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ANALYZING THE EFFECT OF POLYMER INTRUSION AT INTERLOCKED CARCASS GAPS

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Abstract. *The interlocked carcasses are lightweight and have high radial resistance with negligible bend and axial stiffness, which are particularly important for multilayered tubular structures. They are used in flexible pipes and umbilical cables. At both applications, a polymeric layer must be extruded upon the carcass, to act as fluid barrier. The carcass profile has irregular shape, usually with a profile similar to an "S", and an inevitable consequence of the fabrication process the polymer can penetrate the emptiness between gaps profile. Usually this fact is not considered at the design stage. This paper addresses some possible issues caused by this model simplification. To evaluate the polymer effect, a finite element approach is implemented considering the main load condition, the hydrostatic external pressure, that the carcasses are submitted during the operational phase. A simple analytical model is also presented, which does not take this polymer effect into account, to evidence the non-conservative aspect of this assumption.*

Keywords: *interlocked carcass, umbilical cable, flexible pipe, HCR*

1. INTRODUCTION

Petroleum industry has been challenged to improve the offshore technologies to face the new requirements of deeper water offshore wells. The interlocked carcasses are presented in flexible pipes and umbilical cables with quite similar functions. The fluids are usually different, but the interlocked carcasses have to resist to external radial load, for instance, the external hydrostatic pressure. According to the standard ISO 13628-5 (2009) the carcasses must be designed to resist to the hydrostatic pressure lonely, even though some authors as Neto, *et al*. (2012) have been reported that other layers can also aid to resist to the external pressure. Based on ISO 13628-5 (2009) as a secure criterion, the load is defined as applied directed on the carcass of thermoplastic hoses of umbilical cables.

In this paper, umbilicals with thermoplastic hoses will be analyzed, but the study can be extended to flexible pipes with few additional considerations. In Brazil, thermoplastic hoses are preferred than steel tubes, mostly by it has lower costs and facility to install. The analytical approach is used, for instance, in Souza (2006) and Vignoli (2014), modelling flexible pipes under external pressure. It was proposed the use of the Timoshenko's equation for a thin ring Timoshenko (1941), with an empirical correction factor, based in a large number of experimental results, to take the carcass geometrical irregularities into account. Analytical models are easier to be formulated with fewer approximations for pipes than for umbilicals because the pipes are formed by a set of concentric layers, in contrast to umbilicals that usually have complex configuration cross sections.

Based on numerical approach, Zhao et *al*. (2012) presented a numerical study to quantify the influence of a fabrication problem of the carcass profile that changes its moment of inertia. The moment of inertia influence for the structural use of the carcasses is also discussed at Vignoli, et *al*. (2014). Neto et *al*. (2009) presented two papers in this same subject Neto and Martins (2013) and Neto, et *al*. (2012). In Neto and Martins (2013) the authors showed a carcass stress analysis with the influence of initial ovalization as well as plastic effects. Using the finite element software ANSYS as reference, they also show that the stress analysis error for not taking the carcass pitch into account is relatively small. Indeed the helix angle is usually less than 4º, even for thermoplastic hoses. In Neto et *al*. (2012), a numerical study about the wet and dry collapse of flexible pipes was done to evidence the influence of each layer at the structure collapse. Sato, et *al*. (2013), presented a numerical study, using the one quarter symmetry, and taking into account all the layers and compared these results with real scale tests. The main results are related to ovalization, but the use of von Mises criterion for analyzing polymeric layer and few mesh elements through the carcass thickness, can make the model results rather inaccurate.

2. CARCASS MODEL

For this study, the carcass profile and dimensions are presented in Fig.1 and Table 1, respectively. It is important to use the real carcass profile, avoiding excessive geometric simplifications, to estimate the moment of inertia, as in Vignoli, et *al*. (2014). Interlocked carcasses used in flexible pipes have small gap. Once the profile is defined during the design step, the gap can be defined according to the stiffness per unit of length necessary for the project.

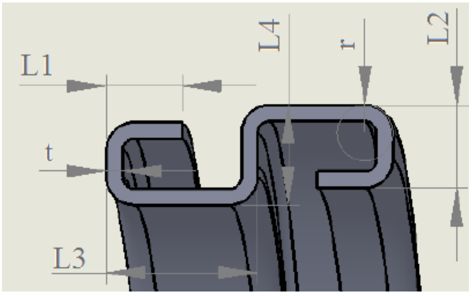

Figure 1. Carcass cross section principal dimensions.

Table 1. Carcass cross section dimensions (Vignoli, 2014)

dimensions	(mm)
L_1	3.25
L_2	2.56
L_3	6.5
L_4	3.3
t	0.7
r	1.28

The profile presented in Fig.1 is used in HCR (High Collapse Resistance) hoses for umbilical cables; the profile of the carcass used for flexible pipe has a quite different shape.

3. ANALYTICAL MODELS

When external pressure is applied on an interlocked carcass, the polymer deposited between the carcass profiles generates a nonlinear behavior on the structure, complicating the analytic modeling.

Usually for large bore hoses the elastic buckling can be a issue. Moreover, for smaller bores, like those used in thermoplastic hoses, the pressure necessary to allow the first mode of buckling is bigger than the necessary to yielding the carcass, indicating that the plastic collapse can occur firstly. Because of these reasons, it is important to emphasize that the both, stress and buckling analysis must be executed.

As proposed by Neto and Martins (2013), the helix angle will be not considered. For a distributed load q (pressure per unit of length) applied, for the classical result for thin rings, of radius R, without initial ovalization, Crandall, et *al.* (1978), the acting forces N are

$$N = qR \tag{1}$$

For a perfect ring, submitted to a constant external pressure, there is no internal bend moment acting in its cross section for any angle, so just the normal force must be considered. The von Mises stress for this model is $\sigma_{vM}^{(c)}$ is:

$$\sigma_{vM}^{(c)} = \left| \frac{E_c A_c}{(EA)_{eq}} \frac{N}{A_c} \right| \tag{2}$$

Where E is the Young modulus, A is the cross section area, the indexes c and eq means, respectively, carcass and equivalent. Moreover, for a perfect ring, submitted to constant external pressure, analyzing critical unit load for the first buckling mode considers only the bending effects can be found, for instance, at the classical approach presented by Timoshenko and Gere (2009). This model will be mentioned from now on as TB (Timoshenko-Bend) and the critical unit load p_{cr} can be written as

$$p_{cr} = \frac{3(Ei)_{eq}}{R_{med}^3} \tag{3}$$

Where R_{med} is the medium radius and i is the second moment of area per unit of length.

A general formulation is proposed by McNamara, et *al*. (2001), includes the shear and axial effects, in addition to the bend loading, will be cited for now on as MC (McNamara-Complete), and the critical unit load at first buckling mode p_{cr} is

$$p_{cr} = \frac{1}{B_c} \frac{3\left(a_{sf}GA\right)_{eq}\left(EA\right)_{eq}\left(EI\right)_{eq}}{R_{med}\left[4\left(EA\right)_{eq}\left(EI\right)_{eq} + \left(a_{sf}GA\right)_{eq}\left(EI\right)_{eq} + \left(EA\right)_{eq}\left(a_{sf}GA\right)_{eq}R_{med}^2\right]} \tag{4}$$

Where G is the shear modulus, B_c is the carcass width, a_{sf} is a dimensionless shape factor correction for the shear strain energy by a transverse force. Despite (4) takes the axial and shear effect into account, the additional issue of the shape factor is a drawback of this model, because the carcass irregular shape, its evaluation becomes quite complicated.

4. FINITE ELEMENT MODELS

The finite element developed in this paper is based in Neto, et *al*. (2009), which proved that the simplified model (one quarter of a carcass and not considering the pitch) is efficient to simulate the carcass submitted to operational loads. Fig. 2 shows the geometry used in this simulation.

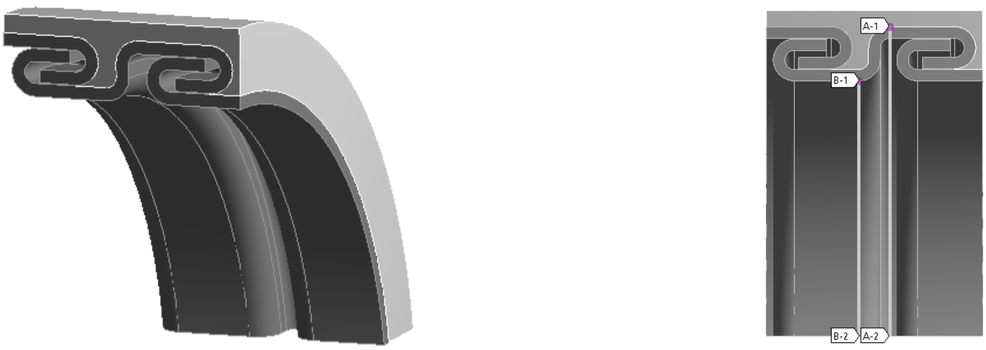

Figure 2. (a) the geometry perspective view and (b) two paths created to compare the results.

Note that the polymer extruded is represented on the carcass geometry. To compare the numerical results with the analytical models, two paths were created, as shown at Fig. 2.b. An uniform pressure of 10 MPa is applied at the external surface of the assembly (hydrostatic load). All the polymer and carcass faces that have cut to make the one quarter model are prevented to have any displacement on the direction of the respective face.

The mesh generated is presented in Fig.3. The carcass elements are hexagonal, as the carcass thickness is constant and the polymer elements are tetragonal because its irregular shape. Higher order elements were used to improve the geometric description and the performance of the utilized contact formulation.

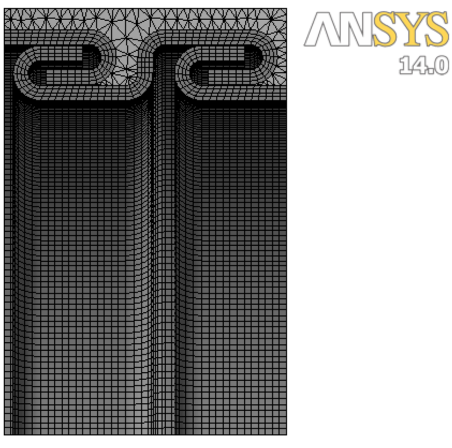

Figure 3. Mesh used on the finite element model.

The contacts are set as "No Separation", where the faces in contact are frictionless to slip but cannot have normal relative displacement. Another contact type, the "Frictionless", was also experimented to evaluate the hypothesis of disregard normal relative movement between faces.

For this type of contact, the Augmented Lagrange formulation was used and the polymer's mesh was refined because penetration issues. Comparing "No Separation" with "Frictionless" strategies, no significant results improvement have been obtained by the utilization of the last contact type that compensate the increase of associate computational cost. Note that access the real friction coefficient between the contacts is unrealistic, because many variables can affect the friction coefficient, as the weather conditions during the extrusion of the polymer and other parameters that cannot accessed at all. Because these issues, the MPC formulation (Multi-Points Constraint) is adopted in this numerical simulation. This contact definition couples the displacement of the contact faces, detected by nodes, disallowing the penetration and results for faster convergence Lee (2012). By applying the equivalent boundary condition avoids the contact penetration and the need of a friction coefficient. The results are presented at Fig. 4.

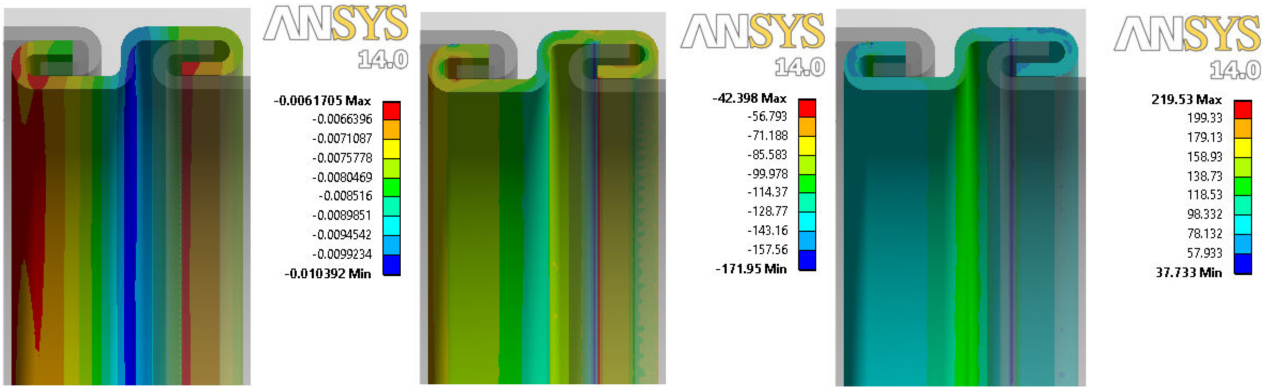

Figure 4. Results of external pressure loading: (a) radial displacement, (b) normal hoop stress, (c) the von Mises stress.

Note that the critical part is along the gaps between the carcasses, where the polymer penetrates and results in higher stress levels. To quantify the extrusion effect, a similar geometry was created at Fig. 5, but without the polymer extruded between the carcass profiles. The same conditions were adopted (load, contact, boundary) for this new model.

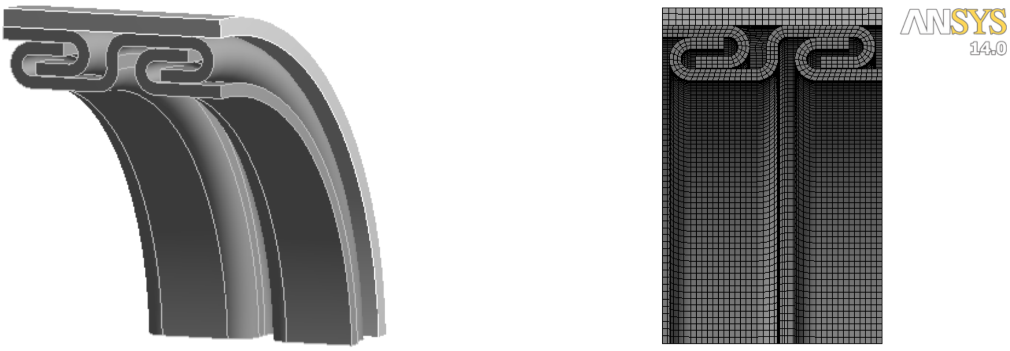

Figure 5. Simplified F.E. model.

The intention of this simulation is to show the error generate by disregard polymer extrusion at the carcass profile, which is an inevitable consequence of the fabrication process. The load and contact settings are kept the same to the case of polymer intrusion to enable a direct comparison. The results of the simplified F.E. model are presented at Fig. 6.

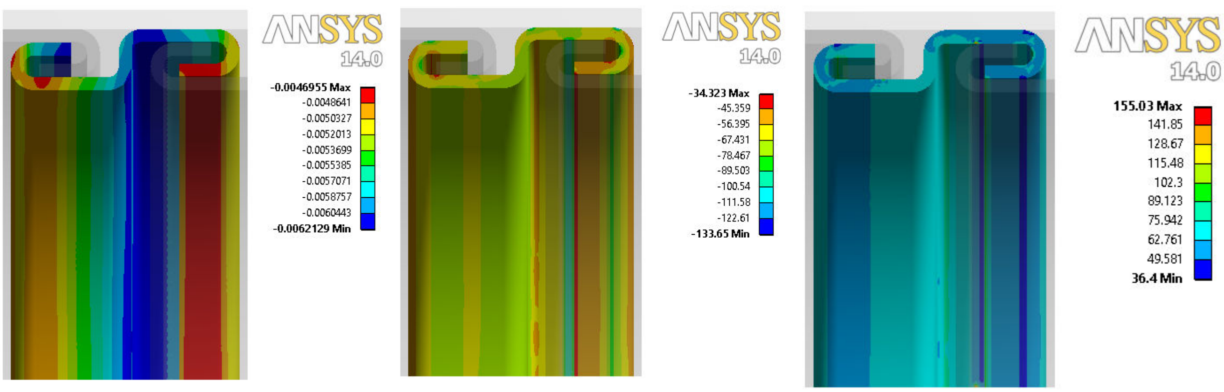

Figure 6. Results of external pressure loading (with simplified geometry): (a) radial displacement; (b) normal stress; (c) von Mises stress.

Comparing Figs. 4 and 6, it becomes clear that the simplified geometry implies in a not conservative stress distribution at the area where the polymer is extruded than for simplified one.

Focusing the comparisons, mainly at the von Mises stress, which represents the failure criterion, it is easy to realize that where is the higher stress level for the complete geometry of Fig.4.c, coincides to the part where the polymer amount is "deposited". On the other hand, at Fig.6.c, the stress level is lower because the polymer and carcass are in contact just on the top of carcass.

5. ANALYTICAL AND FINITE ELEMENT MODELS COMPARISON

In this section the results of both models, analytical and numerical, are compared. The commercial software Mathcad was used to implement the TB and MC models expressions.

For the external pressure load the results are separated in two parts, an elastic buckling study and an stress distribution analysis. The first one is presented in Table 2 and shows a great agreement between the numerical value for the first critical pressure and analytical models TB and MC, quantified through direct comparison Finite Element model (FE), used as reference, through the use of the Error column.

Table 2. Buckling results comparison.

Model	Critical Pressure [MPa]	Error [%]
TB	203.7	2.6
MC ($\alpha_{sf} = 0.5$)[1]	190.3	8.9
MC ($\alpha_{sf} = 1$)[1]	196.5	6.0
MC ($\alpha_{sf} = 2$)[1]	199.7	4.5
FE	209.1	-

[1]a_{sf} is a dimensionless shape factor correction

It is possible to conclude that the classical approach for a thin ring, which takes into account only the bending effect (TB), has a better performance than the more complex equation proposed at MC model, which has in the calculation of the real shape factor value a_{sf} an extra difficulty. For instance, at Table 2 it was used three values, 0.5, 1 and 2, just to get a fast comparison.

It becomes necessary to analysis the critical buckling pressure and the yield or plastic collapse, depending of the project criterion, as two different possible failure mechanism. For this carcass, using the dimensions presented in Table 1, the yield failure could happen before than the elastic buckling. Usually for flexible pipe carcasses, the buckling is the critical point, on the other hand, for thermoplastic hoses for umbilicals; the yield is more significant, because the predominant failure mechanism depends of how slender the ring is.

Table 3. Stress results comparison.

Model	Tangential Stress [MPa]	von Mises Stress [MPa]
Analytical (Eq. 2)	- 98.1	98.1
Complete FE (Inner Path)	- 116.1	109.0
Complete FE (Outter Path)	-117.9	94.2
Simplified FE (Inner Path)	-73.7	70.5
Simplified FE (Outter Path)	- 73.2	60.8

It shows that the analytical model stress distribution obtained results in an intermediary performance between Simplified and Complete FE models, which can be considered a quite unsafe result. For this case, the use of a finite element model is justified by the geometric complexity.

6. CONCLUSIONS

In this paper the effect of the polymer which penetrates the carcass profile gap as a consequence of the fabrication process was analyzed using FE tools. This fact is seldom considered during the design phase, which can induce non-conservative results. This problem can happen for interlocked carcass used in flexible pipe and in hoses for umbilicals and becomes even worse for HCR hoses. The bigger difference which influences directly in this result is the gap size that is larger for umbilical cables in comparison to flexible pipes. On the example used in this paper, an HCR profile, the error generated was up to 20% for the stress analysis. Thus is important to represent this polymer intrusion between the carcass gaps at the FE model, otherwise appreciable errors can be generated.

Simple analytical models were presented and show a good agreement with simulation. For elastic buckling, the error of the analytical predictions were all below 10% and for maximum von Mises stress, the error of the estimation was between 10% and 20%. In both cases the analytic models generates non-conservative values.

As a recommendation for avoid the excessive conservatism assumptions, an inspection program should be implemented at the manufactory site to desiccate carcasses samples after the extrusion process to get a better understanding about the level of polymer penetration in carcass gaps.

7. ACKNOWLEDGEMENTS

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9. RESPONSIBILITY NOTICE