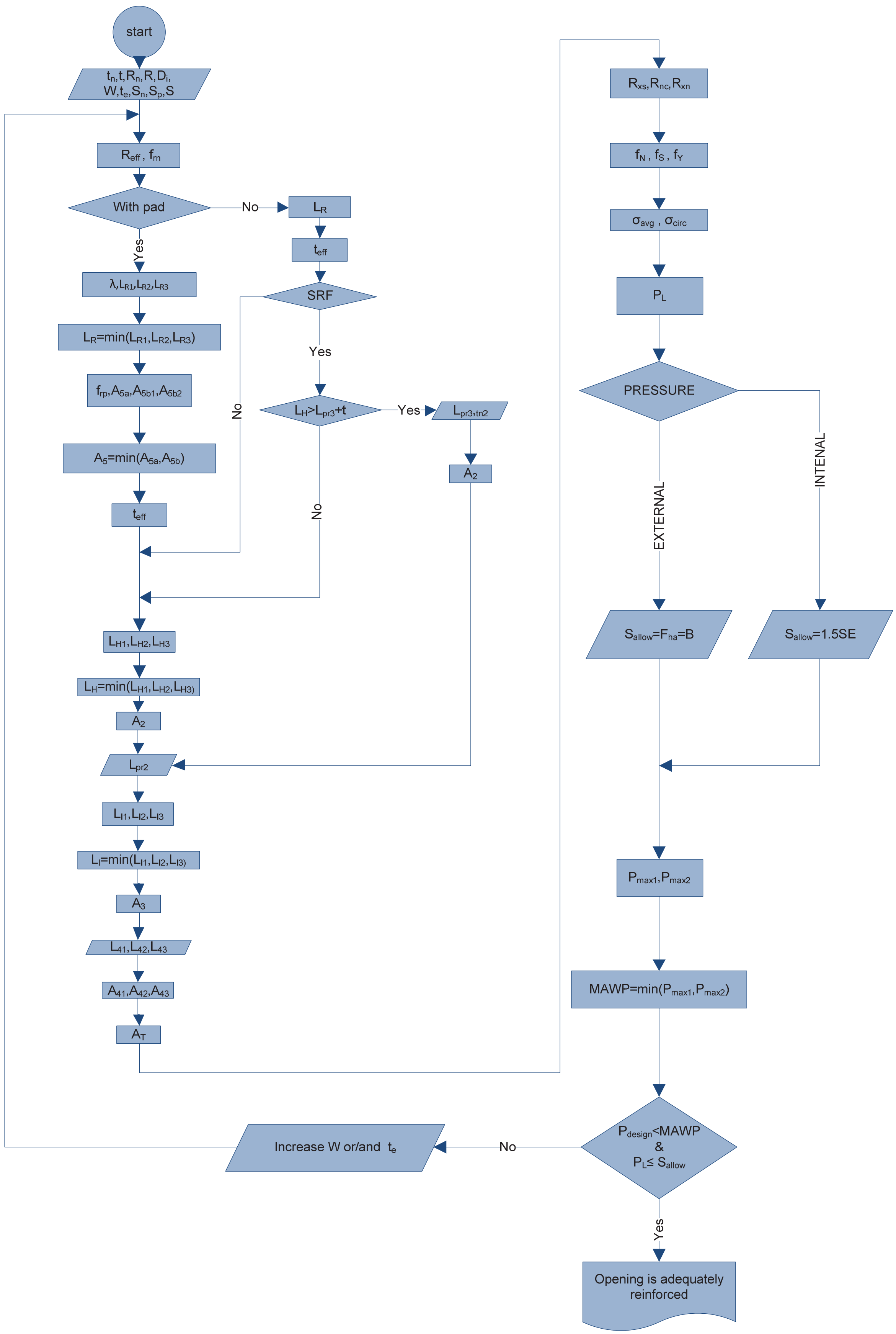


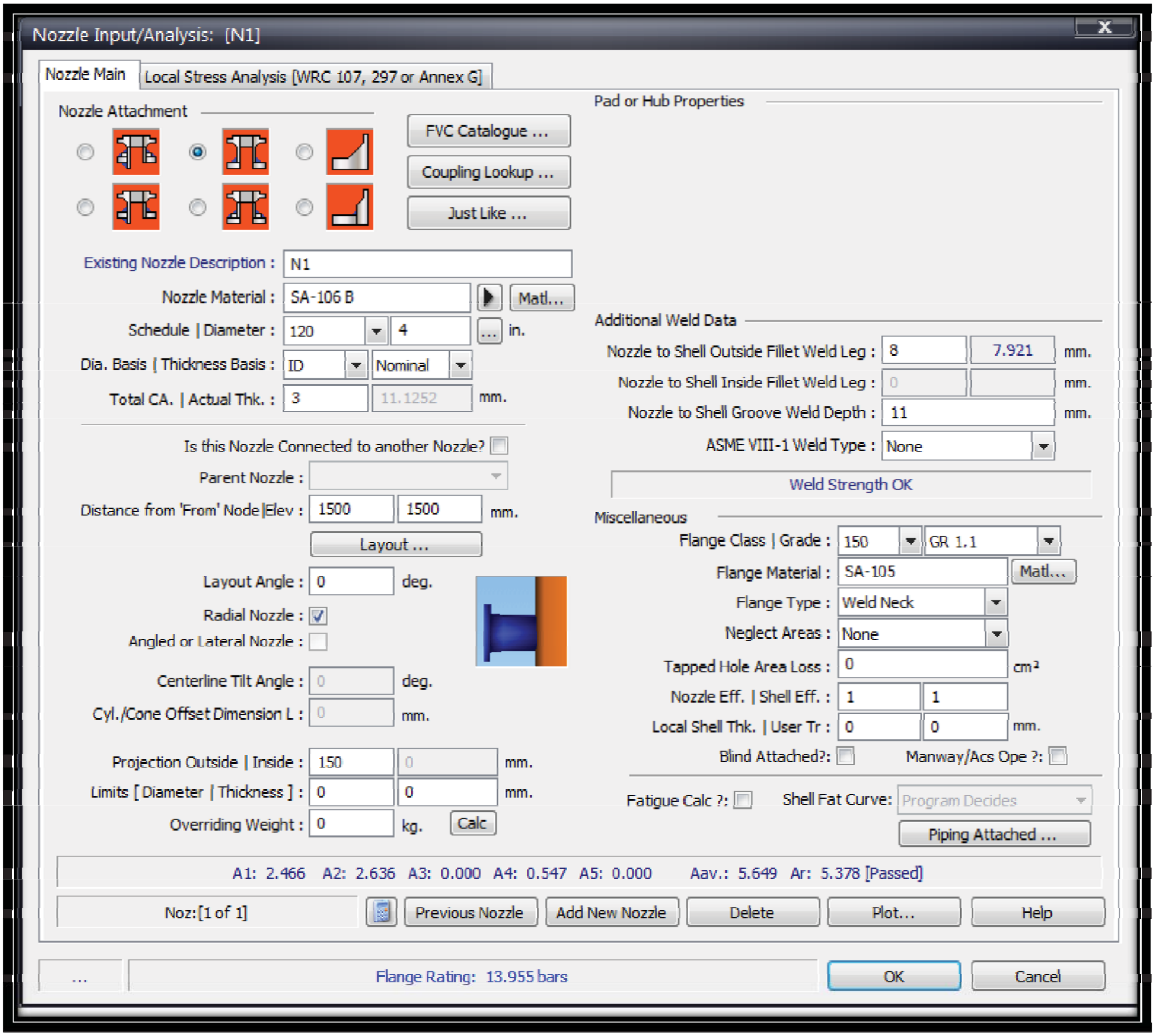
Figure A1.2) CHECKING OPENING AND REINFORCEMENT REQUIREMENT BASED ON ASME SEC VIII DIV.2

APPENDIX 2

EXAMPLE FOR OPENING DESIGN USING PV-ELITE

A 4 in inside diameter nozzle that has 11.13 mm wall thickness, with Material SA-106 B is attached by welding to a vessel that has an inside diameter of 1500mm, thickness11 ,Material SA-516 70.
Check the adequate reinforcement of the opening using DIV.1 AND DIV.2.
Consider: Corrosion allowance: 3mm , Joint efficiency:1

PV-ELITE_ASME SEC.VIII DIV.I



Quick Calculation Results

Weld Strength Reduction Factor [fr1]:
```
 = 1.000
```

Weld Strength Reduction Factor [fr2]:
```
 = min( 1, Sn/S )
 = min( 1, 117.9 /137.9 )
 = 0.855
```

Weld Strength Reduction Factor [fr3]:
```
 = min( fr2, fr4 )
 = min( 0.9 , 1.0 )
 = 0.855
```

Results of Nozzle Reinforcement Area Calculations:

AREA AVAILABLE, A1 to A5		MAWP	External	Mapnc	
Area Required	Ar	5.378	NA	NA	cm²
Area in Shell	A1	2.466	NA	NA	cm²
Area in Nozzle Wall	A2	2.636	NA	NA	cm²
Area in Inward Nozzle	A3	0.000	NA	NA	cm²
Area in Welds	A41+A42+A43	0.547	NA	NA	cm²
Area in Element	A5	0.000	NA	NA	cm²
TOTAL AREA AVAILABLE	Atot	5.649	NA	NA	cm²

The MAWP Case Governs the Analysis.

```
Nozzle Angle Used in Area Calculations                90.00  Degs.
```

The area available without a pad is Sufficient.

Area Required [A]:
```
 = ( d * tr*F + 2 * tn * tr*F * (1-fr1) ) UG-37(c)
 = (98.0496*5.4847*1.0+2*8.1252*5.4847*1.0*(1-1.00))
 = 5.378 cm²
```

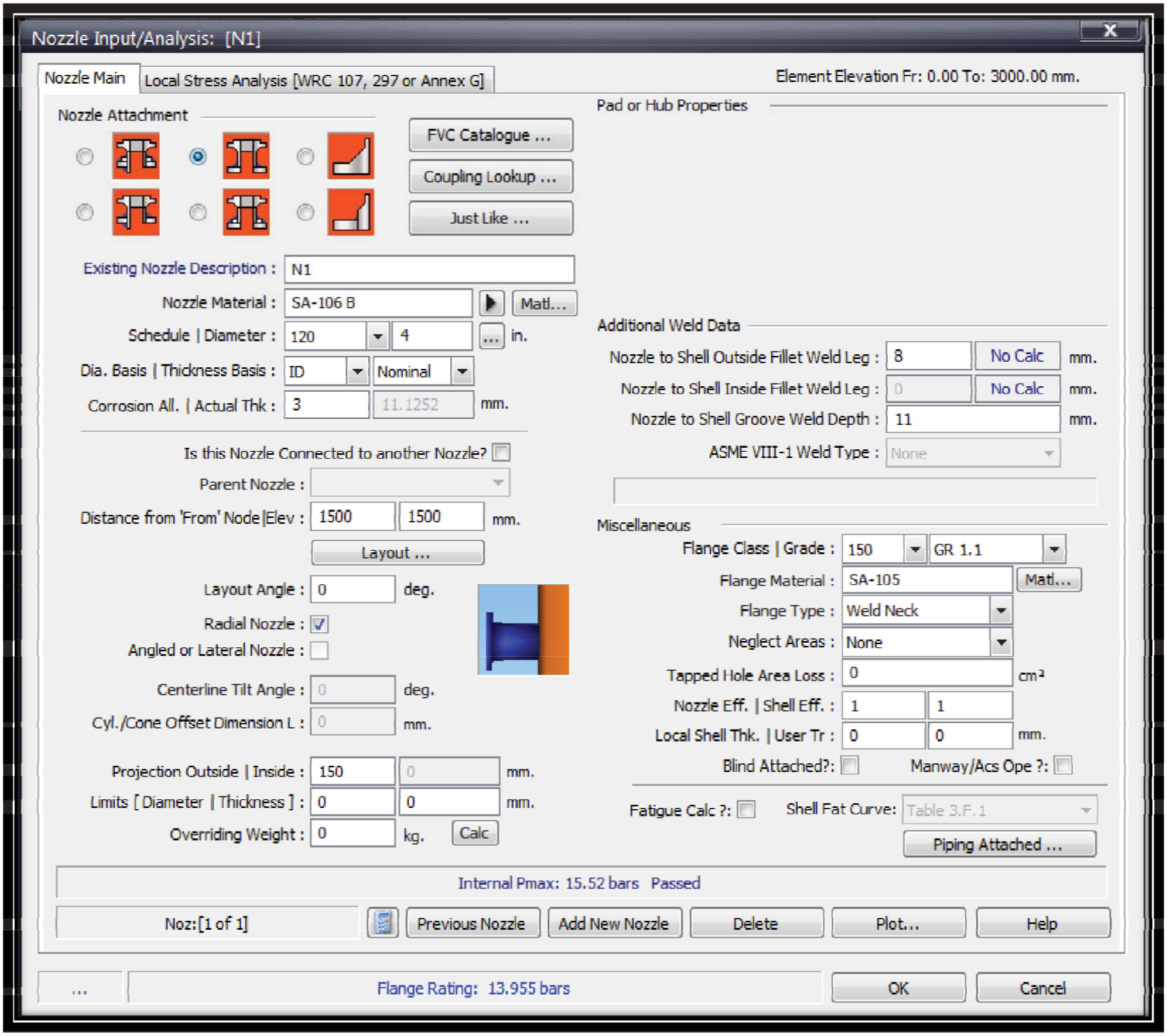
Reinforcement Areas per Figure UG-37.1

Area Available in Shell [A1]:
```
 = d( E1*t - F*tr ) - 2 * tn( E1*t - F*tr ) * ( 1 - fr1 )
 = 98.050 ( 1.00 * 8.0000 - 1.0 * 5.485 ) - 2 * 8.125
   ( 1.00 * 8.0000 - 1.0 * 5.4847 ) * ( 1 - 1.000 )
 = 2.466 cm²
```

Area Available in Nozzle Projecting Outward [A2]:
```
 = ( 2 * tlnp ) * ( tn - trn ) * fr2
 = ( 2 * 20.00 ) * ( 8.13 - 0.42 ) * 0.8550
 = 2.636 cm²
```

Area Available in Inward Weld + Outward Weld [A41 + A43]:
```
 = Wo² * fr2 + ( Wi-can/0.707 )² * fr2
 = 8.0000² * 0.8550 + ( 0.0000 )² * 0.8550
 = 0.547 cm²
```

PV-ELITE ASME SEC.VIII DIV.II



Quick Calculation Results

***Nozzle Calculations per Section 4.5:* Internal Pressure Case:**

Nozzle Material Factor [frn]:
```
 = Sn/S
 = 137.7 / 149.5
 = 0.921
```

Thickness of Nozzle [tn]:
```
 = thickness - corrosion allowance
 = 11.125 - 3.000
 = 8.125 mm.
```

Shell Diameter to Thickness ratio [D/t]:
```
 = Di/t
 = 1506.000 /8.000
 = 188.250  must be less than 400.
```

Effective Pressure Radius [Reff]:
```
 = Di/2 + corrosion allowance
 = 1500.000 /2 + 3.000
 = 753.000 mm.
```

Effective Length of Vessel Wall [LR]:
```
 = min( sqrt( Reff * t ), 2 * Rn )
 = min( sqrt( 753.000 * 8.000 ), 2 * 49.025 )
 = 77.614 mm.
```

Thickness Limit Candidate [LH1]:
```
 = t + te + sqrt( Rn * tn )
 = 8.000 + 0.000 + sqrt( 49.025 * 8.125 )
 = 27.958 mm.
```

Thickness Limit Candidate [LH2]:
```
 = Lpr1
 = 150.000
 = 150.000 mm.
```

Thickness Limit Candidate [LH3]:
```
 = 8 * ( t + te )
 = 8 * ( 8.000 + 0.000 )
 = 64.000 mm.
```

Effective Nozzle Wall Length Outside the Vessel [LH]:
```
 = min[ LH1, LH2, LH3 ]
 = min[ 27.958 , 150.000 , 64.000 )
 = 27.958 mm.
```

Effective Vessel Thickness [teff]:
```
 = t * ( ( t * LR + A5 * frp ) / ( t * LR ) )
 = 8.000*((8.000*77.614+0.000*1.000)/(8.000*77.614))
 = 8.000 mm.
```

Determine Parameter [Lamda]:
```
 = min( 12, ( dn + tn )/( sqrt( ( Di + teff ) * teff )) )
 = min( 12, (98.05 + 8.125 )/( sqrt((1506.00 + 8.000 ) * 8.000 )) )
 = 0.965
```

Compute Areas A1-A43 (No Pad) or A1-A5 (With Pad) :

Area Contributed by the Vessel Wall [A1]:

```
= t * LR * max( (Lamda/5)^(0.85), 1 )
= 8.000 * 77.614 * max( (0.965 /5)^(0.85), 1 )
= 6.209 cm²
```

Area Contributed by the Nozzle Outside the Vessel Wall [A2]:

```
= tn * LH
= 8.125 * 27.958
= 2.272 cm²
```

Area Contributed by the Outside Fillet Weld [A41]:

```
= 0.5 * Leg41^(2)
= 0.5 * 8.000^(2)
= 0.320 cm²
```

The total area contributed by A1 through A43 [AT]:

```
= A1 + frn( A2 + A3 ) + A41 + A42 + A43
= 6.209+0.921(2.272+0.000)+0.320+0.000+0.000
= 8.622 cm²
```

Nozzle Radius for Force Calculation [Rxn]:

```
= tn / ln[ ( Rn + tn )/Rn ]
= 8.125 / ln[ ( 49.025 + 8.125 )/49.025 ]
= 52.984 mm.
```

Shell Radius for Force Calculation [Rxs]:

```
= teff / ln[ ( Reff + teff )/Reff ]
= 8.000 / ln[ ( 753.000 + 8.000 )/753.000 ]
= 756.993 mm.
```

Allowable Local Primary Membrane Stress [Sallow]:

```
= 1.5 * S * E
= 1.5 * 149.500 * 1.000
= 224.2 N./mm²
```

Determine Force acting on the Nozzle [fN]:

```
= P * Rxn * ( LH - t )
= 10.000 * 52.984 * ( 27.958 - 8.000 )
= 107.8 Kgf
```

Determine Force acting on the Shell [fS]:

```
= P * Rxs * ( LR + tn )
= 10.000 * 756.993 * ( 77.614 + 8.125 )
= 6618.7 Kgf
```

Discontinuity Force from Internal Pressure [fY]:

```
= P * Rxs * Rnc
= 10.000 * 756.993 * 49.025
= 3784.5 Kgf
```

Area Resisting Internal Pressure [Ap]:

```
= Rxn( LH - t ) + Rxs( LR + tn + Rnc )
= 52.984 ( 27.958 - 8.000 ) + 756.993 ( 77.614 + 8.125 + 49.025 )
= 1030.7 cm²
```

Maximum Allowable Working Pressure Candidate [Pmax1]:

```
 = Sallow /( 2 * Ap/AT - Rxs/teff )
 = 224.250 /( 2 * 1030.732 /8.622 - 756.993 /8.000 )
 = 15.5 bars
```

Maximum Allowable Working Pressure Candidate [Pmax2]:

```
 = S[t/Rxs]
 = 149.500 [8.000 /756.993 ]
 = 15.8 bars
```

Maximum Allowable Working Pressure [Pmax]:

```
 = min( Pmax1, Pmax2 )
 = min( 15.520 , 15.798 )
 = 15.520 bars
```

Average Primary Membrane Stress [SigmaAvg]:

```
 = ( fN + fS + fY ) / AT
 = ( 107.836 + 6618.659 + 3784.462 ) / 8.622
 = 119.560 N./mm²
```

General Primary Membrane Stress [SigmaCirc]:

```
 = P * Rxs / teff
 = 10.000 * 756.993 / 8.000
 = 94.6 N./mm²
```

Maximum Local Primary Membrane Stress [PL]:

```
 = max( 2 * SigmaAvg - SigmaCirc, SigmaCirc )
 = max( 2 * 119.560 - 94.630 , 94.630 )
 = 144.5 N./mm²
```

Summary of Nozzle Pressure/Stress Results:

Allowed Local Primary Membrane Stress	Sallow	224.25	N./mm²
Local Primary Membrane Stress	PL	144.49	N./mm²
Maximum Allowable Working Pressure	Pmax	15.52	bars