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OPTIMAL DESIGN OF CYLINDRICAL RINGS USED FOR THE SHRINKAGE OF VEHICLE TANKS FOR COMPRESSED NATURAL GAS

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Abstract: The objective of this paper is to present a rapid method that allows the optimal design of cylindrical rings used for the shrinkage of vehicle tanks for compressed natural gas in the automotive industry. Their design is performed using a base of parameterized objects designed for this purpose, based on the Finite Element Method (FEM). The design characteristics, described in the modeling procedure, can be used to quickly modify/update the design.
Keywords: engineering design, Finite Element Method, optimization, shrink, vehicle tank

1. INTRODUCTION

Over the last two decades, the global auto industry has undergone important structural transformations and global manufacturing competition has increased significantly [1-6].

Computer simulation is a key part of the automotive development process to reduce product development cost and time while improving the safety, comfort, and durability of the vehicles they produce [2, 7-13]. The 3D solid modeling provides a framework to model and represents an object's shape in the computer, and to perform operations [2]. It provides the basis for design, simulation, and manufacturing of any part and assembly of the vehicles [14].

A natural gas vehicle is an alternative fuel vehicle that uses compressed natural gas (CNG) (which is mainly composed of methane and is stored in compressed form) or liquefied natural gas (LNG) (stored at a low temperature in liquid form) as an alternative to other fossil fuels [3, 15, 16].

CNG are stored in high pressure cylinders that can be molded in intricate shapes to store more fuel and use less on-vehicle space [17]. In the field of vehicle's methane gas tank different solutions for shape optimization of gas tanks were proposed [18-21]. These solutions are based on different complex 3D shapes [22-29] which are represented using various mathematical and graphical methods [30-33].

In our study, multilayer cylindrical covers with preload, represents an alternative to the side symetrical covers for manufacturing pressurized CNG tanks (e.g. methane gas), used for vehicles. By shrinking, it allows the increasing of stress resistence of the tank and a better distribution of the stress in the side cover [3]. Constructively, these covers are made from an assembly of concentric tubes, to which, on the contact surfaces between cylinders appears a shrink pressure that gives elastic and plastic deformation of the components. The stress introduced by shrinkage is merged over the introduced by recipient pressurization [3]. This paper will approach the problem of optimal dimensioning of a cover that presents elastic deformation on the contact surface between cylinders for an assembly containing 2 elements (Figs. 1, 2). Use of this model becomes convenient from the mathematical view point. The mounting of cylinders can be made: by heating of outer cylindrical element (2), or by cooling of inner element (1), thus, before mounting, the radius of external surface of inner element $r = b_1$, is bigger than the radius of internal surface of outer element, $r = b_2$, after mounting the internal cylinder is radial compressed with u_{B1} and the external cylinder is expanding with u_{B2} (Figs. 1, 2).

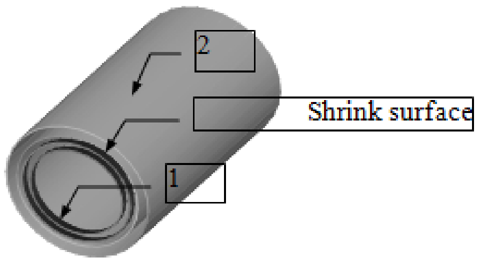

Figure 1. Constructive shape of the cylindrical shrinked cover

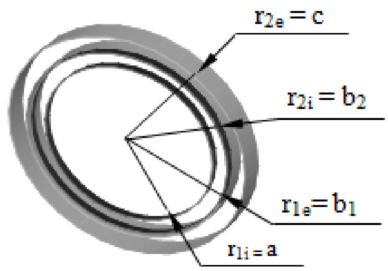

Figure 2. Cylindrical shrinked cover with the relevant dimensions

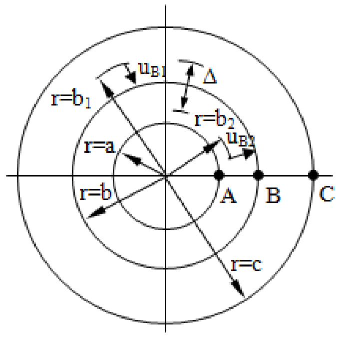

Figure 3. Schematic diagram qualitatively showing the shrinked cylindrical sides cover in initial conditions

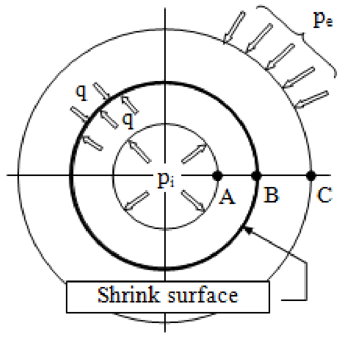

Figure 4. Schematic diagram qualitatively showing the shrinked cylindrical sides cover in final conditions

2. OPTIMIZATION METHOD PRINCIPLE

The paper proposed an optimization method of shrinked cylindrical side covers using a base of 3D primitives parameterized using a combination of numerical calculation programs and the Finite Element Method (FEM) [34]. The optimization process contains the following steps:

a) establishing the number of cylindrical elements that make shrinkage side cover of the gas tank (Figs. 1–4);
b) determination of optimal dimensions of cylinders used for joining and shrink pressure, by using numerical calculation programs special designated;
c) adopting real constructive values and re-calculation of shrinkage corresponding pressures;
d) selecting the cylinders used for covers from the parameterized 3D primitives;
e) choosing the material for components;
f) making a study of stress and spatial deformation of the real model using the FEM.

3. DIMENSIONAL OPTIMIZATION OF SHRINKED COVER

Let's consider a side cover with axial symmetry made by 2 concentric cylinders and from the same material.

3.1. Calculations of stresses and deformations for a pressurized cylinder

In the case of a pressurized cylinder, with constructive symmetry and axial loading symmetry, one element of the cover in radial section is subjected to tangent effort σ_θ and radial effort σ_r, (Fig. 5).

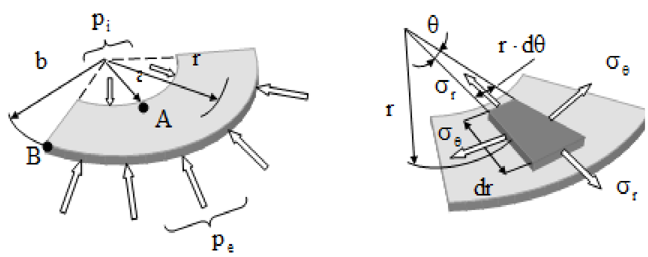

Figure 5. The relevant dimensions and the spatial loads and constraints used in the analysis

The equation of static equilibrium of the element radial projected is:

$$\frac{d\sigma_r}{dr} - \frac{\sigma_\theta - \sigma_r}{r} = 0 \tag{1}$$

and applying the Hooke's law for elastic deformation:

$$\varepsilon_\theta = \frac{u}{r} = \frac{1}{E}\left(\sigma_\theta - \mu\sigma_r\right) \tag{2}$$

$$\varepsilon_r = \frac{du}{dr} = \frac{1}{E}\left(\sigma_r - \mu\sigma_\theta\right) \tag{3}$$

by eliminating σ_r and σ_θ efforts from relations (2) and (3) and replacing in (1), it is obtained the equation of radial displacement:

$$u(r) = K_1 \cdot r + \frac{K_2}{r^2} \tag{4}$$

The integration constants K_1 and K_2 are obtained from boundary conditions for cover stress, on internal surface at $r = a$ (corresponding for pressure p_i) and on external surface at $r = b$ (corresponding for pressure p_e) (Fig. 5). After replacing, we obtain the following general calculation relations:

$$\sigma_\theta = \frac{p_i - k^2 p_e}{k^2 - 1} + \frac{p_i - p_e}{k^2 - 1}\frac{b^2}{r^2} \tag{5}$$

$$\sigma_r = \frac{p_i - k^2 p_e}{k^2 - 1} - \frac{p_i - p_e}{k^2 - 1}\frac{b^2}{r^2} \tag{6}$$

$$u(r) = \frac{1-\mu}{E}\frac{p_i - k^2 p_e}{k^2 - 1} \cdot r + \frac{1+\mu}{E}\frac{p_i - p_e}{k^2 - 1} \cdot \frac{b^2}{r^2} \tag{7}$$

for $k = b/a$.

Because the tank has lids at the ends, the stretching stress σ_z applied to the cover due to internal pressure p_i over the lids, comes out from static equilibrium equation of forces that are acting on a ring section of the tank, projected on symmetry axis of the cover:

$$\sigma_z = \frac{p_i}{k^2 - 1} \tag{8}$$

3.2. Determination of stress and deformation of the cover

Determination of stress and deformation is made using the principle of effect overlaping, as a result of stress summing due to shrink pressure of the assembly and stress due to gas pressure from inside the tank.

Using relations: (5), (6) and (7) for cylinder 1, we have $p_i = 0$ and $p_e = q$, resulting the following calculations:

$$\begin{aligned}\sigma^{r\Delta}{}_\theta(r) &= \left(\frac{p_i - k_{1r}^2 p_e}{k_{1r}^2 - 1} + \frac{p_i - p_e}{k_{1r}^2 - 1}\cdot\frac{b_r^2}{r^2}\right)\Bigg|_{p_i = 0;\, \mathrm{p_e} = q;\, r \in [a_r, b_r]} = \\ &= \left(\frac{-k_{1r}^2 q}{k_{1r}^2 - 1} + \frac{-q}{k_{1r}^2 - 1}\cdot\frac{b_r^2}{r^2}\right)\Bigg|_{r \in [a_r, b_r]} = \frac{-q}{k_{1r}^2 - 1}\cdot\left(k_{1r}^2 + \frac{b_r^2}{r^2}\right)\Bigg|_{r \in [a_r, b_r]}\end{aligned} \tag{9}$$

$$\begin{aligned}\sigma^{r\Delta}{}_r(r) &= \left(\frac{p_i - k_{1r}^2 p_e}{k_{1r}^2 - 1} - \frac{p_i - p_e}{k_{1r}^2 - 1}\cdot\frac{b_{1r}^2}{r^2}\right)\Bigg|_{p_i = 0;\, \mathrm{p_e} = q;\, r \in [a_r, b_r]} = \\ &= \left(\frac{-k_{1r}^2 q}{k_{1r}^2 - 1} - \frac{-q}{k_{1r}^2 - 1}\cdot\frac{b_{1r}^2}{r^2}\right)\Bigg|_{r \in [a_r, b_r]} = \frac{-q}{k_{1r}^2 - 1}\cdot\left(k_{1r}^2 - \frac{b_{1r}^2}{r^2}\right)\Bigg|_{r \in [a_r, b_r]}\end{aligned} \tag{10}$$

$$\sigma^{r\Delta}{}_z = 0 \tag{11}$$

and for cylinder 2, with $p_i = 0$ and $p_e = 0$, it results:

$$\begin{aligned}\sigma^{r}{}_\theta(r) &= \left(\frac{p_i - k_{2r}^2 p_e}{k_{2r}^2 - 1} + \frac{p_i - p_e}{k_{2r}^2 - 1}\cdot\frac{c_r^2}{r^2}\right)\Bigg|_{p_i = q;\, \mathrm{p_e} = 0;\, r \in [b_r, c_r]} = \\ &= \left(\frac{q}{k_{2r}^2 - 1} + \frac{q}{k_{2r}^2 - 1}\cdot\frac{c_r^2}{r^2}\right)\Bigg|_{r \in [b_r, c_r]} = \frac{q}{k_{2r}^2 - 1}\cdot\left(1 + \frac{c_r^2}{r^2}\right)\Bigg|_{r \in [b_r, c_r]}\end{aligned} \tag{12}$$

$$\sigma^r{}_r(r)=\left(\frac{p_i-k_{2r}^2 p_e}{k_{2r}^2-1}-\frac{p_i-p_e}{k_{2r}^2-1}\cdot\frac{c_r^2}{r^2}\right)\Bigg|_{p_i=q;\,\mathrm{p_e}=0;\,r\in[b_r,c_r]}= \\ =\left(\frac{q}{k_{2r}^2-1}-\frac{q}{k_{2r}^2-1}\cdot\frac{c_r^2}{r^2}\right)\Bigg|_{r\in[b_r,c_r]}=\frac{q}{k_{2r}^2-1}\cdot\left(1-\frac{c_r^2}{r^2}\right)\Bigg|_{r\in[b_r,c_r]} \tag{13}$$

$$\sigma^r{}_z=0 \tag{14}$$

Using the same relations: (5), (6) and (7), by replacing $p_i=p_i$ and $p_e=0$, we obtain the following:

$$\sigma^{r\,p_i}{}_\theta(r)=\left(\frac{p_i-k_r^2 p_e}{k_r^2-1}+\frac{p_i-p_e}{k_r^2-1}\cdot\frac{c_r^2}{r^2}\right)\Bigg|_{\mathrm{p_i}=\mathrm{p_i};\,\mathrm{p_e}=0;\,r\in[a_r,c_r]}= \\ =\left(\frac{p_i}{k_r^2-1}+\frac{p_i}{k_r^2-1}\cdot\frac{c_r^2}{r^2}\right)\Bigg|_{r\in[a_r,c_r]}=\frac{p_i}{k_r^2-1}\cdot\left(1+\frac{c_r^2}{r^2}\right)\Bigg|_{r\in[a_r,c_r]} \tag{15}$$

$$\sigma^{r\,p_i}{}_r(r)=\left(\frac{p_i-k_r^2 p_e}{k_r^2-1}-\frac{p_i-p_e}{k_r^2-1}\cdot\frac{c_r^2}{r^2}\right)\Bigg|_{\mathrm{p_i}=\mathrm{p_i};\,\mathrm{p_e}=0;\,r\in[a_r,c_r]}= \\ =\left(\frac{p_i}{k_r^2-1}-\frac{p_i}{k_r^2-1}\cdot\frac{c_r^2}{r^2}\right)\Bigg|_{r\in[a_r,c_r]}=\frac{p_i}{k_r^2-1}\cdot\left(1-\frac{c_r^2}{r^2}\right)\Bigg|_{r\in[a_r,c_r]} \tag{16}$$

$$\sigma^{r\,\mathrm{p_i}}{}_z=\frac{p_i}{k_r^2-1} \tag{17}$$

This result by summing of stresses due to shrink pressure, cover pressurization and axial stress, according to effect overlaping principle, which is made with the following relations:

$$\sigma^r{}_\theta=\sigma^{r\,\Delta}{}_\theta+\sigma^{r\,\mathrm{p_i}}{}_\theta \tag{18}$$

$$\sigma^r{}_r=\sigma^{r\,\Delta}{}_r+\sigma^{r\,\mathrm{p_i}}{}_r \tag{19}$$

$$\sigma^r{}_z=\sigma^r{}_z \tag{20}$$

In points A and B_2 appears the higher stresses with the following values:

- for point A

$$\sigma^r{}_{\theta\,\mathrm{A}}=\sigma^{r\,\Delta}{}_{\theta\,\mathrm{A}}+\sigma^{r\,\mathrm{p_i}}{}_{\theta\,\mathrm{A}}=\frac{-2k_{1r}^2 q}{k_{1r}^2-1}+p_i\frac{k_r^2+1}{k_r^2-1} \tag{21}$$

$$\sigma^r{}_{r\,\mathrm{A}}=\sigma^{r\,\Delta}{}_{r\,\mathrm{A}}+\sigma^{r\,\mathrm{p_i}}{}_{r\,\mathrm{A}}=-p_i \tag{22}$$

$$\sigma^r{}_{z\,\mathrm{A}}=\frac{p_i}{k_r^2-1} \tag{23}$$

- for point B_2

$$\sigma^r{}_{\theta\,\mathrm{B_2}}=\sigma^{r\,\Delta}{}_{\theta\,\mathrm{B2}}+\sigma^{r\,\mathrm{p_i}}{}_{\theta\,\mathrm{B2}}=q\cdot\frac{k_{2r}^2+1}{k_{2r}^2-1}+p_i\frac{k_{2r}^2+1}{k_r^2-1} \tag{24}$$

$$\sigma^r{}_{r\,\mathrm{B_2}}=\sigma^{r\,\Delta}{}_{r\,\mathrm{B2}}+\sigma^{r\,\mathrm{p_i}}{}_{r\,\mathrm{B2}}=-q+p_i\frac{1-k_{2r}^2}{k_r^2-1} \tag{25}$$

$$\sigma^r{}_{z\,\mathrm{B_2}}=\frac{p_i}{k_r^2-1} \tag{26}$$

where: $\mathrm{k_{1r}}=\mathrm{b_r}/\mathrm{a_r}$ $\quad \mathrm{k_{2r}}=\mathrm{c_r}/\mathrm{b_r}$ $\quad \mathrm{k_r}=\mathrm{c_r}/\mathrm{a_r}$

The equivalent stress is calculated using the theory of maximum tangent tensile T_T, after that this value being compared with maximum limit of the material:

$$\sigma_{ech\ \mathrm{TT}}=\max\left(\left|\sigma_\theta-\sigma_r\right|,\left|\sigma_\theta-\sigma_z\right|,\left|\sigma_r-\sigma_z\right|\right)\le\sigma_a \tag{27}$$

in case of points with higher stress: A and B_2, we have:

- for point A

$$\sigma^r{}_{ech\ \mathrm{A}}=\left|-q\frac{2k_{1r}^2}{k_{1r}^2-1}+p_i\frac{2k_r^2}{k_r^2-1}\right|\le\sigma_a \tag{28}$$

- for point B_2

$$\sigma^r{}_{ech\ \mathrm{B}_2} = \left| q \frac{2k_{1r}^2}{k_{1r}^2 - 1} + p_i \frac{2k_{2r}^2}{k_r^2 - 1} \right| \leq \sigma_a \tag{29}$$

By solving the system of inequalities (28) and (29), the following values are obtained (named as optimal):

- the minimum external radius of outer cylinder:

$$c_o = a \cdot \frac{\sigma_a}{\sigma_a - p_i} \tag{30}$$

- the radius of the shrinked surface:

$$b_o = \sqrt{a \cdot c_o} \tag{31}$$

- the shrinkage pressure on the contact surface:

$$q_o = \frac{\sigma_a}{2} \cdot \frac{(k_o - 1)^2}{k_o (k_o + 1)} \tag{32}$$

and tightning of the assembly:

$$\Delta_o = \frac{2q_o}{E} \cdot \frac{b_o \cdot (k_o^2 - 1)}{(k_{1o}^2 - 1) \cdot (k_{2o}^2 - 1)} \tag{33}$$

where non-dimensional ratios: k_0, k_{10} and k_{20} are calculated with relations:

$$\mathrm{k_o = c_o / a}\ ;\ \ \mathrm{k_{1\,o} = b_0 / a}\ ;\ \ \mathrm{k_{2\,o} = c_0 / b} \tag{34}$$

Choosing the real values of dimension $c_r \geq c_0$, and based on formulas (31),..,(34), we recalculated the real values. These values were used for determination of radial displacements u_{B1} and u_{B2}, deduced from the Hooke's law for elastic deformation:

$$\varepsilon_\theta = \frac{u}{r} = \frac{1}{E}\left[\sigma_\theta - \mu(\sigma_r + \sigma_z)\right] \Rightarrow u = \frac{r}{E}\left[\sigma_\theta - \mu(\sigma_r + \sigma_z)\right] \tag{35}$$

By specifying in particular for stresses in points B_1 and B_2:

$$u^r{}_{B_1} = \frac{b_r}{E}\left[\sigma^r{}_{\theta \mathrm{B}_1} - \mu\left(\sigma^r{}_{r\mathrm{B}_1} + \sigma^r{}_{z\mathrm{B}_1}\right)\right] \tag{36}$$

$$u^r{}_{B_2} = \frac{b_r}{E}\left[\sigma^r{}_{\theta \mathrm{B}_2} - \mu\left(\sigma^r{}_{r\mathrm{B}_2} + \sigma^r{}_{z\mathrm{B}_2}\right)\right] \tag{37}$$

and after the replacement in formulas are obtained next relations:

$$u^r{}_{B_1} = \frac{b_r}{E}\left[-q\frac{k_{1r}^2 + 1}{k_{1r}^2 - 1} + p_i \frac{k_{2r}^2 + 1}{k_r^2 - 1} - \mu\left(p_i \frac{2 - k_{2r}^2}{k_r^2 - 1} - q\right)\right] \tag{38}$$

$$u^r{}_{B_2} = \frac{b_r}{E}\left[q \cdot \frac{k_{2r}^2 + 1}{k_{2r}^2 - 1} + p_i \frac{k_{2r}^2 + 1}{k_r^2 - 1} - \mu\left(p_i \frac{2 - k_{2r}^2}{k_r^2 - 1} - q\right)\right] \tag{39}$$

The dimensions for manufacturing of shrinked cylinders (Fig. 2) are calculated with next formulas:

a) for the inner cylinder: - the internal radius: $r_{1i} = a$ and the external radius: $r_{1e} = b_r + u^r{}_{B_1}$.

b) for the outer cylinder: - the internal radius: $r_{2i} = b_r - u^r{}_{B_2}$ and the external radius: $r_{2e} = c_r$.

The real tightening of the assembly is:

$$\Delta^r = u^r{}_{B_2} + u^r{}_{B_1} = b_{1\,\mathrm{r}} - b_{2\,\mathrm{r}} \tag{40}$$

and the real shrink pressure:

$$q = \frac{E \cdot \Delta^r}{2b_r} \cdot \frac{(k_{1r}^2 - 1) \cdot (k_{2r}^2 - 1)}{k_r^2 - 1} \tag{41}$$

This dimensioning problem of shrinked assembly was solved using an original mathematical and computational algorithm [3] written in Maple 12 software, which allows determination of optimal values for cylinders dimensions used in shrinked assembly, or determination of their dimensions in mandatory condition. Let's consider the following input data for the cover calculation:

- internal radius: a = 100 mm; - pressure on internal surface having variation of temperature corresponding to isocore transformation: $p = 0.10233 \cdot T$; - material: AISI 4340, with the following characteristics: σ_a = 710 N/mm², E = 2.05 10^5N/mm², μ = 0.28125; - the radius of the internal surface a = 100 mm. After numerical calculation the following results are obtained: - external radius of outer ring: c = 105.5 mm; - external radius of inner ring: b = 103.8 mm; - shrinkage pressure: q = 3.75 N/mm². The cover stresses and deformations were analyzed with SolisWorks 2012 software [35] considering the temperature variation between T = 0 ^{0}C,.., +55 ^{0}C.

Because the gas tank has axial constructive symmetry (Fig. 6), the study can be made on ¼ from initial model (Fig. 7) considering the corresponding: loadings, boundary conditions and symmetry on the contours. The constructive symmetry conditions are applied on the perimeter of surfaces S_3 and S_4 (Fig. 7), and the boundary conditions on the surfaces S_1 and S_2.

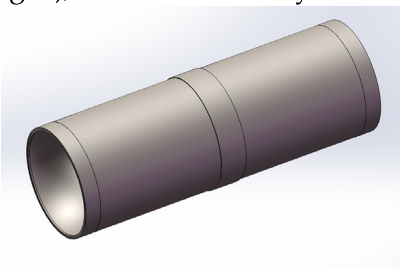

Figure 6. Side cylinder cover shrinked with 3 rings

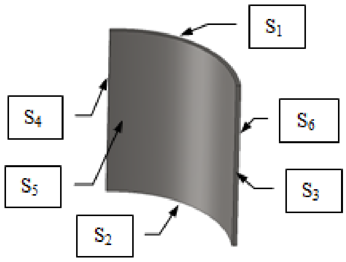

Figure 7. ¼ from the model of side cylinder cover

The loadings applied to the cylinder cover are: - the pressure p depending on the temperature according to the isocore transformation (Table 1), on surface S_5 and the atmospheric pressure on the external surface S_6; - the thermal loading on surfaces S_5 and S_6 with T = 0^0 C, …, +55^0 C.

Table 1. Variation with temperature and pressure of average von Mises $\sigma_r(T)$ stresses and linear resultant deformations $u_r(T)$ on real model of shrinked cylinder cover

No.	T [^{0}C]	p [N/mm]	σ_r [N/mm²]	u_r [mm]	No.	T [^{0}C]	p [N/mm]	σ_r [N/mm²]	u_r [mm]
1	0	27.90	702.17	0.669	7	30	31.00	649.68	0.597
2	5	28.40	691.55	0.659	8	35	31.50	636.90	0.581
3	10	28.90	680.90	0.640	9	40	32.00	615.24	0.557
4	15	29.50	670.30	0.628	10	45	32.50	606.95	0.546
5	20	30.00	659.70	0.614	11	50	33.00	594.23	0.531
6	25	30.50	649.10	0.600	12	55	33.60	606.90	0.546

For the side cover, the spatial distributions corresponding to average values of resultant linear deformation $u_r(T)$ and resultant von Mises stress $\sigma_r(T)$, for temperatures of T = 0 ^{0}C si T = 55 ^{0}C are shown in Figs. 8 and 9. The variation laws of $\sigma_r(T)$ and $u_r(T)$, obtained by interpolation with the least square method, based on data from Table 1 are as follows:

$$\sigma_r(T) = 699{,}644285 - 1{,}32001 \cdot T - 0.03812454 \cdot T^2 + 0.00052487956 \cdot T^3 \quad (42)$$

$$u_r(T) = 0.66720146 - 0.001817545 \cdot T - 0.00005106 \cdot T^2 + 0{,}747992748 \cdot 10^{-6} \cdot T^3 \quad (43)$$

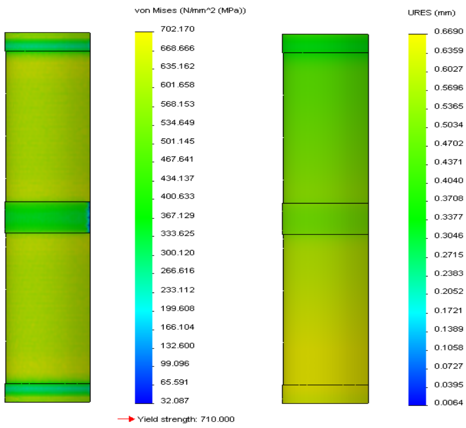

Figure 8. The resulting linear deformation and von Mises effort for real constructive solution at T = 0^0C

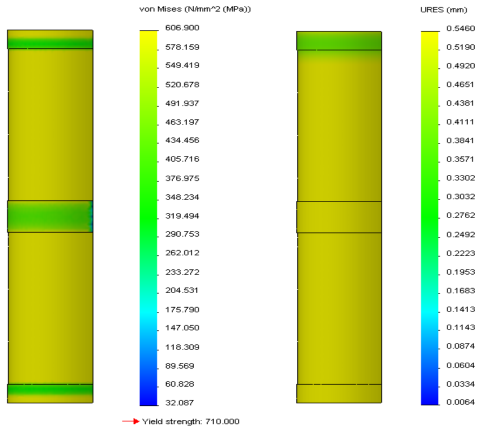

Figure 9. The resulting linear deformation and von Mises effort for real constructive solution at T = 55^0C

The graphics corresponding to: the variation of pressure with temperature (from Table 1), the resultant von Mises stress and the resultant linear deformation, are shown in Figs. 10-12.

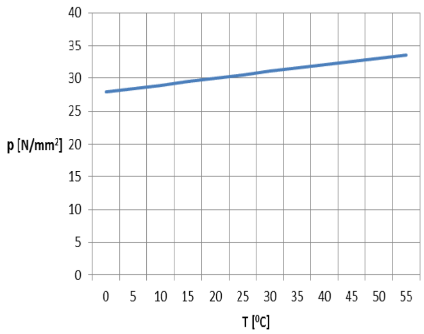

Figure 10. Variation of p (T)

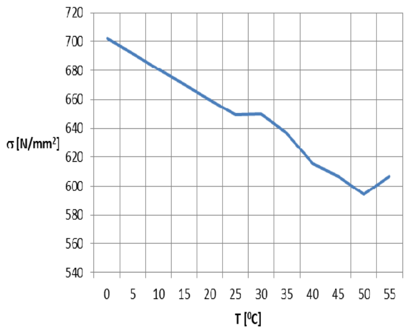

Figure 11. Variation of σ_r (T)

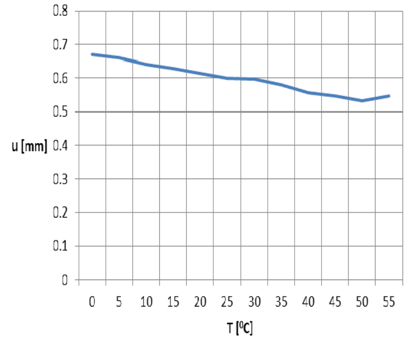

Figure 12. Variation of u_r (T)

4. CONCLUSIONS

This method allows the dimensioning of shrinked rings assembly from the ends of pressurized tank using numerical programs special made for this purpose, written in Maple 12, by adopting of convenient constructive dimension of the assembly. Thus, by imposing the dimensions of internal ring equal with: a = 100 mm and b = 103.8 mm, together with the internal pressurization of the gas tank, corresponding to isocore transformation from Table 1, for the material AISI4340, within limits of temperature variation T = 0...55°C, it results the external dimension of outer ring as c = 103.8 mm and a shrink pressure of q = 3.75 N/mm², to which assure an elastic deformation on shrinked surface during service.

On the other hand, it was obtained the total thickness of the tank wall as s = 3.8 mm, for which we have a resultant maximum von Mises stress of $\sigma_{rez\ (T\ =\ 0\ °C)}$ = 702.17 N/mm² which is under the admissible value of the material σ_a = 710 N/mm² and a maximum resultant linear deformation of $u_{rez(T\ =\ 0\ °C)}$ = 0.669 mm. The resultant stress is decreasing when the temperature increase, having a minimum of $\sigma_{rez(T= 50\ °C)}$ = 594.23 N/mm² at the same temperature of T = 50⁰C (Table 1).

Mounting of the shrinked rings at the ends of the pressurized gas tank has the advantage of reducing the side cover thickness in these areas by overtaking the supplementary stresses due to bending moment generated by hemispheres covers from the ends of the tank. From the analysis of these results, it can be seen that the proposed dimensioning method is an efficient method.

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