INFLUENCE OF SKIRT SUPPORT ON THE STRESSES IN LARGE VESSELS WITH A CONICAL BOTTOM

By

P. Reuss

Department of Chemical Machines and Agricultural Industries, Technical University, Budapest

(Received March 28, 1974)

Presented by Prof. Dr. S. Szentgyörgyi

Introduction

For the strength analysis of large-size thin-walled, conical bottomed containers and technological vessels of industrial chemistry no suitable standard formulae are available. Such vessels lie outside the usual size range of pressure vessels, thus the experimentally determined shape factors for conical bottoms without rounding off the corners are not valid for the formers. The outline drawing of a technological vessel with conical bottom, which can be said typical, is seen in Fig. 3.

The method presented in the following is based on the theory of thin shells and permits, on account of introducing the shape factor, a relatively simple design.

1. Investigation of the junction of conical and cylindrical shells

1.1 *Edge influence coefficients*

In the general case, the elastic deformations of a long conical shell, loaded axisymmetrically by gas pressure and edge forces, can be described in the following dimensionless form:

$$\begin{aligned} \bar{w} &= w_Q\bar{Q} + w_M\bar{M} + w_p\bar{p}\,, \\ w' &= v_Q\bar{Q} + v_M\bar{M} + v_p\bar{p}\,, \end{aligned} \qquad (1a-b)$$

where

$$\begin{aligned} \bar{w} &= w/R\,, \\ \bar{Q} &= Q/ER\,, \\ \bar{M} &= M/ER^2\,, \\ \bar{p} &= p/E. \end{aligned} \qquad (2a-d)$$

In the following, the dash mark on top always denotes the dimensionless form of the given force, moment or deformation.

The coefficients w_Q, w_M, etc. are called edge influence coefficients.

Since the vessels under investigation are made of thin-walled non-shallow conical shells, Geckeler's relatively simple approximation is applicable, giving, in the size range investigated, a practically perfect approximation.

It can be derived that

$$\begin{aligned} w_Q &= 2\psi k^{3/2}\sqrt{\cos\alpha}\,, \\ w_M &= v_Q = 2\psi^2 k^2\,, \\ v_M &= 4\psi^3 k^{5/2}/\sqrt{\cos\alpha}\,, \\ w_p &= \frac{2-\mu}{2\cos\alpha}\cdot k\,, \\ v_p &\cong 0\,, \end{aligned} \tag{3a—e}$$

where

$$k = \frac{R}{\delta}\,, \tag{4}$$

$$\psi = 3(1-\mu^2)^{1/4} = 1.2854 \quad (\mu = 0.3)\,. \tag{5}$$

Formulae (3a to e) yield, in case of $\alpha = 0$, the edge influence coefficients of the cylindrical shell.

The terms with subscript p include the deformations due to membrane loads (charge pressure) and membrane forces.

1.2. *The system of deformation equations*

In the manner usual for hyperstatic structures, let the vessel be divided into parts at the bottom junction, and indicate the total load acting upon the individual shell parts (Fig. 1). For the sake of simple treatment, let us separate the three shells at the common junction line, and separate the common seam weld, otherwise regarded as of zero rigidity.

Let us write, according to the expressions (1a, b), the deformations of the edges of the individual shells, taking the directions indicated in Fig. 1 as positive.

$$\begin{aligned} \overline{w}_1 &= -w_{Q1}\bar{Q}_1 + w_{M1}\overline{M}_1 + w_{p1}\overline{p}\,, \\ w_1' &= -v_{Q1}\bar{Q}_1 + v_{M1}\overline{M}_1 + v_{p1}\overline{p}\,, \\ \overline{w}_2 &= -w_{Q2}\bar{Q}_2 + w_{M2}\overline{M}_2 + w_{p2}\overline{p}\,, \\ w_2' &= v_{Q2}\bar{Q}_2 - v_{M2}\overline{M}_2 - v_{p2}\overline{p}\,, \\ \overline{w}_3 &= -w_{Q3}\bar{Q}_3 + w_{M3}\overline{M}_3 - w_{p3}\overline{p}\,, \\ w_3' &= v_{Q3}\bar{Q}_3 - v_{M3}\overline{M}_3 + 0\,. \end{aligned} \tag{6a—f}$$

The equality of the deformations of joining shell edges, further the equilibrium of moments and forces perpendicular to the axis are expressed by the following equations:

$$
\begin{aligned}
\bar{w}_1 &= \bar{w}_2 , \\
\bar{w}_1 &= \bar{w}_3 , \\
w'_1 &= w'_2 , \\
w'_1 &= w'_3 , \\
\bar{Q}_1 &+ \bar{Q}_2 + \bar{Q}_3 - \bar{P} = 0 , \\
-\bar{M}_1 &+ \bar{M}_2 + \bar{M}_3 = 0 .
\end{aligned}
\tag{7a—f}
$$

Also following relationships can be written:

$$\bar{V}_1 - \bar{V}_2 + \bar{V}_3 = 0 ,$$

$$\bar{V}_1 = \bar{N}_{x1} = \frac{\frac{p_0 R}{2}}{ER} = 0.5\bar{p}_0 , \tag{8a}$$

$$\bar{V}_2 = \bar{N}_{x2} \cos\alpha = \frac{p_0 R^2 \pi + G_1 + G_2}{2\pi R \cdot ER} = 0.5\bar{p} + 0.5 \frac{H\gamma}{E} L , \tag{8b}$$

$$\bar{V}_3 = \bar{N}_{x3} = \frac{G_1 + G_2}{2\pi R \cdot ER} = 0.5 \frac{H\gamma}{E} (1 + L) , \tag{8c}$$

$$\bar{P} = \bar{V}_2 \operatorname{tg} \alpha , \tag{9}$$

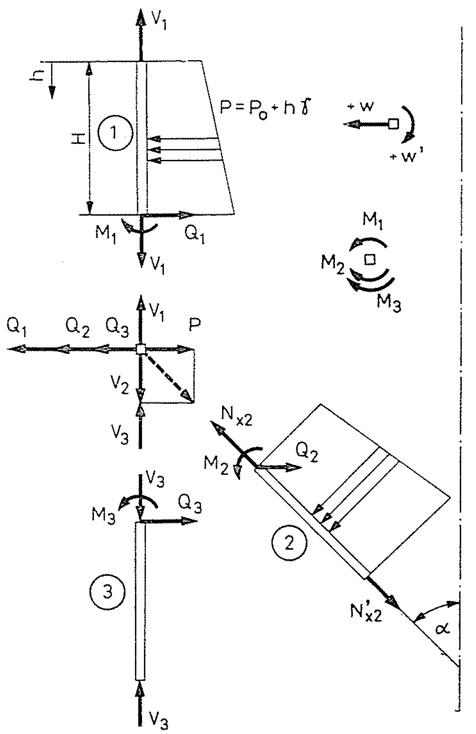

Fig. 1. Force system at the junction of shells

where

$$p = p_0 + H\gamma, \tag{10}$$

$G_1 = R^2\pi H\gamma$, the weight of the charge of the cylindrical part,

G_2 the total weight of the conical part, and

$$L = \frac{G_2}{G_1}. \tag{11}$$

The coefficients of Eqs (7a to f) can easily be calculated by means of formulae (6a to f) and (3a to e). The solution of the system of equations, that is, the determination of unknown forces Q and moments M, with given geometrical data and under given loading conditions, can be obtained by a simple computer program.

1.3. *Solutions in special cases*

In a general case, the loads arising at the junction are complicated algebraic functions of $k_1, k_2, k_3, \alpha, L, p$, extremely cumbersome for establishing in a closed form. The investigation of several practically important cases leads, however, to simple and perspicuous results. These results will be presented, disregarding the lengthy algebraic transformations, in the following:

Vessel without skirt, loaded by gas pressure

Be $\delta_3 = 0$, $H\gamma = 0$ and $\delta_2 = \delta_1$. Then the moment is

$$\overline{M}_1 = \overline{M}_2 = \frac{1}{4\psi\sqrt{k_1}}\,\overline{P}c = \frac{\operatorname{tg}\alpha}{4\psi\sqrt{k_1}}\,c\overline{V}, \tag{12a}$$

the shear force is

$$\overline{Q}_1 = \overline{Q}_2 = 0.5\overline{P}c, \tag{12b}$$

the secondary displacement is

$$\Delta\overline{w} = -0.5\psi k^{3/2}\,\overline{P}c, \tag{12c}$$

where

$$c = \frac{2\sqrt{\cos\alpha}}{1+\sqrt{\cos\alpha}}, \tag{12d}$$

$$\overline{P} = \frac{1}{2}\operatorname{tg}\alpha\,\frac{p_0}{E}.$$

Formulae (12a to d) contain no term directly including pressure p_0, since the stresses arising from unequal membrane deformations are negligible in comparison to stresses due to a break in the meridian curve.

The relationships (12a to d) can reasonably be applied to the case of two cylinders, too, which are loaded on their adjoining edges by a line load $\overline{P}$. Then $\alpha = 0$, $c = 1$.

It is remarkable that in case $\alpha = 30^\circ$ $c = 0.964$, that is, the shear force and moment arising in a conical-cylindrical shell under line load $\bar{P}$ hardly differ from the shear force and moment arising in a cylindrical shell under the same load.

With the method presented, also the analysis of the junction of two conical shells is simple, but now this case is disregarded.

Vessel with skirt support

A practically very important case is that of $\delta_1 = \delta_3$, that is, the wall thickness of the cylindrical container part and of the skirt are the same. Further, be $\alpha \leqq 45^\circ$, $k_1 = R/\delta_1$, $z = \delta_2/\delta_1$.

In case of an edge load $\bar{P}$ (case A of loading), by considering formula (9),

$$\bar{Q}_{2A} = \frac{z^2(0.5z^2 + 2/\sqrt{z\cos\alpha})}{4D}\,\mathrm{tg}\,\alpha\,\bar{V}_2\,,$$

$$\bar{M}_{1A} = \frac{1 + 0.5z^2(3 - 1/\sqrt{z\cos\alpha})}{D}\,\frac{\mathrm{tg}\,\alpha}{4\psi\sqrt{k_1}}\,\bar{V}_2\,, \qquad \text{(13a—d)}$$

$$\bar{M}_{2A} = \frac{z^2}{D}\,\frac{\mathrm{tg}\,\alpha}{4\psi\sqrt{k_1}}\,\bar{V}_2\,,$$

$$\bar{M}_{3A} = \bar{M}_{1A} - \bar{M}_{2A}\,,$$

and the secondary displacement

$$\Delta\bar{w}_A = -0.5\psi k_1^{3/2}\,\frac{1 + z^2(1 - 0.5/\sqrt{z\cos\alpha}\,)}{D}\,\mathrm{tg}\,\alpha\,\bar{V}_2\,, \qquad \text{(13e)}$$

where

$$D = 0{,}125\,z^4 + z^2 + 1\,. \qquad (14)$$

In case of a container loaded by a gas pressure $\bar{p}_0$ (case B of loading), the moments, force and displacement arising from the difference of membrane deformations are the following:

$$\bar{M}_{2B} = \frac{2-\mu}{2}\,\frac{z^2}{16\psi^2}\,\frac{z^2 + 4\sqrt{z\cos\alpha} + 8/(z\cos\alpha) - 4}{D}\,\frac{\bar{p}}{k_1}\,,$$

$$\bar{Q}_{2B} = (0.5z^2 + 2/\sqrt{z\cos\alpha}\,)\psi\sqrt{k_1}\,\bar{M}_{2B} - \frac{2-\mu}{2}\,\frac{z^2}{4\psi}\,\frac{\bar{p}}{k_1}\,, \qquad \text{(15a—e)}$$

$$\bar{M}_{1B} = \frac{1}{2}\bar{M}_{2B} - \frac{\bar{Q}_{2B}}{4\psi\sqrt{k_1}}\,,$$

$$\bar{M}_{3B} = \bar{M}_{1B} - \bar{M}_{2B}\,,$$

$$\Delta\bar{w}_B = -2\psi k_1^{3/2}\sqrt{\cos\alpha}\;Q_{2B} + 2\psi^2 k_1^2 \bar{M}_{2B}\,.$$

2. The method of design

2.1. *Calculation method of stresses*

The stresses arising in the extreme fibres of a shell are given by the following formulae:

$$\sigma_{xR} = \sigma_{xM} + \sigma_{xH} ,\qquad (16a-b)$$
$$\sigma_{\varphi R} = \sigma_{\varphi M} + \sigma_{\varphi 0} + \sigma_{\varphi H} ,$$

where for a conical, in case of $\alpha = 0$ for a cylindrical shell

$$\sigma_{xM} = \frac{N_x}{\delta} = E\overline{N}_x k ,$$
$$\sigma_{\varphi M} = \frac{p_0 + H\gamma}{\cos\alpha}\frac{R}{\delta} = \frac{p_0 + H\gamma}{\cos\alpha} k , \qquad (17a-e)$$
$$\sigma_{\varphi 0} = E\frac{\Delta w}{R} = E\Delta\overline{w} ,$$
$$\sigma_{xH} = \pm\frac{6M_x}{\delta^2} = \pm 6\,\overline{M}k^2E ,$$
$$\sigma_{\varphi H} \cong \mu\sigma_{xH} .$$

With the methods given in the previous chapters, the moment on the edges ($\overline{M}_x = \overline{M}$) and the secondary displacement ($\Delta\overline{w}$) can be determined and the stresses calculated.

It lies outside the scope of this paper to discuss in detail, how the stresses at a greater distance from the edge of the shell can be calculated. In this connection we refer to the technical literature [3].

It must be noted, however, that the stresses rapidly decrease when moving away from the edge; therefore a change (decrease) in the wall thickness at a suitable distance from the junction has no influence on the value of the stress peak.

In the numerical example No. 2, the stress development in a vessel of given dimensions is shown.

2.2. *Introduction of stress concentration factors*

Let us relate the concentration factors of the stresses to the hoop membrane stress arising in the cylinder, that is, be

$$F = \frac{\sigma_{xR}}{\sigma_{\varphi M1}} = \frac{\sigma_{xR}}{pk_1} . \qquad (18)$$

The greatest stress in the range investigated is always axial.

Be the cylinder and the cone denoted by subscripts 1 and 2, respectively. Quantities with no subscript refer to the tested shell. It can be shown that in a general case

$$F = \frac{0.5}{\cos\alpha}\,\frac{2\overline{V}}{\overline{p}} + g + f\sqrt{k_1}\,\frac{2\overline{V}_2}{\overline{p}}\,, \tag{19}$$

where the first term is characteristic for the membrane stress, the second for the stress arising from the difference of membrane deformations, the third, finally, the stress originating from the break in the meridian curve. If $\delta_1 = \delta_3$, both g and f are merely functions of α and $z = \delta_2/\delta_1$.

It is valid that

$$f = \frac{1}{\sqrt{k_1}} \cdot \frac{\sigma_{xHA}}{\sigma_{\varphi M1}} = \frac{1}{\sqrt{k_1}} \cdot \frac{6\overline{M}_A k^2}{(2\overline{V}_2/\overline{p})k_1} = 6\overline{M}_A \sqrt{k_1}/z^2/(2\overline{V}_2/\overline{p})\,,$$

$$g = \frac{\sigma_{xHB}}{\sigma_{\varphi M1}} = \frac{6\overline{M}_B k^2}{\overline{p}k_1} = 6\overline{M}_B k_1/(z^2\overline{p})\,, \tag{20a—b}$$

where k_1 and k are the radius to thickness ratios of the cylinder and of the just tested shell, respectively.

Vessel without skirt, loaded by gas pressure

Take the above case, with $\delta_2 = \delta_1$, $k_2 = k_1$. With the use of the relationships (20a), (12a), (12d), (8a) it can be derived that

$$f_1^* = f_1 = f_2 = \frac{6k_1^2}{k_1\sqrt{k_1}} \cdot \frac{\overline{M}_1}{2\overline{V}_2/\overline{p}_0} = \frac{2\sqrt{\cos\alpha}}{1+\sqrt{\cos\alpha}} \cdot \frac{3}{4\psi}\,\mathrm{tg}\,\alpha\,. \tag{21}$$

Obviously $g_1 = g_2 = 0$; further considering that $\overline{V}/\overline{p}_0 = 0.5$, the stress concentration in the cone:

$$\frac{\sigma_{xR}}{p_0 k_1} = \frac{0.5}{\cos\alpha} + 0.583\,\frac{2\sqrt{\cos\alpha}}{1+\sqrt{\cos\alpha}}\,\mathrm{tg}\,\alpha\sqrt{k_1}\,. \tag{22}$$

The stress concentration factors f_1^* calculated with the formula (21) are given in the second column of Table 1. The numerical values agree with the values found in literature [2].

Vessel with skirt support

The calculation is entirely analogous with the previous case. As a result, it is obtained that

$$f_1 = \frac{6\overline{M}_{1A}k_1^2}{k_1\sqrt{k_1}(2\overline{V}_2/\overline{p})} = \frac{1 + 0.5z^2(3 - 1/\sqrt{z\cos\alpha})}{D}\frac{3}{4\psi}\operatorname{tg}\alpha\,,$$

$$f_2 = \frac{6\overline{M}_{2A}k_2^2}{k_1\sqrt{k_1}(2\overline{V}_2/\overline{p})} = \frac{1}{D}\frac{3}{4\psi}\operatorname{tg}\alpha\,, \tag{23a—c}$$

$$g_1 = \frac{6\overline{M}_{1B}k_1^2}{k_1\overline{p}} = 6\overline{M}_{1B}k_1\,,$$

$$g_2 = \frac{6\overline{M}_{2B}k_2^2}{k_1\overline{p}} = \frac{2-\mu}{2}\frac{3}{8\psi^2}\frac{z^2 + 4\sqrt{z\cos\alpha} + 8/(z\cos\alpha) - 4}{D}\,.$$

The values of factors calculated in this way are given, for the case of $\delta_1 = \delta_2 = \delta_3$, in Table 1.

Table I

α^0	f_1^*	f_1	g_1	f_2	g_2
10	0,103	0,097	0,089	0,048	0,829
20	0,209	0,198	0,073	0,100	0,853
30	0,323	0,310	0,030	0,158	0,905
45	0,533	0,523	—0,039	0,275	1,062

It is obvious from the table that the stress state of the upper cylindrical shell is not considerably different whether there is, or there is no skirt ($f_1^* \cong f_1$). The factor g_1, characterizing the difference of membrane deformations, is not high.

The factor f_2 of the conical shell is practically half of f_1; thus, in this regard, the skirt releases the cone. At the same time, however, a considerable increase appears in the secondary stress arising from the difference of membrane deformations (g_2).

2.3. *Introduction of the shape factor*

The conical and cylindrical parts of the junction are statically adequate, if

$$\sigma_{xR} = \frac{pR}{\delta} F \leq \sigma_m \cdot v\,, \tag{24a}$$

and

$$s' = \delta \geqq \frac{pR}{\sigma_m \cdot v} F\,, \tag{24b}$$

where $F = F(\alpha, \delta_2/\delta_1)$ is the stress concentration factor of the tested shell.

These formulae are valid in the elastic range. In non-elastic range, the stress concentration factors are already not characteristic of the value of stresses, but they are suitable for the limitation of the degree of plastic deformations.

Be the necessary wall thickness

$$s' = s_0' \cdot y \quad \text{and} \quad s_0' = \frac{pd}{2\sigma_m \cdot v}, \tag{25a—b}$$

where the shape factor y expresses how many times thicker wall is needed in case of a structural shape different from a cylindrical membrane shell.

In order to limit the degree of plastic deformations, be

$$y = \frac{F}{2.6}, \tag{26}$$

that is, since

$$\sigma_m = \frac{\sigma_F}{1.5} \text{ or } \frac{\sigma_B}{2.6}, \tag{27a—b}$$

$$\sigma_{\text{nominal}} = \frac{2.6}{1.5}\sigma_F = 1.73\sigma_F\text{ , or}$$

$$\sigma_{\text{nominal}} = \sigma_B\,. \tag{28a—b}$$

The above consideration is not valid to cases where the structural material has no suitable plastic reserve, if certain causes may provoke embrittlement, alternating loading or stress corrosion may occur, discontinuities exist on the adjoining seam of the conical bottom, giving rise to fracture.

Comparing the suggested design formula with the Hungarian, German, etc. standard shape factors, valid for conical bottoms, the application of standard shape factors can be stated not to be sufficiently safe in case of relatively thin shells occurring in container-building; i.e. they permit too large plastic deformations, likely to cause the failure of the device. Namely, the shape factors of standard formulae are based on experimental investigations on the usual size range of pressure vessels.

The shape factor can be determined, in knowledge of the stress concentration factor, by iteration, or, in knowledge of s_0', in the following way:

Be $$k_0 = R/s_0', \quad k_1 = R/s_1'. \tag{29a—b}$$

The shape factor of the cylindrical part, according to formulae (19) and (26), for $p_0 = 0$, $p = H\gamma$,

$$y_1 = \frac{F_1}{2.6} = \frac{g_1 + f_1\sqrt{k_1}\,2\bar{V}_2/\bar{p}}{2.6}.$$

Considering also formula (25a),

$$(2.6y_1 - g_1)^2 = (f_1 2\bar{V}_2/\bar{p})^2 k_0/y\,.$$

Since for the cylindrical part $g_1 \ll 2.6y_1$,

$$y_1 = \sqrt[3]{\left[\frac{f_1 2\bar{V}_2/\bar{p}}{2.6}\right]^2 \cdot k_0}\,. \tag{30}$$

The values calculated by formula (30) are shown in Fig. 2.

The suggested method of strength analysis is as follows: assuming $\delta_1 = \delta_2 = \delta_3$, the values of y_1 and s_1' are determined from formulae (30) and

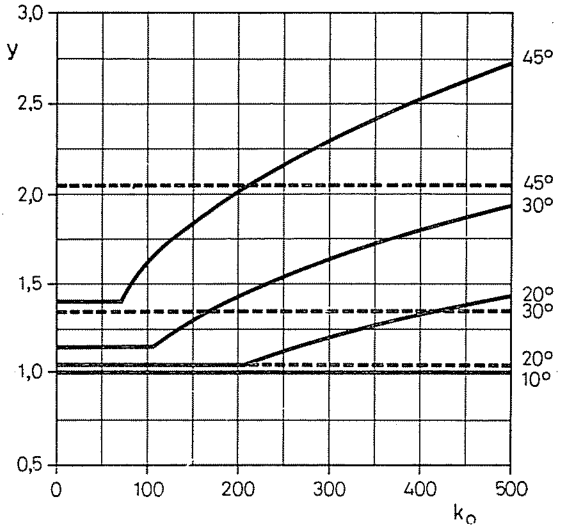

Fig. 2. Values of the shape factor for $\delta_1 = \delta_2 = \delta_3$; $k_0 = R/s_0' = H\gamma/(\sigma_m \cdot v)$, $\sigma_{mH}/\sigma_m = 2.6$ $L = 0$. Note: Full line = computed shape factor, Dashed line = standard shape factor, valid for pressurized conical heads

(25a), then the stresses are calculated and checked by formulae (23), (19), (24), with the wall thickness obtained above, both for the cone and the cylinder. If necessary, after changing the wall thickness of the cylinder or of the cone, calculation and checking of the stresses are iterated. Modifications may be required e.g. by the joint efficiency factor different between the cylinder and the cone. Of course, y_1 cannot be smaller than 1.

Be the length of the thickened part surrounding the junction $2.0\sqrt{Rs'}$.

Let it be noted that the shape factor introduced in this way is analogous with the shape factor, valid for conical reducers, to be found in the ASME Boiler and Pressure Vessel Code (Section VIII, Div. 2).

Numerical example No. 1

The data of the container with structural design according to Fig. 3 are the following:

$H = 2500$ cm $\alpha = 20°$

$\gamma = 1.5 \cdot 10^{-3}$ kp/cm^3 $d = 1000$ cm

$L = 0.1$ $\sigma_m \cdot v = 1600$ kp/cm^2.

From formulae (25b) and (29a):

$$s_0' = \frac{2500 \cdot 1.5 \cdot 10^{-3} \cdot 1000}{2 \cdot 1600} = 1.172 \text{ cm},$$

$$k_0 = \frac{500}{1.172} = 426.7 .$$

Assuming $z = 1$, from Table 1, $f_1 = 0.198$, $g_1 = 0.073$, from formula (8b) $2\overline{V}_2/\overline{p} = 2 \cdot 0.5(1 + L) = 1.1$.

According to formula (30)

$$y = \sqrt[3]{\left[\frac{0.198}{2.6} \cdot 1.1\right]^2 \cdot 426.7} = 1.445 .$$

Thus, from formulae (25a) and (29b)

$$s' = 1.17 \cdot 1.445 = 1.69 \text{ cm},$$

$$k_1 = \frac{500}{1.69} = 295.8 .$$

The stress concentration factor according to formula (19):

$$F_1 = 0.073 + 0.198 \sqrt{295.8} \cdot 1.1 = 3.82 .$$

The maximum nominal stress:

$$\sigma_{\text{nom}} = pkF = 3.75 \cdot 295.8 \cdot 3.82 = 4237 \text{ kp/cm}^2.$$

$$\frac{\sigma_{\text{nom}}}{2.6} = \frac{4237}{2.6} = 1630 \text{ kp/cm}^2 \cong 1600 \text{ kp/cm}^2 .$$

Accordingly, the wall thickness of the cylinder is suitable.

The wall thickness of the cone can similarly be checked.

If no plastic deformation is permitted, but it is allowed for the bending stress to exceed the membrane stress by 1.4, then

$$y = \sqrt{\left[\frac{0.198}{1.4} \cdot 1.1\right]^2 426.7} = 2.18 .$$

The German design specification AD-Merkblatt B2—1969, further the Hungarian Standard MSz 13825—1970 specify the shape factor of a vessel bottom of dimensions as given above, as $y = 2.0/2 = 1.0$; they are evidently not valid in this size range.

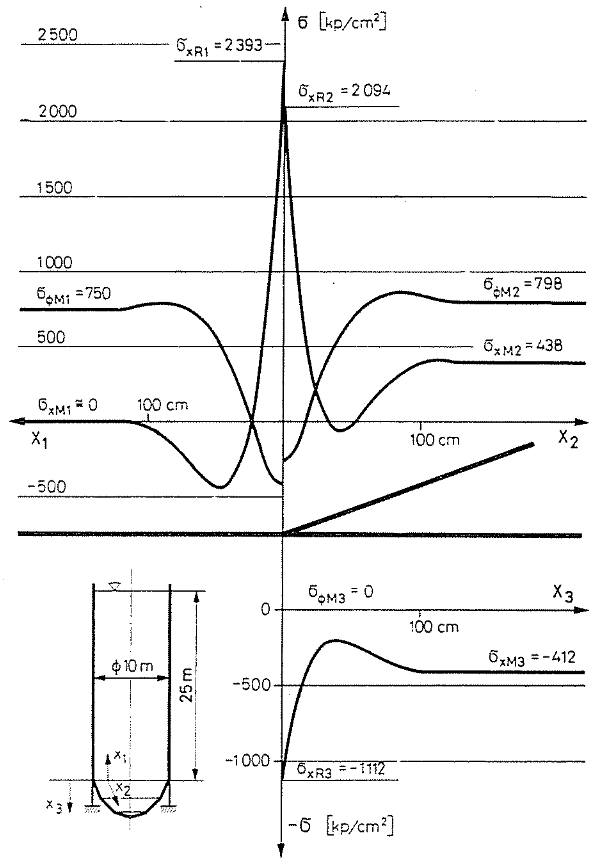

Fig. 3. Diagrams of stresses in the elastic range $\alpha = 20°$; $R/\delta_1 = 200$; $H\gamma = 3{,}75$ kp/cm^2; $L = 0{,}1$; $\delta_1 = \delta_2 = \delta_3$

Numerical example No. 2

Fig. 3 shows the diagrams of characteristic stresses arising in the vessel investigated in example No. 1, in case of $R/\delta = 200$, computed by a computer type ODRA-1204.

Summary

The stress concentration factor for the surrounding of the cone-cylinder-skirt junction is determined using the edge influence coefficients calculated with Geckeler's approximation. For practical calculation purposes, a shape factor is introduced, allowing a limited plastic deformation of the wall. In case of large-size thin-walled structures, the necessary wall thickness is considerably larger than the one calculable according to standards applied in the design of pressure vessels of conventional dimensions.

Notations

d cm	mean diameter of the cylindrical shell
f, g	stress concentration factors
k	radius to wall thickness ratio
p_0 kp/cm^2	gas pressure
p kp/cm^2	total pressure at the junction
s' cm	structurally necessary wall thickness near the junction
s_0 cm	wall thickness of the cylindrical shell at the junction, calculated from the membrane hoop stress
v	joint efficiency factor
w cm	displacement of the middle surface of the shell
Δw cm	displacement under the effect of secondary loads
w' rad	rotation of the meridian curve
x cm	co-ordinate in meridian direction
y	shape factor
z	ratio of wall thicknesses of conical to cylindrical shell
D	calculation factor
E kp/cm^2	Young's modulus
F	stress concentration factor
G kp	charge weight
$H\gamma$ kp/cm^2	static head of the fluid
L	ratio of charge weights
M cmkp/cm	edge moment
N kp/cm	meridional edge force
P kp/cm	radial line load
Q kp/cm	radial edge force
R cm	mean radius of the cylindrical shell
α [°]	half apex angle of the bottom
γ kp/cm^3	specific weight of the charge
δ cm	shell thickness
μ	Poisson's ratio
ψ	calculation factor
σ kp/cm^2	stress

Subscripts

1, 2, 3	referring to cylinder, cone, skirt
x	meridional direction
φ	hoop direction
H	bending
M	membrane
R	resultant
m	allowable
F	yield point
B	tensile strength

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Dr. Pál REUSS, H-1521 Budapest