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LIGHT WEIGHT METAL AND STEEL ALLOY NATURAL GAS CYLINDER: NUMERICAL AND ANALYTICAL STUDY

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Abstract:
Type 2 cylinders are made of an inner steel alloy wrapped with a composite fiber system of several layers. However, the composite layer does not have radial resistance. A light weight metal could be used as replacement for the fiber composited layers. Thus a gnv container formed by an inner steel alloy shell and an outer aluminum cover is proposed. The elasticity solution for the contact stresses between a pressurized steel alloy cylindrical shell and the outer aluminum cylindrical shell is developed. A failure criterium is proposed to form a set of design parameters in terms of a safety factor. Analytical results are compared to the finite element solution.

Keywords: *natural gas, finite element method, cylindrical shell, failure criterium*

1. INTRODUCTION

High pressure cylinders for storage of natural gas in automotive vehicles are regulated under ISO 11439 standard. There are four storage cylinder types. Type 1 cylinder is the most commonly used. Type 1 cylinder made of alloy steel. Type 2 cylinders are a metal cylinder involved by a fiber composite.

Type 1 cylinder is made of steel alloy and normally has thickness close to 10 mm, enough to hold an internal natural gas pressure of 20 MPa. Normally it is the cheapest of the four types of cylinders. However, they are the heaviest among the cylinders.

A type 2 cylinder is made of steel alloy and has thickness around 50% of similar type 1 metal cylinder. They are more expensive than similar capacity type 1 cylinders. It has a metal body with a composite fiber system wrapped along the circumferential direction forming several layers. The angle of winding the reinforcement layers vary from 0^0 to 90^0 with respect to the longitudinal direction of the cylinder.

The material used to make the composite as well as the angle and winding of the fibers over the metal body are very important. It needs to be done carefully in order to achieve the aimed results. In general, the total thickness of the composite layers is approximately the same as the metal thickness. Furthermore, the composite fiber reinforcement does not have any radial resistance.

For the mentioned reasons, let us consider an alternative to type 2 cylinders, assuming a replacement of the composite fiber system by a light weight metal of similar thickness. The internal cylinder then becomes a steel alloy surrounded by another cylindrical shape, a light weight and cost effective metal. It could bring the benefit of simpler fabrication and supply the assembly with some extra radial stiffness.

2. MATERIALS

Let us consider a small cylinder having a 30L hydraulic capacity. The inner cylinder will be made of steel alloy involved by an aluminum cylinder, having a small gap between the external steel surface and the internal aluminum surface. The gap is expected to account for some circumferential imperfections of the internal and external cylinders. Table 1 shows the mechanical properties of the 4130 alloy steel after heat treatment.

Table 1: Mechanical Properties of AISI 4130

E (GPa)	n	S_y (MPa)	S_u (MPa)	g (kN/m^3)
207	0.292	815	906	76.5

E = Modulus of Elasticity
n = Poisson ratio
S_y = Yield tensile stress
S_u = Ultimate tensile stress
g = Unit weight

Table 2 shows the mechanical properties of the aluminum alloy.

Table 2: Mechanical Properties of ALUMINUM

E (GPa)	n	S_y (MPa)	S_u (MPa)	g (kN/m^3)
71.7	0.333	169	324	26.2

E = Modulus of Elasticity
n = Poisson ratio
S_y = Yield tensile stress
S_u = Ultimate tensile stress
g = Unit weight

Figure 1 represents the cad model of a natural gas cylinder with the aluminum covering the central cylindrical region. Let us consider a White-Martins model 30244850, which has 30 liters of hydraulic capacity, external diameter of 224 mm and total length of 850 mm as a reference case. Type 1 steel alloy cylinder is produced normally with a thickness approximate of 10 mm. A lighter and more expensive version of this model has an average 20% reduced thickness, employing more expensive heat treated steel alloy.

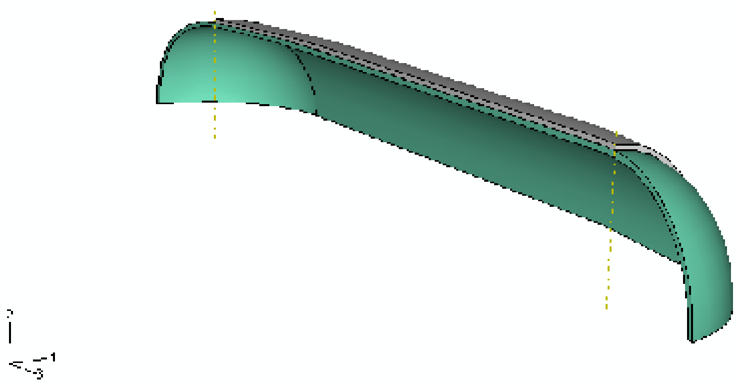

Figure 1: Alloy steel-aluminum shell section.

3. ANALYTICAL MODELLING

Elasticity theory predicts the axial, circumferential and radial stress distributions of a thick pressurized cylindrical shell, which will be used as a design estimative for the stresses. It can be shown that the tangential stress s_q, radial stress s_r and the axial stress s_a have magnitudes given by (Timoshenko, S, 1973)

$$s_q = \frac{a^2 p_i - b^2 p_o}{(b^2 - a^2)} + \frac{a^2 b^2 (p_i - p_o)}{r^2 (b^2 - a^2)} \tag{1}$$

$$s_r = \frac{a^2 p_i - b^2 p_o}{(b^2 - a^2)} - \frac{a^2 b^2 (p_i - p_o)}{r^2 (b^2 - a^2)} \tag{2}$$

$$s_a = \frac{p_i \bar{r}}{2t} \tag{3}$$

where $\bar{r}$ is the average radius of the hemisphere, t is the wall thickness, (a,b) are the internal and the external radius and (p_i, p_o), are the internal and the external pressure values, respectively.

The radial component of the cylindrical deformation u_r is given by:

$$u_r = \frac{1-n}{E} \frac{a^2 p_i - b^2 p_o}{(b^2 - a^2)} r + \frac{1+n}{E} \frac{a^2 b^2 (p_i - p_o)}{(b^2 - a^2)} \frac{1}{r} \tag{4}$$

3.1. Failure Criteria

The failure criterion that we will be used is an equivalent stress to be compared to the yielding stress of the material. The equivalent stress is the shear stress t_0 on the octahedral plane and is given by:

$$t_0 = \frac{1}{3}\sqrt{(s_a - s_q)^2 + (s_q - s_r)^2 + (s_a - s_r)^2} \tag{5}$$

It is assumed that the material fails when the equivalent stress t_0 reaches a critical value, t_0^c which corresponds to:

$$t_0^c = \frac{\sqrt{2}}{3} S_y \tag{6}$$

where S_y is the yielding stress of the material.

The safety factor h_0 becomes:

$$h_0 = \frac{t_0^c}{\bar{t}_0} \tag{7}$$

where $\bar{t}_0$ is an admissible octahedral stress.

In general a natural gas container for automotive use is constructed from a combination of a cylindrical shell, the main body, with two closing cap shells. The junction between the cylindrical part and the end cap may be done by welding. Figure 1 shows the reference axes, as well as the cad model used for the problem. The hydraulic capacity is determined by the internal volume V of the cylindrical shell portion, which has an internal radius a, length L, plus the volume $\bar{V}_0$ of both end caps.

The shell pressurization may be considered to be a quasi-static process ending when an operational pressure p_i is reached. The principal stress components presented above can be written in terms of the wall thickness as:

$$s_q = a\, p_i - b\, p_0 + \sqrt{ab}\,(p_i - p_0)\frac{a_0{}^2(1+l_0)}{r^2} \tag{8}$$

$$s_r = a\, p_i - b\, p_0 - \sqrt{ab}\,(p_i - p_0)\frac{a_0{}^2(1+l_0)}{r^2} \tag{9}$$

$$s_a = a\, p_i - bp_0 - \sqrt{ab}\,\frac{P}{p\; a_0{}^2(1+l_0)} \tag{10}$$

where the multipliers a and b are functions of $l_0 = \frac{t}{a}$. The dimensionless thickness coefficient l_0 is defined by the ratio between the wall thickness t and the internal radius a. The expressions above become more suited to design purposes. The parameter α and β correspond to:

$$a = \frac{1}{(1+l_0)^2 - 1} \tag{11}$$

$$b = \frac{1}{1-(1+l_0)^{-2}} \tag{12}$$

being p_i the internal pressure and p_o the external one. Internal pressure is set by the available pressure in the filling line, whereas the external one is in general the atmospheric pressure. The axial force P in most of the assemblies is never applied. Temperature is supposed to be constant. The cylinders manufacturers state that their products have a safety coefficient over 2.5 (White-Martins, 2007, Rama Cylinder, 2008)

Manufacturers state that a cylinder of 30 liters of hydraulic capacity has average wall thickness in the range of $8\,\mathrm{mm}$ to $10\,\mathrm{mm}$.. For the modeling, let us consider a lighter weight cylinder with thickness $t = 6\,\mathrm{mm}$ which will be covered with an aluminum outer shell. Let us assume that the steel cylinder has an external radius $b = 112\,\mathrm{mm}$ and an internal radius $a = 106\,\mathrm{mm}$. Table 3 shows the stresses at the internal and external surfaces as well the safety coefficient under the operational pressure $p_i = 20$ MPa.

Table3: Stresses and Safety Coefficient for the Steel Cylinder.

Stress	s_a (MPa)	s_r (MPa)	s_q (MPa)	h
Internal	**181.7**	**-20.0**	**363.6**	**2.35**
External	**181.7**	**0.0**	**343.6**	**2.60**

Let us consider that an aluminum cylinder is introduced with a gap between the external and internal shell. Let us consider that the external surface of the steel cylinder effectively contacts the aluminum internal surface when the internal pressure reaches a given value. For example, if the contact is established once 65% of the total pressure is reached, then. Eq. (4) shows that the radial displacement of the external steel surface is $u_r = 0.15\,\mathrm{mm}$, and the internal pressure is $p_i = 13.5\,\mathrm{MPa}$.

For a given value of hydraulic capacity of the container, after choosing the volume of the end caps, the value of cylindrical shell length L can de defined. If a specific material is considered and a value for the safety factor

$h_y^0 = \frac{t_0^c}{\bar{t}_0}$ taken, being $\bar{t}_0$ the admissible octahedral stress, then Eq. (6), written in terms of l_0, renders a polynomial expression. The design of the gnv cylinder for a specified safety factor has turned into the finding of roots for the polynomial expression:

$$P(l_0) = l_0^4(2-z) + l_0^3(48-8z) + l_0^2(73-8z) + 48l_0 + 12 \tag{13}$$

where $z = 3(\frac{\bar{t}_0}{p_i})^2$.

For commercial cylinders, Eq. (13) may also be used to find out the safety factor since the geometry is already defined. In fact, for the steel cylinder, with material properties given by Table 1, with internal radius $a = 106$ mm, internal pressure $p_i = 20$ MPa, $L = 596$ mm for the cylindrical part, with a thickness $t = 8$ mm it turns out a safety factor $h_y^0 = 3.09$.

3.2. Shell in Shell Design

The idea of cylinders that carry the same natural gas volume, that have the same safety factor for the critical section, but yet have a lighter weight structure has immense appeal. Keeping steel as the internal cylinder material, for its low cost, and aluminum for the external one, for its low weight may be a good starting point.

A few criteria may be considered in the construction of the solution, as a procedure tool. First, the container needs to have the same hydraulic capacity, which is $V = p\ a^2 L + V_0$ As the same operational pressure p is used, volume split between shells may use this term as control. If the inner shell has a material volume v and the outer shell, separated from the internal one by an initial radial gap g_r, has a volume v', supposition that the composed shell fits in the same compartment volume leads to:

$$t' = \frac{t_0 - t}{1 + \frac{t}{a} + \frac{g_r}{a}} \tag{14}$$

which may be solved to define thickness t' of the external shell as long as the inner shell thickness t and radial gap g_r are defined up front.

Let us assume that the gap is such that there is no contact between the inner and the outer shells until a certain level of the operational pressure is reached, say $0,8p_{\max}$. In addition, that we are using steel for the inner shell, and an allowable octahedral stress $\bar{t}_0$ or safety factor similar to the commercial containers. Eq. (13) allows us to find the corresponding thickness. In fact, the solution of Eq (13) produces a new $l' = \frac{t'}{a}$ which corresponds to the new thickness t'. Evidently if another construction gap value is chosen, another closing pressure would be required, but again we would use the same procedure. For the adopted values, it turns out that $t = 6,265$ mm.

In order to solve Eq. (14), initial value of radial gap has to be found. The radial displacements at the outer surface of the inner thick cylinder follow from Eq. (4):

$$u_r = f\frac{a^2}{b^2 - a^2} p_i r + j\frac{a^2}{b^2 - a^2} p_i \frac{b^2}{r} \tag{15}$$

being $f = \dfrac{1-n}{E}$; $j = \dfrac{1+n}{E}$ and $b = a + t$. Introducing the pressure value and setting $r = b$ the value of the radial gap is found. For the dimensions used, g_r equals $0.17\ \mathrm{mm}$, approximately.

Once the radial gap between the inner and outer shell disappears, a contact pressure between them will take place. The magnitude of the contact pressure can be found by the condition of equal radial displacements for the inner and outer shells. Denoting the contact pressure at the common interface by p_c , the individual radial displacements at the interface are:

$$u_r = f\frac{b}{b}[a^2 p_i - b^2 p_c] + j\frac{b}{b}a^2(p_i - p_c) \tag{16}$$

whereas for the outer shell, inner radius, $a' = b + t + g_r$

$$u'_r = f'\frac{a'}{a'}a'^2 p_c + j'\frac{a'}{a'}b'^2 p_c \tag{17}$$

both expressions only valid for internal pressures in the range $0{,}8p \le p_i \le p$. Equating both radial displacements leads to the expression for the contact pressures:

$$\frac{p_c}{p_i} = \frac{ba(f + j)}{\{b[fb + ja] + a'[f'a' + j'b']\}} \tag{18}$$

which can be transformed into:

$$p_c = \frac{\frac{b}{b}a^2(f + j)p_i}{\frac{a'}{a'}(f'a'^2 + j'b'^2) + \frac{b}{b}(fb^2 + ja^2)} \tag{19}$$

Let us assume that at this point the gap between the steel cylinder and the aluminum cover is overcome, i.e., the contact between the steel internal cylinder and the aluminum external cylinder starts. Now we can set the dimensions for aluminum cover with the internal radius $a' = 112.17\ \mathrm{mm}$, and the external radius $b' = 116.17\ \mathrm{mm}$.

The steel-aluminum section will have a total thickness similar to a type 1 steel cylinder, but lighter weight. The radial displacement given by Eq. (4) at the contact face should be the same for the external steel surface and internal aluminum surface. It allows finding the theoretical contact pressure, which for the case is $p_c = 3.39\ \mathrm{MPa}$.

Table 4 shows the stresses at the internal and external surfaces of the steel cylinder as well the safety coefficient under an internal pressure $p_i = 20$ MPa and an external contact pressure $p_c = 3.39\mathrm{MPa}$.

Table 4: Stresses and Safety Coefficient for the Cylinder under Contact Conditions.

Stresses	s_a (MPa)	s_r (MPa)	s_q (MPa)	h
Internal	158.0	-20.0	298.4	3.04
External	158.0	-3.4	281.8	3.26

Table 3 and Table 4 show the safety factors involved with and without the aluminum outer shell. We can observe the variation on the safety coefficient due to the presence of the aluminum cover favors the increase on the safety.

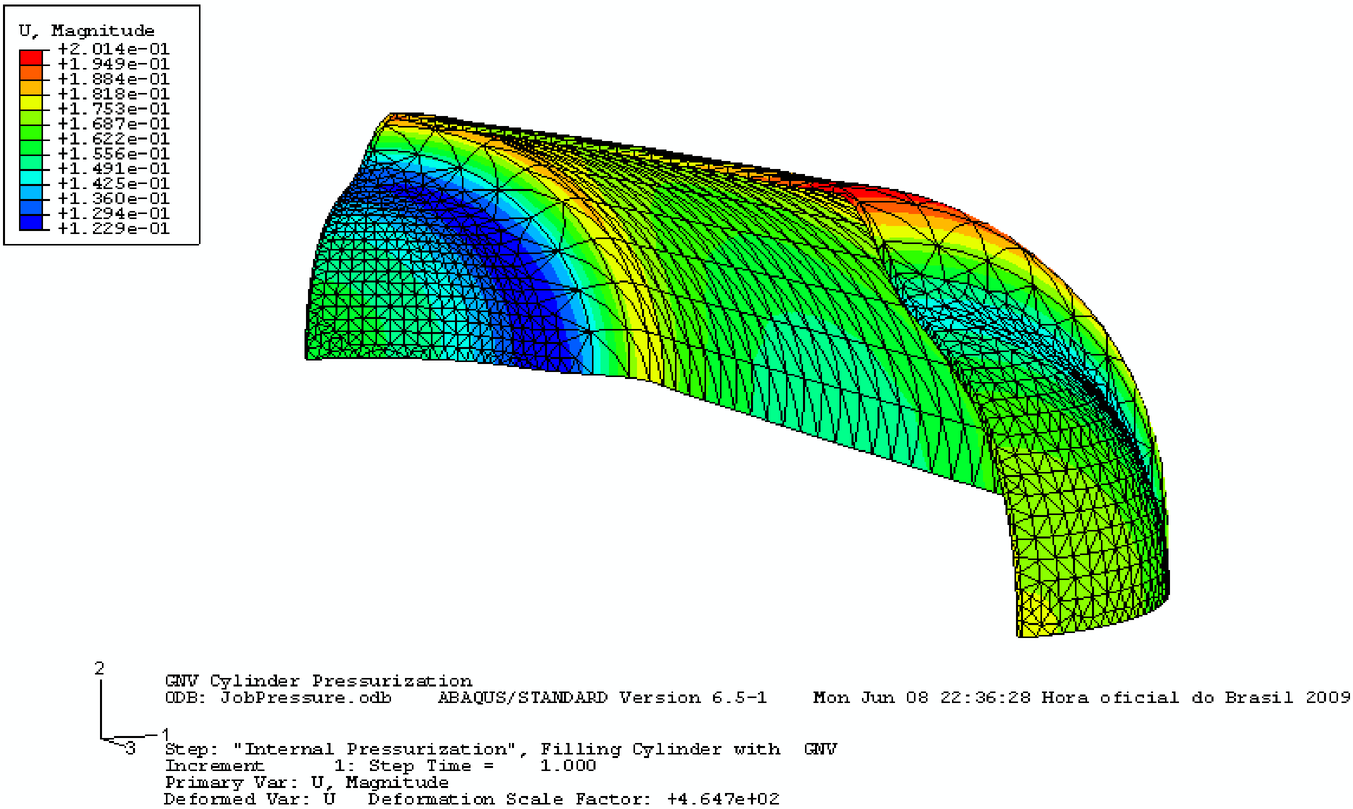

Figure 2: Aluminum and steel alloy container: radial displacement magnitudes

The theoretical results presented above can be verified numerically using finite element software. Figure 2 shows that the radial displacements are mainly concentrated on the central cylindrical part and the magnitudes are similar to the theoretical values, except on the region of curvature discontinuity.

It can be observed in Figure 3 that the maximum finite element principal stress of the steel cylinder is correctly predicted numerically.

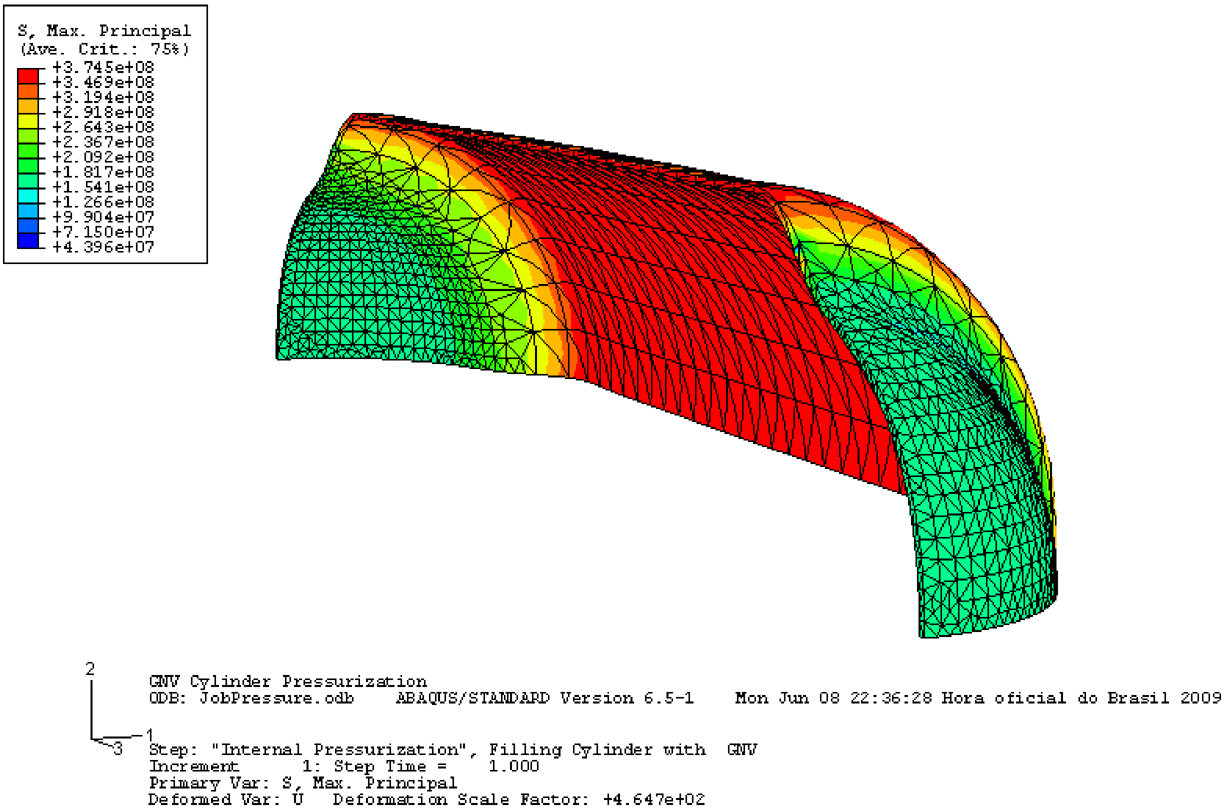

Figure 3: Principal stresses on steel alloy – aluminum gnv container.

4. RESULTS AND CONCLUSIONS

The analytical and numerical comparison permits us to conclude that the combination of a steel alloy and an aluminum cover results in a lighter container. In fact, in the example addressed, the outer shell made of aluminum had a $4\ \mathrm{mm}$ thickness. Since the weight per unit of volume of the aluminum is roughly speaking 30% of the alloy steel, the equivalent container would have approximately $7.2\ \mathrm{mm}$ of wall thickness, which means it is lighter then the best $8\ \mathrm{mm}$ cylinder for the same hydraulic capacity. The comparison takes into account the volume per unit length of the cylindrical region since the end caps are similar. We would find an equivalent steel cylinder with internal radius $a^* = 119\ \mathrm{mm}$ and external radius $b^* = 127\ \mathrm{mm}$. The corresponding safety factor would be around 2.8.

The results suggest that the composed section of two metals in fact has some advantages in terms of manufacturing as compared to fiber composite system.

5. REFERENCES

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