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MODELS FOR ESTIMATING THE AUTOFRETTAGE PRESSURE AND RESIDUAL STRESSES IN WALLS OF TYPE 2 PRESSURE VESSELS

Tsz L. "Elaine" Tang[1], Letchuman "Sri" Sripragash[2], Santosh B. Narasimhachary[2], Ashok Saxena[3]

[1] Siemens Technology, Princeton, NJ
[2] Siemens Technology, Charlotte, NC
[3] WireTough Cylinders, Bristol, VA

ABSTRACT

Autofrettage is a manufacturing process used during fabrication of Type 2 pressure vessels designed for usage at high pressures to enhance the fatigue life. The process locks high compressive residual stresses in the vessel's liner wall that reduce the maximum and the mean stress during service loading. To achieve a consistent residual stress profile across the wall of the vessel, the choice of autofrettage pressure is critical. Several finite element analyses runs were conducted to estimate the correct autofrettage pressure that is a function of the yield strength of the liner material, the wall thickness, and the outside diameter of the liner for a wrap thickness of 17.06 mm (48 wire layers). From the finite element analyses results, models were developed to estimate the correct autofrettage pressure and the residual stress distribution across the liner wall as a function of the yield strength of the liner material, the wall thickness, and the outside diameter of the liner.

Keywords: Autofrettage, Pressure vessel, Hydrogen storage, SA 372 Steel

NOMENCLATURE

P = Pressure
S_y = Yield Strength
D = Diameter
t = Liner Thickness
t_w = Wire wrap thickness
σ_θ = circumferential stress
ID = Inner Diameter
OD = Outer Diameter

1. INTRODUCTION

Autofrettage is used in the pressure vessel industry to enhance the fatigue life of vessels for storing hydrogen at high pressures [1][2] and [3]. As the storage pressures increase, the amplitude of circumferential cyclic stress during each pressurization–depressurization cycle also increases, resulting in a shorter fatigue life cycle of the vessel. Autofrettage is used to enhance the fatigue life of the vessel by locking permanent compressive stresses in the entire liner wall of the vessel. High compressive stresses generated by autofrettage not only decrease the maximum fatigue stress, but also decrease the stress ratio during the service fatigue cycle, thereby significantly enhancing the environment assisted fatigue crack growth life of the pressure vessel. In this paper, the results of finite element studies are reported and used to establish closed form equations to estimate how the correct autofrettage pressure varies with the yield strength of the liner material, the thickness of the liner, and the outside diameter of the liner.

2. FINITE ELEMENT ANALYSES CASES OF THE WIRE-WRAPPED CYLINDER

To determine the autofrettage pressure and the residual stresses from the autofrettage process, the entire pressure vessel assembly was evaluated using finite element analyses to determine stresses generated during the wrapping process and the residual stresses due to autofrettage. The finite element analyses were conducted using Siemens' Simcenter 3D [4]. The simulation workflow was automated using NX Open API., The workflow includes the following steps: (i) geometric design and mesh update, (ii) setting the appropriate initial and boundary conditions, (iii) solving the Simcenter Nastran simulations, and (iv) results extraction and analysis after each simulation. The design space exploration was performed with Siemens Simcenter HEEDS [5]. HEEDS accelerates the design space exploration by enabling automation of the simulation analysis workflows and enabling distributed execution using available computational resources. Finite element simulations included consideration of

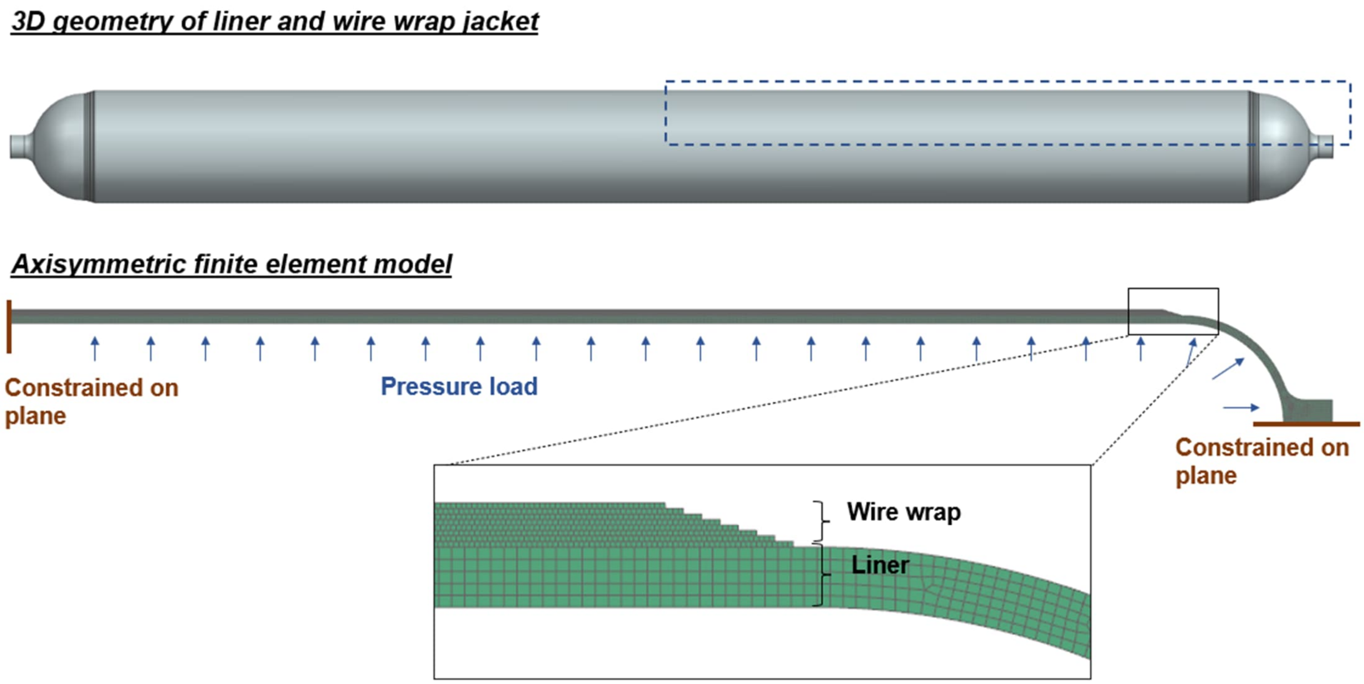

FIGURE 1: FINITE ELEMENT MODEL OF THE WIRE-WRAPPED PRESSURE VESSEL WITH BOUNDARY CONDITIONS

different liner outer diameters (ODs), liner wall thicknesses and the yield strengths of the liner material. The liner was made from SA 372 Grade E Class 70 steel [6] and the wire jacket wrap consisted of 48 layers of SA905 high strength wire with a diameter of 0.355 mm [7]. Isotropic elastoplastic material model was used for the liner and orthotropic linearly elastic material model was used for the wire jacket. The wire was applied as 8 stacks of wire with each stack consisting of six layers of wire wraps. The first stack was offset 9 mm from the start of the dome and each subsequent layer was offset 7 mm from the layer beneath it. Figure 1shows the axisymmetric finite element model of the wire-wrapped vessel with boundary conditions. The edges at the symmetry planes were constrained in the corresponding directions. Pressure load was applied on the inner edge of the liner. Every layer of wire is wrapped with an offset as shown, for the ease of manufacturing. Glue contact sets were defined to model the liner-wire interface. Simcenter Nastran solution 401 (Multi-Step Nonlinear) was utilized for the nonlinear structural analysis. It is a structural solution which supports a combination of static (linear or nonlinear) subcases and modal (real eigenvalue) subcases. Four loading conditions were simulated and summarized in Table 1. The convergence of the numerical simulations was determined by error tolerance for load, displacement, and work.

Several finite element cases were run in the range of liner wall thickness, t, $22.9 \leq t \leq 25.2$ mm and the yield strength, S_y, $482.5 \leq S_y \leq 703.5$ MPa and for outer liner diameters of 406, 508, and 617 mm. The specific cases for which finite element analyses were conducted are summarized in Table . The results are described in the next section.

TABLE 1: INITIAL AND LOADING CONDITIONS OF THE FINITE ELEMENT ANALYSIS

Simulation	Initial Condition	Loading condition
1. Pre-stress of wire		Circumferential stress of 140MPa was applied on the wire wrap layers
2. Determination of autofrettage pressure	Stress field from (1)	Pressure load inside the liner was increased linearly from 0 MPa to identify autofrettage pressure
3. Autofrettage simulation	Stress field from (1)	Pressure load inside the liner was increased linearly from 0 MPa to the autofrettage pressure identified (2), and then decreased linearly back to 0 MPa
4. Operating condition	Stress field from (3)	Operating pressure load applied inside liner

TABLE 2: MATRIX OF CASES ANALYZED USING FINITE ELEMENT ANALYSES

OD, mm	Sy, MPa	Liner Wall Thickness, mm					Wrap thickness, mm
617		22.9	23.5	24.0	24.5	25.2	17.06
	482.5	X	X	X	X	X	
	517.0	X					
	552.0	X					
	587.0	X					
	622.0	X					
	658.0	X					
	703.5	X	X	X	X	X	
508	703.5					X	
406	703.5					X	

3. MODEL FOR ESTIMATING THE AUTOFRETTAGE PRESSURE

This section presents a model to determine the appropriate autofrettage pressure that accounts for the liner wall thickness, the liner material yield strength, and the liner OD and will lead to consistent and repeatable residual stress pattern in the liner walls. Results from the cases listed in Table were used to derive an equation for estimating the autofrettage pressure as a function of the above variables. Figure 2 shows a typical plot of von Mises' equivalent strain on the OD of the liner as a function of the applied internal pressure in the wire-wrapped cylinder. The autofrettage pressure was determined when the equivalent strain on the OD of the liner just exceeds the elastic strain for the stress level applied plus a plastic strain of 0.003 (or 0.3%) to ensure the material has reached plasticity. This ensures that the entire liner wall has deformed plastically at the autofrettage pressure, so when the pressure is removed, it locks high compressive stresses throughout the cross-section of the liner. The autofrettage pressure determined in this way is marked on Figure 2.

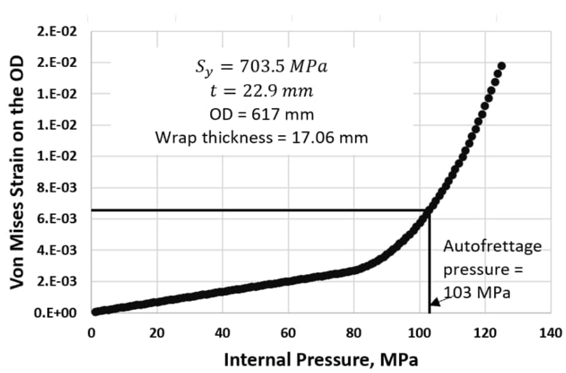

FIGURE 2: VON MISES STRAIN AT THE OD OF THE LINER AS A FUNCTION OF INTERNAL PRESSURE. THE AUTOFRETTAGE PRESSURE IS CHOSEN WHEN THE STRAIN EQUALS THE ELASTIC STRAIN PLUS AN EQUIVALENT PLASTIC STRAIN OF 0.3%.

Figure 3a and b show the typical distribution of hoop stress across the cylinder wall (a) after wrapping (b) at the autofrettage pressure, and (c) after removal of the autofrettage pressure, in cylinders containing liners with a wall thickness of 22.9 mm and yield strengths of 482.5 and 703 MPa, respectively. The magnitudes of residual stresses due to autofrettage in the higher yield strength liner are significantly greater than in the liner with lower yield strength. As expected, the compressive residual stresses in the liner are equilibrated by tensile stresses in the wrap. Also note that the tensile stress in the wrap is less than 825 MPa which is much less than the strength of the wire, estimated to be greater than 2 GPa. Therefore, the wrap is not at risk of fracturing during autofrettage. Also note that the interface between the wrap and the liner is expected to be under a high contact pressure which is beneficial for the integrity of the cylinder.

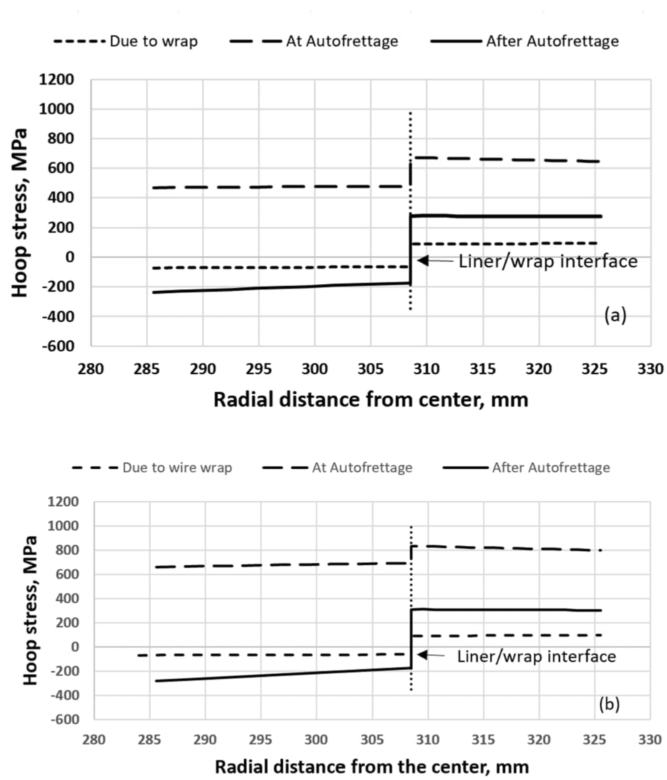

FIGURE 3: DISTRIBUTION OF CIRCUMFERENTIAL STRESS ACROSS THE LINER WALL THICKNESS AND THE THICKNESS OF THE WIRE WRAP DUE TO WRAP PRE-STRESS, AT THE AUTOFRETTAGE PRESSURE, AND AFTER AUTOFRETTAGE. ALL LINERS IN THE IN THESE ANALYSES HAD AN OD OF 617 MM, A WALL THICKNESS OF 22.9 MM, AND THE WRAP THICKNESS OF 17.06 MM. (A) IS FOR A LINER MADE FROM MATERIAL WITH YIELD STRENGTH OF 482.5 MPA AND (B) FOR A LINER MADE FROM A MATERIAL WITH A YIELD STRENGTH OF 703.5 MPA.

Figure 4a shows the calculated autofrettage pressure as a function of the wall thickness of the liner material for two levels of yield strength, and Figure 4b shows the relationship between the autofrettage pressure and the yield strength for a liner with a fixed thickness of 22.9 mm. In both cases the liner OD was 617 mm, and the wrap thickness was 17.06 mm. The autofrettage pressure is shown to increase linearly with both the thickness and the material yield strength.

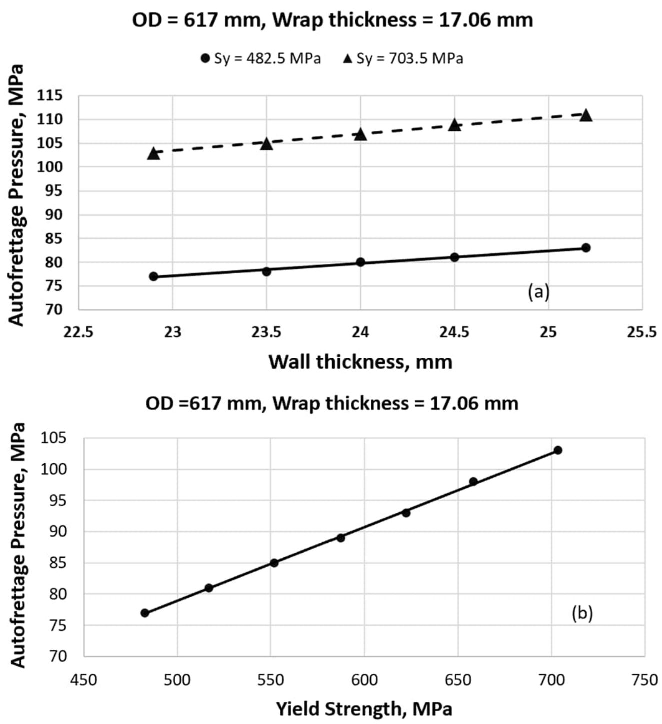

FIGURE 4: THE RELATIONSHIP BETWEEN AUTOFRETTAGE PRESSURE AND (A) WALL THICKNESS FOR TWO YIELD STRENGTHS, AND (B) YIELD STRENGTH FOR A LINER THICKNESS OF 22.9 MM. THE LINER OD WAS 617 MM, AND THE WRAP THICKNESS WAS 17.06 MM IN BOTH CASES.

Figure 5 shows the variation in the autofrettage pressure with the liner OD for two yield strength levels and two wall thicknesses. Again, the relationship with OD appears to be linear in both cases. Based on these observations from the results of the finite element analyses, it is stated that the following functional relationship exists between the autofrettage pressure and various variables:

$$\frac{P_A}{P_{Ao}} = f\left[\frac{S_y}{S_{yo}}, \frac{t}{t_o}, \frac{D}{D_o}\right] \tag{1}$$

In equation (1), P_A = autofrettage pressure, P_{Ao} = autofrettage pressure of a reference design with reference thickness, t_0 = 22.9 mm, reference yield strength of the liner material, S_{yo} =482.5 MPa, reference OD of D_o = 617 mm, and a reference wrap thickness, of 17.06 mm. S_y = actual yield strength of the liner material, t = the actual liner thickness, and D = actual OD. From the finite element results, an equation of the following form was derived to collectively represent the effects of yield strength, wall thickness, and the OD for a constant wrap thickness, on the autofrettage pressure.

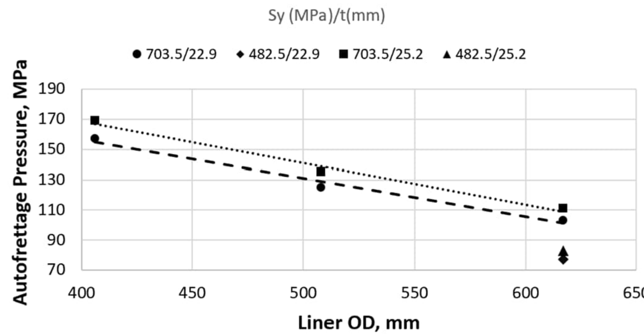

FIGURE 5: AUTOFRETTAGE PRESSURE AS FUNCTION OF OUTSIDE DIAMETER, WALL THICKNESS (22.9 & 25.2MM) AND YIELD STRENGTH OF THE LINER MATERIAL (482.5 & 703.5MPA).

$$\frac{P_A}{P_{A0}} = \left(0.26 + 0.74\frac{S_y}{S_{yo}}\right)\left(0.206 + 0.794\left(\frac{t}{t_0}\right)\right)\left(2.4 - 1.4\frac{D}{D_0}\right) \tag{2}$$

Where P_{A0}=77 MPa, S_{yo}=482.5 MPa, D_o=617 mm, t_o=22.9 mm, t_{wo} =17.06 mm. Further, the intervals in which these equations are valid are as follows:
$22.9 \leq t \leq 25.2$ mm, $406 \leq D \leq 617$ mm, $482.5 \leq S_y \leq 703.5$ MPa.

Figure 6 shows a comparison between the calculated autofrettage pressure from equation (2) and that determined directly from the finite element analyses. The various points in Figure 6 include these comparisons for liners with a variety of yield strengths, liner wall thicknesses and ODs. The predicted autofrettage pressure values from equation (2) seem to agree well with the those estimated from finite element analyses.

4. ESTIMATION OF RESIDUAL STRESSES FROM AUTOFRETTAGE

The objective of autofrettage is to develop a pattern of predictable compressive residual stresses in the wall of the liner that reduce the maximum stresses during service. Figure 7 shows the residual stress distribution following autofrettage for liners of different thicknesses and yield strength. All liners included in Figure 7 had an OD of 617 mm that were wrapped in a wire jacket that was 17.06 mm thick. The compressive residual stresses are the highest in magnitude on the ID and lowest at the OD and are distributed linearly between the ID and OD. Since the autofrettage pressure is linked to a fixed amount of plastic deformation on the OD, the residual stress pattern for different thicknesses lies along the same trend. Thus, the magnitude of residual stress does not significantly depend on the liner thickness, if the autofrettage pressure is chosen correctly.

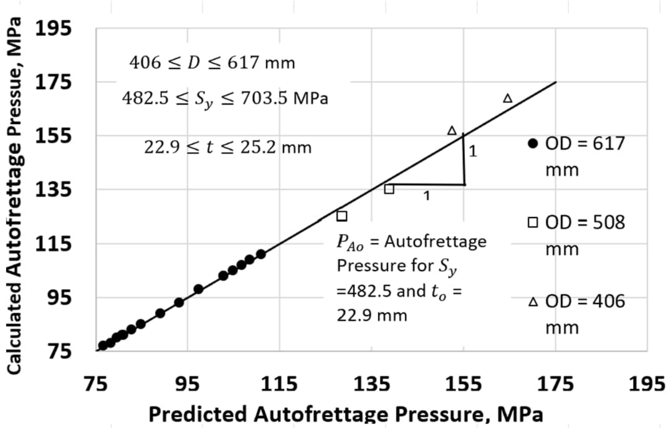

FIGURE 6: COMPARISON BETWEEN PREDICTED AUTOFRETTAGE PRESSURE FROM EQUATION (2) AND THAT CALCULATED BY FINITE ELEMENT ANALYSES

Figure 8 shows the variation in residual stress distribution along the liner thickness for cylinders from liners with different ODs and yield strengths for a constant wrap thickness of 17.06 mm. The distribution of stress between ID and OD is assumed to be linear consistent with the observation from Figure 7. The residual stress changes significantly on the OD of the liner and the observation that the stress pattern is similar for liners with different thicknesses observed earlier for liners with an OD of 617 mm, also holds for the ODs of 508 mm and 406 mm.

Equations (3) and (4) represent the residual compressive circumferential stress in the liner wall at the ID and OD of the liner, respectively. These stresses are for the mid-region of the cylinder along its length. The equations account for the variation in residual stress due to yield strength, OD, and wrap thickness. Between the ID and OD, the stress distribution is expected to be approximately linear. As shown before, the ID residual stresses are independent of liner thickness, but the OD residual stress do change linearly with liner thickness but, predictably.

$$\sigma_{\theta ID} = \sigma_{\theta IDo}\left(0.6 + 0.4\left(\frac{S_y}{S_{yo}}\right)\right)\left(1.4 - 0.4\frac{D}{D_0}\right) \tag{3}$$

$$\sigma_{\theta OD} = \sigma_{\theta ID} - \left(-195 + 105\left(\frac{D}{D_0}\right)\right)\frac{S_y}{S_{yo}}\frac{t}{t_o} \tag{4}$$

$\sigma_{\theta ID}$ = circumferential stress at the ID of the liner, $\sigma_{\theta IDo}$ is the circumferential stress at the ID of the reference liner with a value of $238.84\ MPa,\ S_{y0} = 482.5\ MPa,\ D_0 = 617\ mm,\ t_0 = 22.9\ mm$. Figure 9 and Figure 10 show a direct comparison between the predicted residual circumferential stress using the derived equations and the simulated values at the ID and OD for all cases analyzed by finite element analyses. There is excellent agreement between the residual stresses predicted from equations (3) and (4) and those estimated by finite element. In these figures, lines corresponding to an error band of $\pm 5\%$ are plotted to demonstrate that all predicted residual stresses lie within the error band.

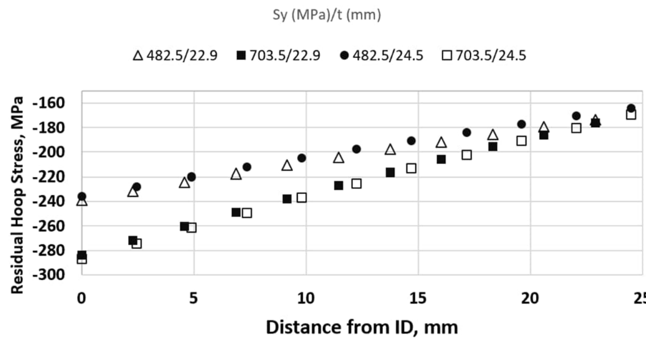

FIGURE 7: THE RESIDUAL STRESS DISTRIBUTION IN THE LINER WALL FOLLOWING AUTOFRETTAGE, WITH DIFFERENT YIELD STRENGTH (482.5 & 703.5 MPA) AND LINER THICKNESSES (22.9 & 24.5 MM). THE ANALYSES RESULTS ARE FOR LINERS WITH AN OD OF 617 MM AND A WRAP THICKNESS OF 17.06 MM.

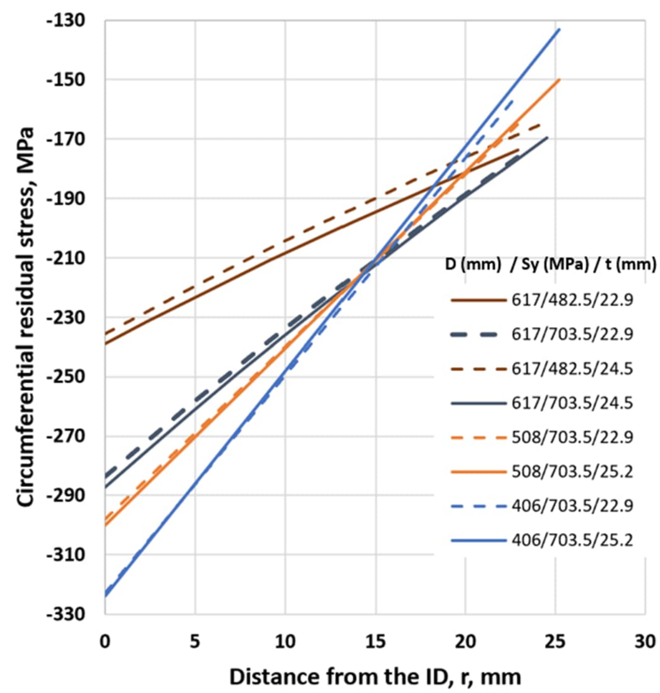

FIGURE 8: RESIDUAL STRESS PATTERN IN THE LINER AFTER AUTOFRETTAGE FOR CYLINDERS OF DIFFERENT ODS (617, 508, 406MM), YIELD STRENGTH (482.5, 703.5MPA) AND LINER THICKNESS (22.9, 24.5, 25.2MM).

5. SUMMARY AND CONCLUSIONS

Finite element analyses were conducted for several cases of wire wrapped cylinders of pressure vessels containing liners with OD ranging from 406 mm to 617 mm, liner wall thickness varying between 22.9 and 25.2 mm, and the liner yield strength ranging from 482.5 to 703.5 MPa. The wrap thickness in all analyses was held constant at 17.06 mm, or 48 layers of wire wrap. From the results of the finite element analyses, models were developed to estimate the autofrettage pressure as a function of the liner OD, material yield strength, and the liner wall thickness. Equations were developed to estimate the compressive residual stress at the ID and OD in the mid-section of the cylinder due to autofrettage in the liner walls. Both models were shown to yield results in excellent agreement with the values obtained by finite element analyses, which could help in improving the design (geometric dimensions and material choice) of autofrettaged pressure vessels.

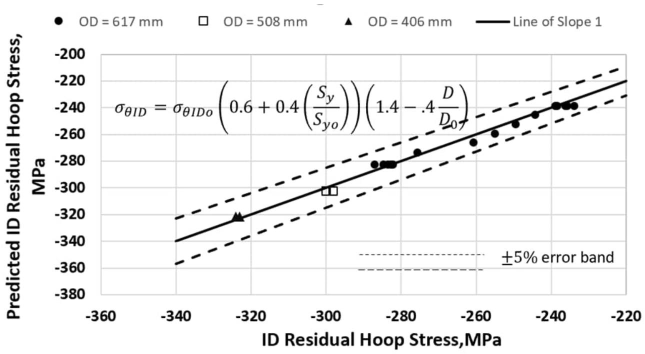

FIGURE 9: PREDICTED RESIDUAL CIRCUMFERENTIAL STRESS AT THE ID FROM EQUATION (3) AND THAT CALCULATED FROM THE FINITE ELEMENT ANALYSES UNDER SIMILAR CONDITIONS.

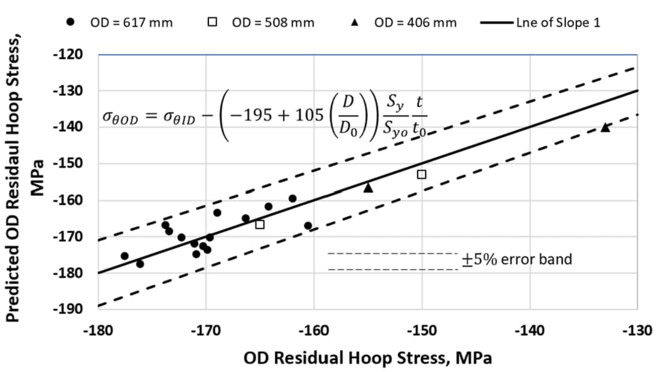

FIGURE 10: PREDICTED RESIDUAL CIRCUMFERENTIAL STRESS AT THE OD FROM EQUATION (4) AND THAT CALCULATED FROM THE FINITE ELEMENT ANALYSES UNDER SIMILAR CONDITIONS.

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