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Computational assessment of the residual stresses of a wind turbine tower steel shell and their effect on its buckling capacity

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Abstract
As wind energy structures and especially wind turbine tower structures are becoming more popular over the years, their design has also evolved significantly. The increasing demand for more wind turbine towers has led to their increase in energy production and consequently to their height increase. This increase comes with many challenges for the mechanics of the tower itself as these towers now exceed heights of 140m. The most commonly used structural system for these towers is a cylindrical or conical steel shell tower which is highly vulnerable to residual stresses effects. This effect is more profound for inelastic buckling modes which are relatively common for wind turbine tower geometries. This paper presents ongoing work which is intended to lead to a computational assessment of the residual stresses for a wind turbine steel shell specimen and then study the effect on how residual stresses reduce the critical buckling load of the specimen.

1. General introduction – Thin-Walled Cylinders under bending

The problem of the stability response of thin-walled cylindrical shells under axial compression or bending moment has been defined as one of the most prominent problems of engineering mechanics. Thin-walled cylindrical members are used today across many disciplines, including lately the field of wind turbine tower structures. As the area of wind energy is growing rapidly in the last decades, the technology for more efficient thin-walled cylindrical (or slightly tapered) towers has been also growing.

Early work on thin-walled cylinders under compression by Koiter (1963, 1967) and Von Karman (1941) found that the main difficulty of the problem lies on the fact that there is a strong interaction between many simultaneous buckling modes. As mentioned in Houliara (2011) the problem of thin-walled cylindrical shells under compression has received significantly more attention than the problem of thin-walled cylindrical shells under bending moment. The present work is focusing on the stability response of these shells under bending moment, a problem which is closer to the simulation of the response of a tall wind turbine tower (the lateral wind effect load is higher than the compressive axial load, leading to a bending loading).

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The design of these shells has been dominated until today by the so-called knockdown factors (Hutchinson 2010). Figure 1 includes the knockdown factors recommendation by NASA (1965) and Seide (1960) based on the gathering of the experimental data up to that date. The figure plots the knockdown factor 'a' for the case of pure compression and for the case of pure bending as a function of the shell radius to thickness ratio R/t. The two equations used are (NASA 1965):

$$a_{bending} = 1 - 0.901(1 - e^{\phi}) \tag{1}$$

$$a_{axial} = 1 - 0.731(1 - e^{\phi}) \tag{2}$$

Where:

$$\phi = \frac{1}{16}\sqrt{\frac{r}{t}} \tag{3}$$

These knockdown factors are defined to be used as reducing the critical compressive stress which appears in the wall of the cylinder under compression. Essentially, they reduce the stress from the well-known equation:

$$\sigma = \frac{\mathrm{E}}{\sqrt{3(1-n^2)}}\left(\frac{t}{r}\right) \tag{4}$$

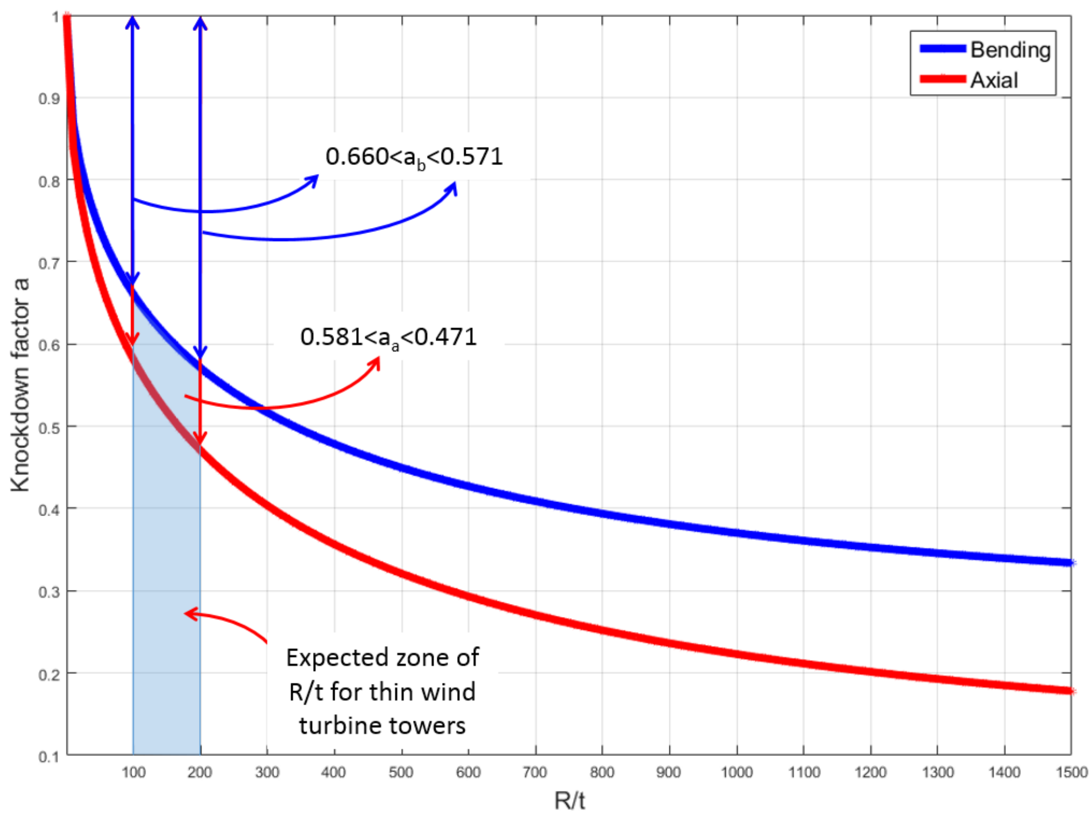

Figure 1: Knockdown factors adopted from Seide (1960) and NASA (1965).

It is clearly observed that thin cylindrical shells can experience a knockdown factor as low as 0.2 for pure compression and as low as 0.4 for pure bending (for values of R/t higher than 1200). A full derivation of Eq. 1 and Eq. 2 is included in NASA (1965).

In Fig. 1, the common range of wind turbine towers is also sketched. Although the literature in wind turbine tower geometries is rather scarce, the range of R/t for these thin shells is assumed by the author to be today in the range of $100<R/t<200$. This refers to the thinnest part of the towers (thinnest here does not refer to the lowest value of thickness, but to the lowest value of R/t). For these values of R/t, the expected knockdown factors are in the range of $0.66<a_b<0.571$ for pure bending and $0.581<a_a<0.471$ for pure compression. These low values demonstrate the sensitivity of the shells to all kinds of imperfections.

A note on shell imperfections here is necessary. The effect of imperfections is not limited to geometric imperfections only. As mentioned in Schmidt (2000), imperfections can be considered as "equivalent" geometric imperfections. This is the concept that has prevailed today in most shell design codes (including the Eurocode). Hence, the effects of other types of imperfections such as residual stresses, inhomogenities, anisotropies, boundary and loading inaccuracies (Schmidt 2000) are included in the 'equivalent' geometric imperfections.

The present paper includes the preliminary computational work by the author in an attempt to assess the effect of residual stresses in the stability response of thin-walled cylindrical shells under pure bending. It must be very clearly noted that this paper presents only the preliminary work and does not reach the intended assessment. The reader is kindly referred to future publications by the author for more results.

2. Buckling of thin cylinders under bending – Elastic Bifurcation Analysis

The work in this paper is focusing on a specific thin-walled cylindrical steel shell which consists of a part of a wind turbine tower. The length of this cylinder is 30m, its radius is 2m and the thickness is considered constant at 0.02m (R/t is 100).

First, an elastic bifurcation analysis is performed. The first critical bifurcation mode is given by Eq. 5 and it involves a wrinkling mode shape at the compressive side of the cylinder:

$$M_{cr} = \frac{\pi E t^2 r}{\sqrt{3(1-\nu^2)}} \tag{5}$$

where, E is the modulus of elasticity of steel, t is the thickness of the shell, r is the radius and ν is the Poisson's ratio. For the case of the cylinder studied in this paper, the critical moment from Eq. 5 is 304.2MNm.

The computational model for the bifurcation analysis includes the cylinder modelled with shell elements (S4R in ABAQUS). One end of the cylinder is fixed (all six degrees of freedom are constrained) using a reference point in the middle of the end, while the other end of the cylinder is free. At the free end, a bending moment is applied. It must be clearly mentioned here that both ends in the model are simulated using a rigid body constraint. This constraint essentially provides infinite radial shell stiffness at the two ends. This stiffness is provided in real wind turbine towers by the stiffening rings between segments of the tower. Although this stiffness in reality is not infinite, this assumption is considered adequate, since the rings are stiff enough radially.

The modeling and the bifurcation analysis is performed in ABAQUS. Several mesh densities are considered to achieve a reliable model. The mesh convergence study results are presented in Fig. 2. In total, 7 different mesh densities are analyzed, starting from a model with only 240 elements up to a model with 137520 elements. The first conclusion from this mesh convergence study is that a model with 15280 elements provides a critical moment of 312.6MNm which is only 2.74% away from the theoretical value of Eq. 5.

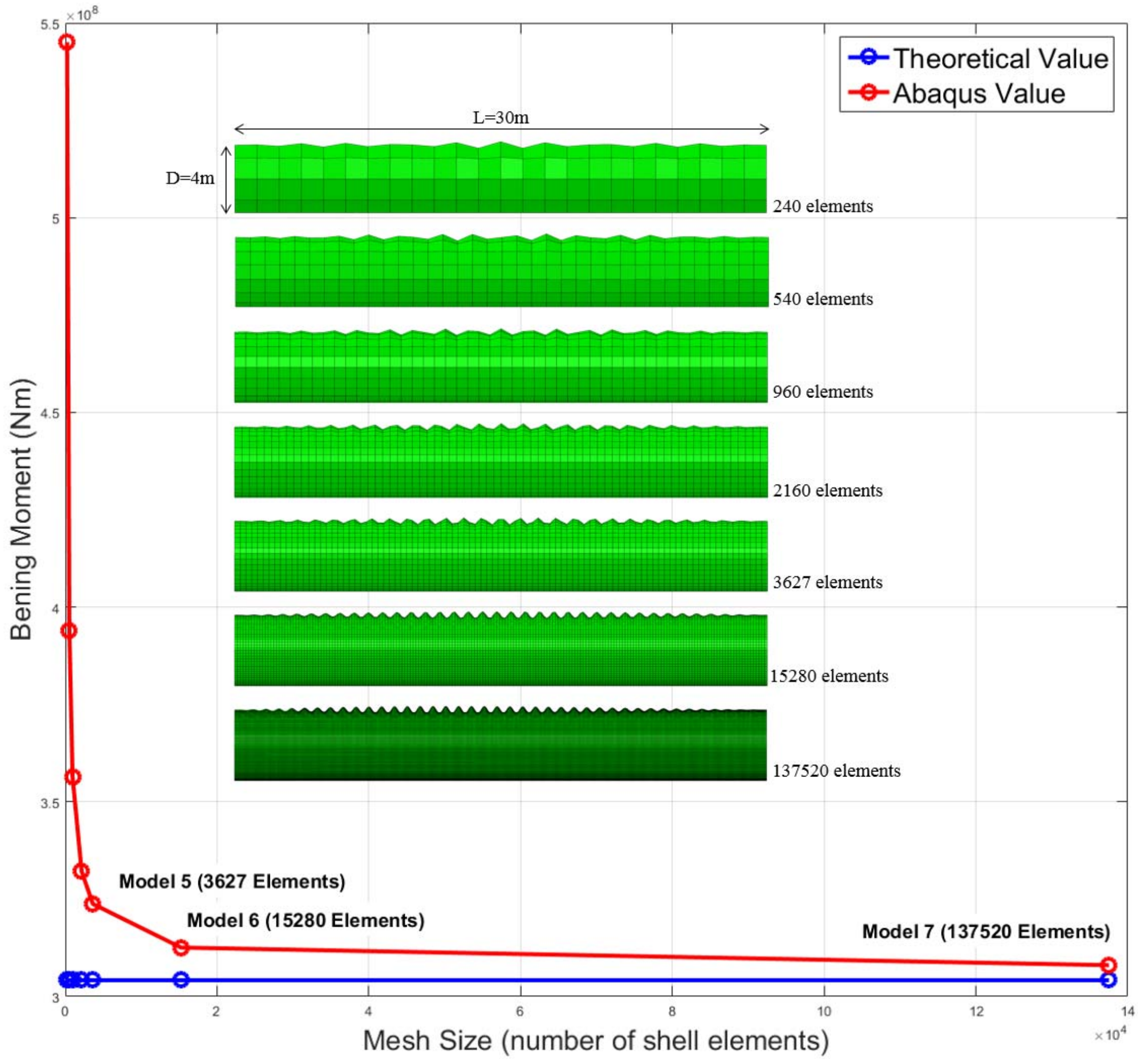

Figure 2: Mesh convergence study.

Figure 3 includes a close-up view of the wrinkling buckling mode of the cylinder. In this figure, the half-wave length is also defined on the mode which is 0.36652m. Timoshenko (1961) mentions that the half-wave length of that mode is given by Eq. 6:

$$l = 1.72\sqrt{tR} \qquad (6)$$

Which yields a value of 0.344m in this particular case. This accuracy is also considered adequate.

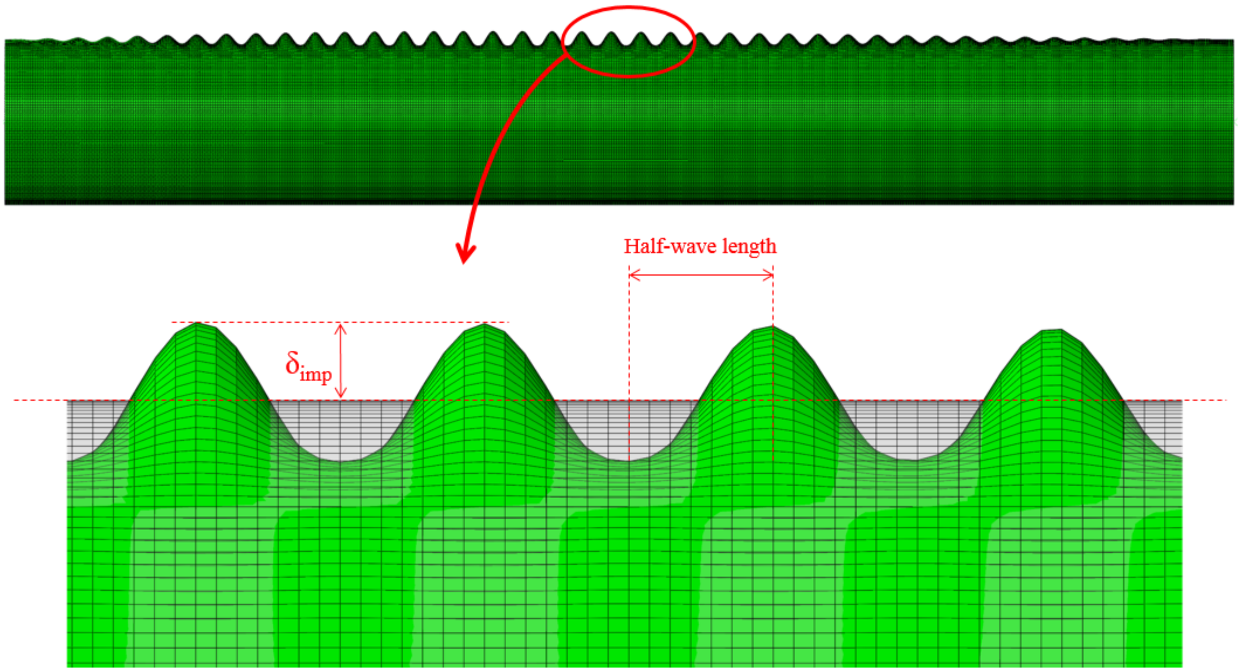

Figure 3: Wavelength from the bifurcation analysis.

3. Buckling of thin elastic cylinders under bending – Elastic Ovalization

The second part of the presented work involves the computation of the critical bending moment for the specific cylinder by using a static, elastic analysis. This method of analysis is capable of accounting the prebuckling deformation which is governed by the effect of ovalization as it was described for the first time by Brazier (1927). This effect is important for long cylinders such as wind turbine towers and it is attributed to the curvature of the cylinder which comes along with flexure. The analytical solution for this phenomenon was given by Brazier (1927):

$$\overline{M} = \frac{2\sqrt{2}}{9} \cdot \frac{E\pi r t^2}{\sqrt{1-\nu^2}} \tag{7}$$

For the specific geometry studied in the paper, the critical ovalization bending moment is 165.6MNm. It is already clear that the critical ovalization elastic bending moment is significantly lower than the bifurcation critical bending moment.

A very important point needs to be made here. According to Axelrad (1965) the effect of the end boundary conditions has no influence on the ovalization response when:

$$l / r \geq 2.97\sqrt{r / t} \tag{8}$$

For the specific case studied here:

$$l / r = 15 < 2.97\sqrt{r / t} = 29.7 \qquad (9)$$

Which indicates that the end conditions actually have an influence on the ovalization response of the cylinder. This was justified by the results of the ABAQUS model which provided an ovalization critical bending moment of 227MNm (in contrast to the 165.6 value from Eq. 7).

For the validation of the model, another longer cylinder was also analyzed (l=100m). This model yielded a value of 158.6MNm which is very close to the theoretical value from Eq. (7). A first conclusion therefore is that the end conditions of the specific segment of the tower have an effect on the ovalization response of the cylinder. It is important to note here that wind turbine towers usually have internal stiffnening rings to effectively remove the phenomenon of ovalization.

4. Inelastic Buckling of thin elastic cylinders under bending

Although ovalization can happen in the inelastic range as well, most cylindrical shells in the range of $R/t \approx 100$ will buckle through a wrinkling buckling failure of the compressed wall (Houliara 2011).

An interesting observation on the outcome of several methods of analysis is presented at this point. Table 1 includes the results of the FEM model when different analysis methods are used under different assumptions.

Table 1: Critical Moments from different analysis methods

Method of analysis	Elastic or Inelastic	Shell geometry	Critical Moment (MNm)
Bifurcation	Elastic	Perfect	312.6
Static	Elastic	Perfect	227 for the 30m cylinder (165.6 for the 100m cylinder)
Static	Elastic	Imperfect	164
Static	Inelastic	Perfect	89.7
Static	Inelastic	Imperfect	83.2

As shown from the table above, the method of analysis has a great impact on the critical moment calculated, highlighting the importance of appropriate simulation of the problem. It must be also mentioned here that the imperfect shell geometry involves imperfections in the form of the 1st eigenmode of the shell. The amplitude for this imperfection is considered always to be t/10 for all the analysis including imperfections.

Fi.g 4 includes a graph which describes the post-buckling behavior of the cylinder in bending, along with its deformed shape after buckling has occurred. The graph is presented in terms of applied angle at the free end, since a deformation controlled analysis has been followed, along with the Riks algorithm in ABAQUS. Fig. 5 and Fig. 6 present the Von Mises stresses on the top-half (compressed) wall of the cylinder and the bottom-half (tension) wall of the cylinder respectively.

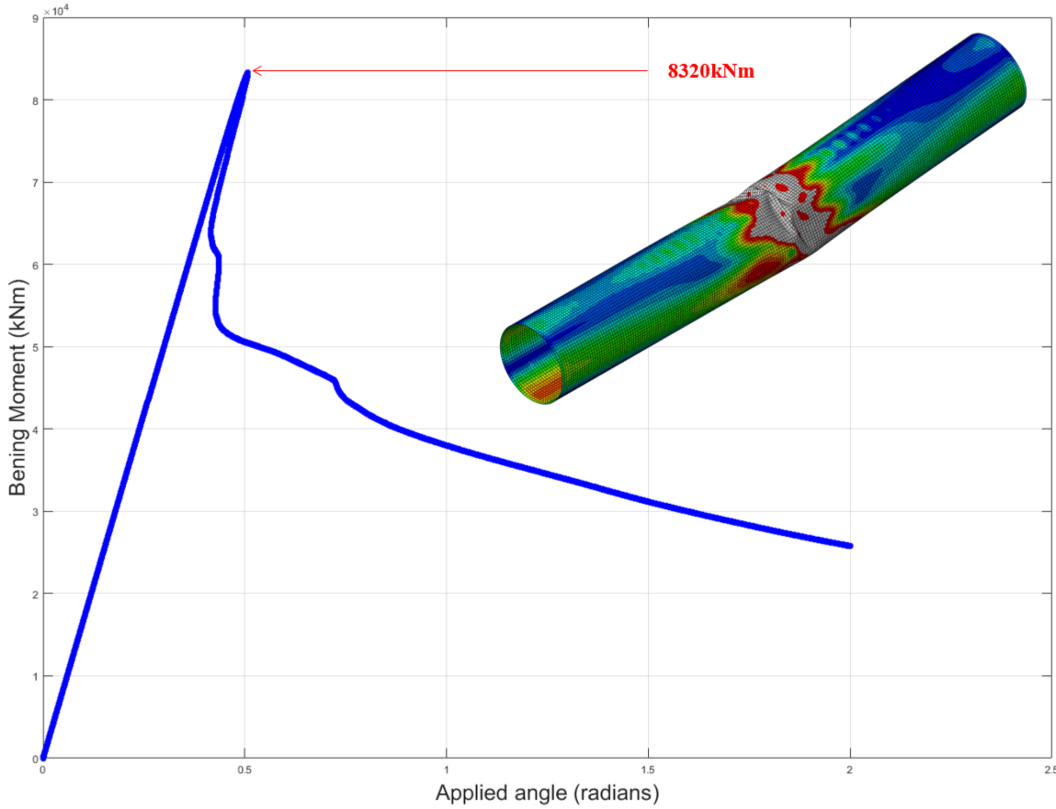

Figure 4: Moment-end angle path from the inelastic, imperfect analysis.

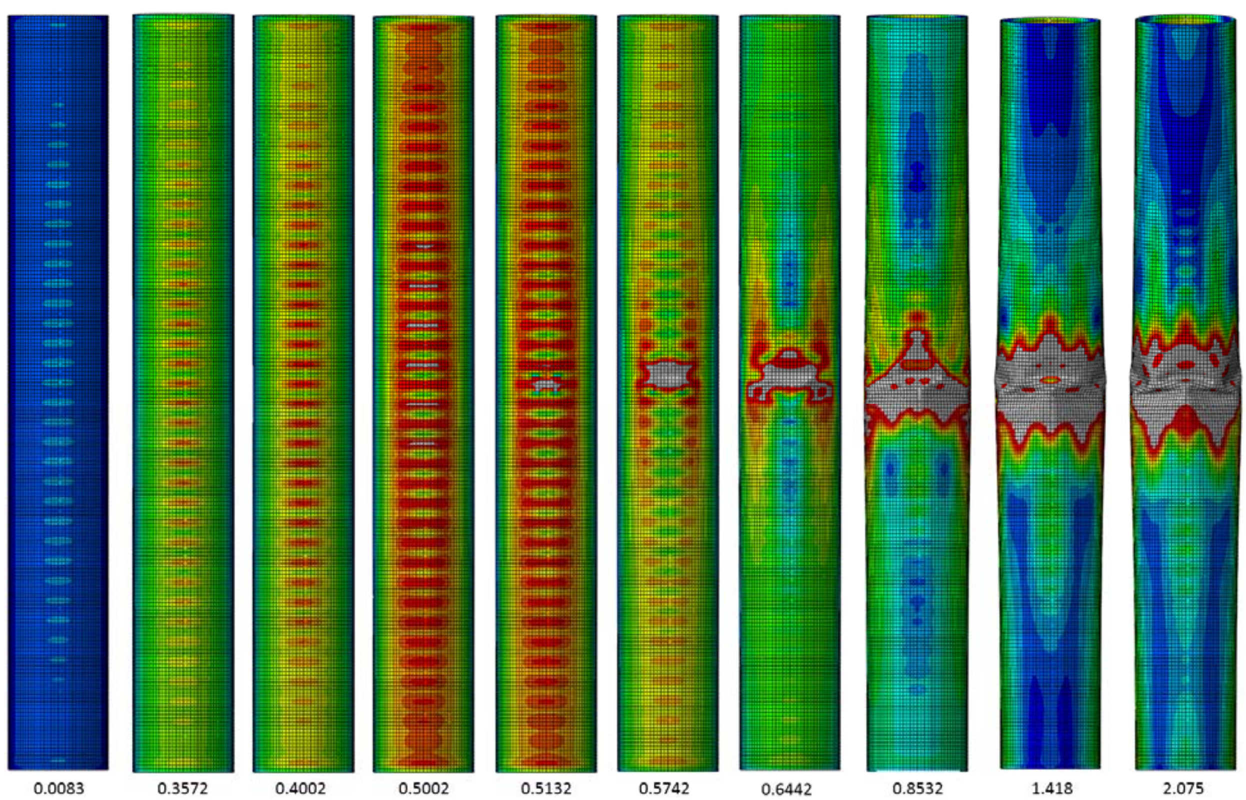

Figure 5: Buckling Shape and Progression of Von Mises Stresses at various applied angles in rads. Compressed wall of cylinder.

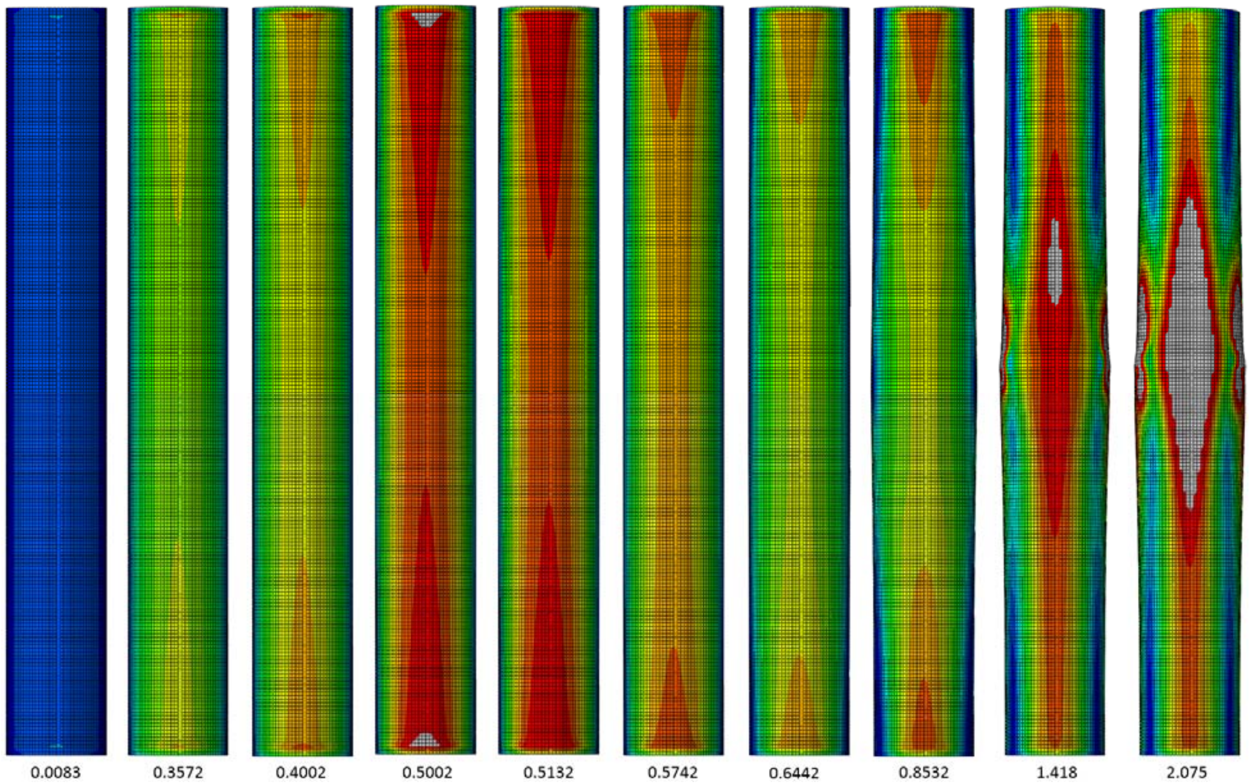

Figure 5: Buckling Shape and Progression of Von Mises Stresses at various applied angles in rads. Tensile wall of cylinder.

5. Conclusions

The present paper only includes the first findings of the ongoing work by the author on the investigation of the effects of residual stresses on the buckling of a thin-walled cylinder. It is clearly noted that this paper presents only the preliminary work and does not reach the intended assessment. The reader is kindly referred to future publications by the author for more results.

Acknowledgments

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References

Axelrad, E.L. (1965). "Refinement of buckling-load analysis for tube flexure by way of considering precritical deformation." *Izvestiya Akademii Nauk SSSR, Otdelenie Tekhnicheskikh Nauk*, Mekhanika I Mashinostroenie, 4 133-139 (in Russian).

Brazier, L.G. (1927). "On the flexure of thin cylindrical shells and other "thin" sections." Proc. R. Soc. London, Ser. A, 116 104-114.

Hutchinson, J.W. (2010). "Knockdown factors for buckling of cylindrical and spherical shells subject to reduced biaxial membrane stress." *International Journal of Solids and Structures*, 47 1443-1448.

Houliara, S., Karamanos, S.A. (2011). "Buckling of thin-walled long steel cylinders subjected to bending." ASME *Journal of Pressure Vessel Technology*, 133 (1) doi: 10.1115/1.4002902.

Koiter, W.T. (1967). "On the stability of elastic equilibrium." (English Translation: NASA Technical Translation F10), H.J. Paris, Amsterdam.

Koiter, W.T. (1963). “The effect of axisymmetric imperfections on the buckling of cylindrical shells under axial compression.” *Proc. K. Ned. Akad. Wet.*, Ser. B: Phy. Sci., 66 265-279.

NASA SP-8007. (1965). “Buckling of thin-walled circular cylinders”.

Seide, P., Weingarten, V.I., Morgan, E.J. (1960). “The development of design criteria for elastic stability of thin shell structures.” *Final report: STL/TR-60-000-19425,* Space Technology Laboratories, Inc., Los Angeles, CA.

Seide, P., Weingarten, V.I. (1961). “On the buckling of circular cylindrical shells under pure bending.” *ASME Journal of Applied Mechanics,* 28 112-116.