DESIGN OF NOZZLES IN REACTOR CONTAINMENTS

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SUMMARY

Cylindrical nozzles in pressurized spherical steel reactor containments are usually designed on the basis of different pressure vessel codes. Hoop and bending stresses at the junction of both shells may be reduced by locally increasing the thickness of the cylinder and/or the sphere. Using a thin sell idealization the bending solution usually overestimates the real stress state at the joint.

In this paper different elastic analyses are performed on protruding radial cylindrical nozzles in a PWR containment with a diameter of 56 m and an effective thickness of 2.9 cm. Two studies are discussed in detail. In a first analysis a thin shell representation of the structure is used. Geometry and stiffness of both structural elements are varied. The thickness and length of the nozzle as well as the radius and thickness of a pad reinforcement in the spherical shell are taken as parameters. It has been found that the stress state in each component is primarily influenced by its own stiffness or being more specific: An increase of the thickness of the cylindrical shell causes a considerable reduction of hoop and bending stresses in the nozzle but has little effect on the spherical shell. There the corresponding stresses can essentially be reduced through increasing the thickness of the pad which on the other hand has only minor influence on the stress state of the nozzle. Therefore, it is recommended to distribute the additional reinforcement (material) on both structural elements. This results in an increase of the thickness of the pad by 50 percent and of the nozzle by 150 percent compared to the spherical shell. Furthermore, the size of the pad should be about 1.5 to 2.0 times that of the nozzle radius.

In the second study the joint zone of a nozzle in a pad—reinforced pressure vessel is idealized as a thick structure. A three-dimensional finite element analysis is carried out producing a more realistic stress representation of the intersection. The investigation shows that the thin shell analysis gives a satisfactory picture of the membrane stresses but overstates the bending stresses up to 20 percent.

The results of both studies allow an improved and more economic design of cylindrical nozzles in spherical reactor containments.

1. Introduction and Scope of the Study

The design of cylindrical nozzles in pressurized spherical reactor containments is normally based on the requirements of different pressure vessel codes. These usually consist of more or less simplified routine calculation procedures which do not give much insight into the response of the structure. It is in particular not conclusive whether the containment or the nozzle should be reinforced. Hoop and bending stresses at the junction of both shells may be reduced by increasing the thickness of the cylinder and/or the sphere locally. The purpose of this paper is to aid in the understanding of the described problem. Both analytical and numerical solution procedures for the analysis of the structure considered are common in the classical engineering calculations and therefore need not be elaborated on, see for example [1], [2], [3].

In the paper different elastic analyses are performed on protruding radial cylindrical nozzles in a spherical PWR containment with a diameter of R = 56 m and an effective thickness of t_s = 2.9 cm. A typical nozzle with a radius r_n = 75.0 cm has been selected for a parametric study. Figure 1 shows the geometry of the pad reinforced structure under consideration. Uniform thickness is assumed for the nozzle as well as the pad to minimize the manufacturing cost. Stress concentrations at the pad-sphere intersection are avoided by continously increasing the thickness to the final value of the reinforcement.

Geometry and stiffness of both structural elements are varied using the following parameters:

the thickness of the nozzle	$3.0 \text{ cm} \leq t_n \leq 9.0 \text{ cm}$
the thickness of the pad	$2.9 \text{ cm} \leq t_p \leq 6.0 \text{ cm}$
the pad size	$1.0\, r_n \leq r_p \leq 2.5\, r_n$
the nozzle length	$0.4\, r_n \leq \ell_n \leq 0.8\, r_n$

Although the absolute magnitude of the results in this parameter study cannot be transferred to nozzles of another size, certain tendencies may be generalized. The main differences can be seen in the loading conditions of different nozzle types. The following load cases are studied in the analyses with the weight of the structure ignored:

① pressure load p (assuming a flat end closure of the nozzle)
② thrust load $\pm$ T applied to the exterior part of the nozzle
③ bending moment M applied to the exterior part of the nozzle

The last two load cases lead to axial forces n in the cylindrical shell:

$$n^{②} = \frac{T}{2\pi r_n} \qquad \text{(axisymmetric m = 0)}$$

$$n^{③} = \frac{M}{\pi r_n^2} \cos\vartheta \qquad \text{(asymmetric m = 1)}$$

It is assumed that there is no constraint caused by the flat plate end closure of the cylinder.

The parametric study is performed using a finite element analysis for shells of revolution. The program is based on the idealization of the structure as a thin shell [4]. The accuracy of test runs was satisfactorily in comparison to the analytical solution. The analysis is confined to a small segment of the containment which permits sufficient decay of the edge disturbance. In Figure 2 the normalized membrane forces and bending moments are plotted. The functions of the non-reinforced structure (t_n = 3.0 cm, t_p = 2.9 cm) for the three different load cases are shown.

For the further investigation the stresses at the outer and inner surface of the spherical and the cylindrical shell at the intersection point are evaluated. The treatment of the transition zone of the unreinforced spherical shell/pad is skipped. The equivalent stress σ_e based on the hypothesis of the strain energy of distortion is taken as the mean value which has to meet the allowable stress limits.

Section 2 describes the influence of the thickness variation of nozzle and pad. In section 3 the effect of the nozzle length and the pad size on the stress state is discussed. In section 4 the parametric study is carried out for a specific load combination.

It is known that a thin shell analysis does not produce the right picture of the stress distribution in the region of the node. Usually the bending solution overestimates the real stress state. Furthermore the rounded weld fillet is not at all represented. In order to study the influence of the thin shell idealization a three-dimensional finite element analysis of the joint zone is performed. This investigation is discussed in section 5.

2. Thickness-Variation of Nozzle and Pad

In a first parametric study the thicknesses of the nozzle and the pad are varied within the above given ranges. Pad size and nozzle length are taken as $r_p = 2.0\, r_n$ and ℓ_n = 50 cm, respectively. The influence of the thickness variation on the stress state of the pressurized structure is shown in Figure 3 a (nozzle) and Figure 3 b (sphere). In these graphs the normalized equivalent stresses at the external and internal surface are plotted versus the nozzle thickness t_n. The pad thickness t_p is taken as an additional parameter. From Figure 3 a it is evident that an increase of the thickness of either the nozzle or the pad results in an appreciable reduction of the nozzle stresses. But it may also be seen that the stress state in the nozzle is much stronger influenced by a stiffness change of the cylindrical shell than that of the sphere. The contrary follows from Figure 3 b for the stresses in the pad, however a smaller influence of the thickness variation may be observed.

Figures 4 and 5 show the corresponding plots of the normalized equivalent stresses due to the loading cases thrust and moment, respectively. Apart from absolute values both loading conditions indicate similar tendencies: The nozzle stresses are essentially reduced by an increase of the nozzle thickness. Doubling the thickness results in more than 50 percent

decrease of the equivalent stress. A variation of the pad stiffness has only little influence on the cylindrical shell but affects primarily the stress state in the containment.

3. Variation of Nozzle Length and Pad Size

The parametric study described in section 2 is repeated varying the length ℓ_n of the nozzle and the pad size r_p.

First of all it can be stated that the variation of the nozzle length has only a minor influence on the stress state at the juncture of both shells. Even for the extreme case of $\ell_n = 0.4\, r_n = 30$ cm and a nozzle thickness of $t_n = 9$ cm (with a half-period of disturbance $> 2\,\ell_n$) the increase of the equivalent stress is less than 6 percent compared to the corresponding structure with $\ell_n = 0.67\, r_n = 50$ cm. A further reduction of the nozzle length may be considered depending on the structural requirements.

The parameter "pad size" shows a similar behavior. In Figure 6 an example with a small nozzle thickness of $t_n = 3.0$ cm ($\ell_n = 50$ cm) has been selected in order to demonstrate the influence of the pad size. In this figure the equivalent stress of pad and nozzle are plotted versus the pad radius r_p taking the pad thickness t_p as parameter. It is illustrated that even a small pad has already a considerable effect on the stress state. Furthermore the graph shows that a pad size r_p larger than 1.5 to 2.0 times the nozzle radius r_n has very little additional influence. A similar result is stated in [2]. On the other hand it has to be noted that the edge disturbance should not influence too much the transition zone between the pad and the unreinforced shell. So the above given values seem to be reasonable for the size of the pad reinforcement.

4. Example for a Prescribed Load Combination

Let the following combinations of the single load cases be chosen:

pressure load	p	$= 48\ \mathrm{N/cm^2}$
and thrust	T	$= \pm 275$ kN
or moment	M	$= 250$ kNm

The parametric study of section 2 is repeated for the given load conditions ($\ell_n = 50$ cm, $r_p = 2\, r_n = 150$ cm). Figure 7 illustrates the influence of the thickness variation for the nozzle and the sphere. Only the maximum values of the equivalent stresses are included. The steel selected is a grain refined steel St E 47 with a guaranteed high temperature yield strength $\sigma_{0.2}$ of 403 N/mm² for a thickness $t \leq 50$ mm or 390 N/mm² for 50 mm $< t <$ 110 mm, respectively. Since the stresses are caused by a gross structural discontinuity they have to meet the following requirements:

$0.9 \cdot \sigma_{0.2}$	for primary and secondary membrane and bending stresses
$0.75 \cdot \sigma_{0.2}$	for primary local membrane stresses

Adding these limits to the graphs in Figure 7 (σ_a) the following thicknesses may be

selected for the design of the pad reinforced nozzle (marked ○):

t_n = 7.5 cm
t_p = 4.5 cm

As the pad in this case is not utilized in an optimal way it might be worthwhile to choose the thicknesses (marked ●):

t_n = 8.0 cm
t_p = 3.5 cm

In a separate study it has been proven that the primary local membrane stresses do not exceed the allowable limits.

5. Idealization as a Thick Shell

The pad reinforced nozzle with t_p = 3.8 cm, t_n = 7.0 cm, r_p = 165 cm, r_n = 74.5 cm, ℓ_n = 50 cm has been selected for a refined thick shell analysis of the pressure load case. The program SAP [5] is used to perform an elastic three-dimensional finite element analysis. A quadrilateral and a triangular axisymmetric element has been chosen to analyse the joint zone of the structure. The rounded fillet has been added to the finite element model. The previous study has shown that the stresses at the inside surface of the nozzle dominate the dimensioning of the structure. Therefore this part of the nozzle has been selected in order to demonstrate the difference of either idealization. Figure 8 shows the functions of the axial and hoop stresses plotted over the length of the cylinder. The thin shell idealization represents the membrane stresses satisfactorily but overstates the axial stresses at the intersection of both shells. This results also in an increase of the circumferential bending stresses. As an average value the corresponding equivalent stress of the three-dimensional analysis is reduced to about 80 %. It has been proposed in a similar study [6] that the stress values at the edge of the joint zone at the beginning of the rounded fillet - increased by a factor of 10 percent - may be taken as design variables. In this investigation the factor turned out to be nearly 20 percent. However it should be mentioned that the stress state at the junction is very much influenced by the physical as well as the computational model of the rounded fillet zone so that stress concentrations will be avoided.

6. Conclusions

The parametric study for a protruding cylindrical nozzle in a spherical PWR containment presented in this paper may lead to the following conclusions:

* For the pressure load case a thickness increase of either the nozzle or the pad results in a considerable reduction of the stresses in both structural elements.
* For the thrust and moment load case a thickness variation of the nozzle influences primarily the cylindrical shell and not the spherical shell and vice versa.
* Nozzle length and pad size have a minor influence on the stress state of the structure.

* For the combined load case (pressure and thrust or moment) the additional reinforcement (material) should be distributed on both structural elements.
* The thin shell idealization produces a fairly good representation of the stress state in the joint zone. A reduction of the equivalent stresses at the intersection point of up to 20 percent may be considered.

It should be mentioned that this result cannot be extrapolated to other loading conditions or different geometries.

References

[1] KRAUS, H., "Thin Elastic Shells", J. Wiley and Sons, New York 1967.

[2] GILL, S.S., LECKIE, F.A., PENNY, R.K., "Pressure Vessel Branches" in "The Stress Analysis of Pressure Vessels and Pressure Vessel Components", Pergamon Press, 1970.

[3] KRAUS, H. (editor), "The Computer in Pressure Vessel Analysis", Proc. of a ASME Computing Seminar, Dallas, Texas 1968, ASME, United Engng. Center, New York.

[4] ZIENKIEWICZ, O.C., "The Finite Element Method in Engineering Science", Mc Graw-Hill, London 1971.

[5] WILSON, E.L., "SAP - A Structural Analysis Program", Report UC SESM 70-20, Department of Civil Engineering, University of California, Berkeley 1970.

[6] CECH, F., FISCHER, D., ZWINGENBERGER, L., "Neueste Erfahrungen bei der Berechnung und Konstruktion von Sicherheitsbehältern aus der Sicht des Herstellers", 2nd Int. Conf. Struct. Mech. Reactor Techn., Berlin 1973, Paper J 2/2.

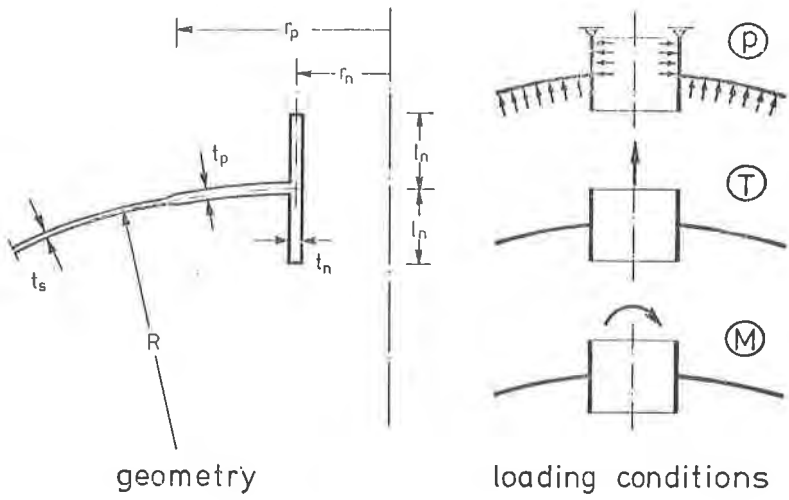

Figure 1: Scheme of Nozzle and Vessel Geometry and Loading Conditions

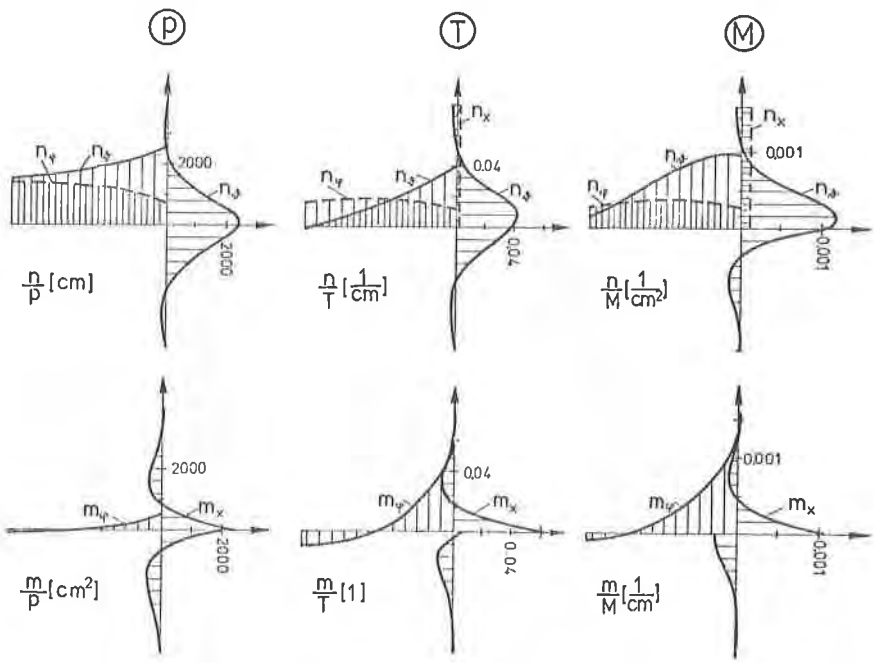

Figure 2: Membrane Forces and Bending Moments

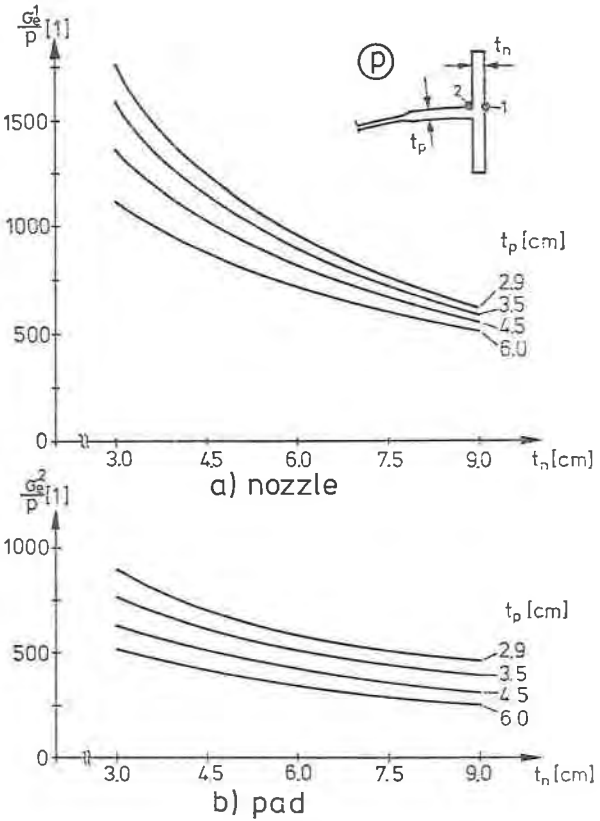

Figure 3: Equivalent Stress σ_e - Load Case p

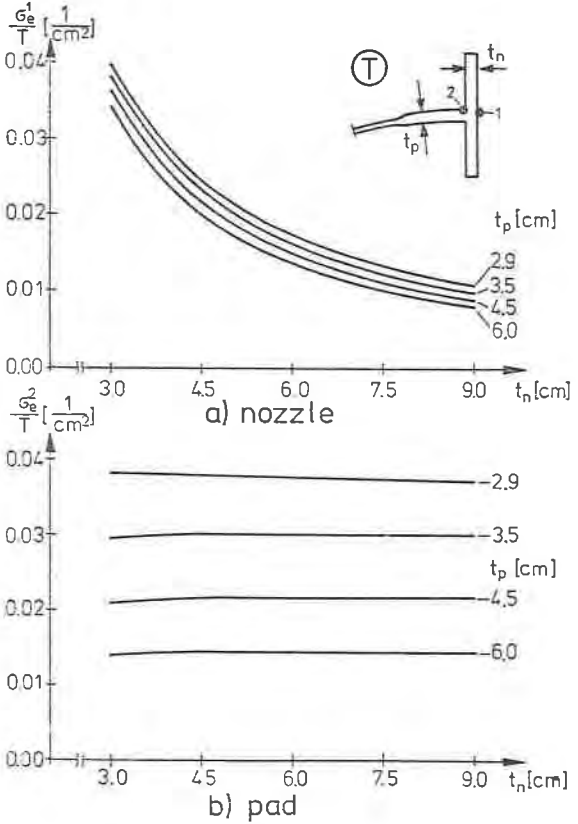

Figure 4: Equivalent Stress σ_e - Load Case T

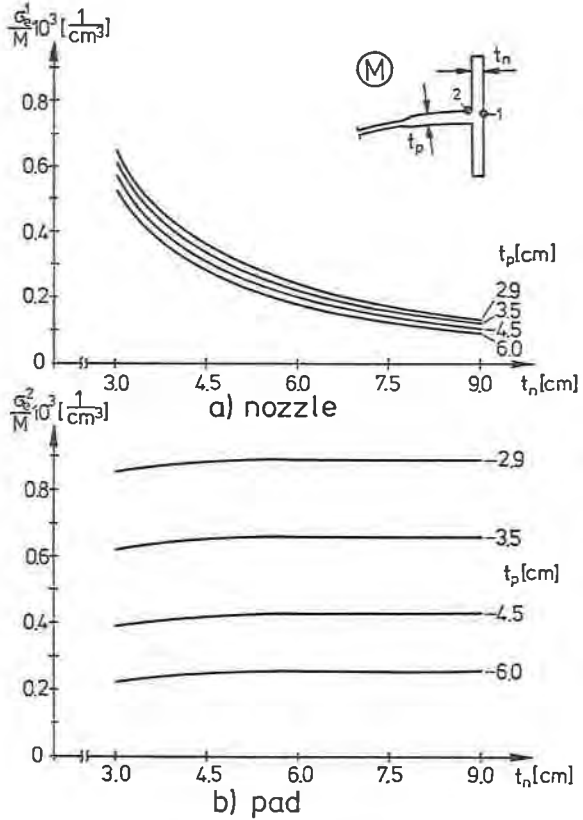

Figure 5: Equivalent Stress σ_e - Load Case M

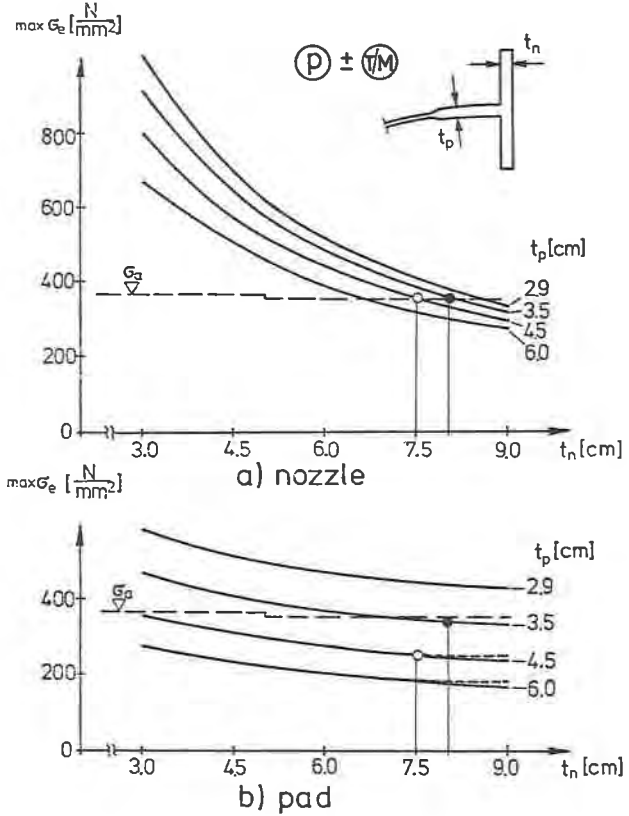

Figure 7: Equivalent Stress σ_e - Combined Load Case p/T/M

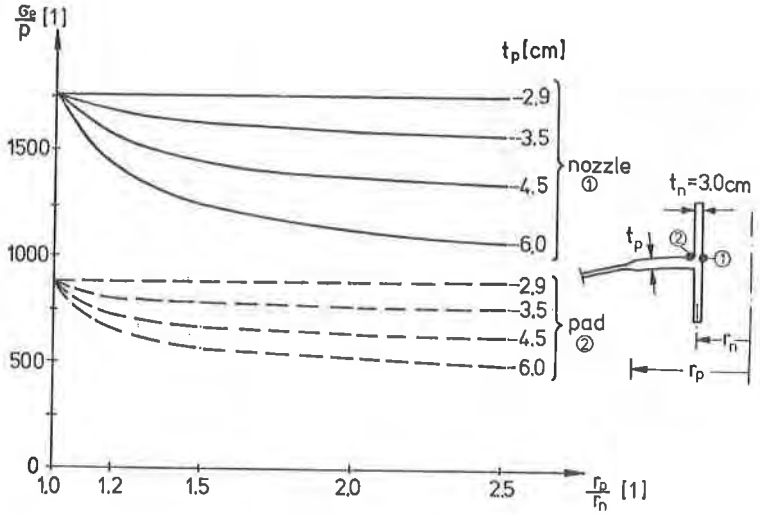

Figure 6: Variation of the Pad Size: Load Case p

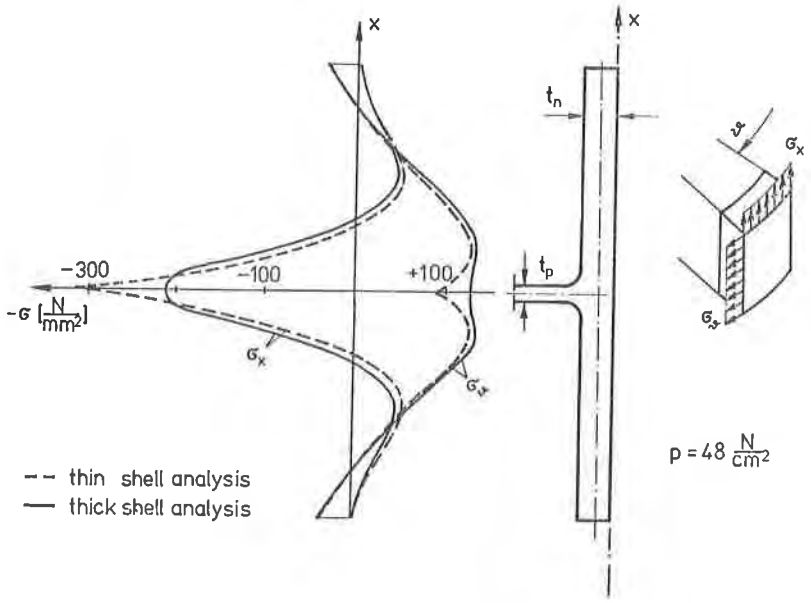

Figure 8: Thick Shell versus Thin Shell Idealization: Axial and Hoop Stresses