Finite Element Analysis of Pressure Vessel with Flat Metal Ribbon Wound Construction under the Effect of Changing Helical Winding Angle

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Abstract - Main objective of this research was to develop a Finite Element Model (FEM) of thin walled flat metal ribbon wound pressure vessel (FMRWPV) using ANSYS software. In this research work, analytical formulae for hoop and axial stress in thin walled FMRWPV under the effect of changing helical winding angle have been developed. Finite Element Analysis (FEA) has been carried out using ANSYS software. The design consists of an inner shell with thickness 40% of the total design thickness and six layers of flat metal ribbons (FMRs) each with 2mm thickness. Ribbons have been wound on the inner shell with same pre-tension by pulling them during winding. The results from analytical and FEA methods in terms of hoop and axial stress have been compared. Suggestions have been made to compensate for variations between analytical and FEA results.

Keywords: FMRWPV, FEA, FMR, FEM.

NOMENCLATURE

A_i: Inner shell effective area for hoop stress case (mm^2)
A_i^*: Inner shell effective area for axial stress case (mm^2)
A_1: Area of ribbon1 in case of hoop stress (mm^2)
A_1^*: Area of ribbon1 in case of axial stress (mm^2)
P_i: Internal operating pressure (MPa)
R_i: Inside radius of the pressure vessel (mm)
S_{afl}: Final axial stress in ribbon layers (MPa)
S_{afs}: Final axial stress in the inner shell (MPa)
S_{ai}: Residual axial stress in inner shell (MPa)
S_f: Final stress (MPa)
S_{hfl}: Final hoop stress in each ribbon (MPa)
S_{hfs}: Final hoop stress in inner shell (MPa)
S_{hi}: Residual hoop stress in inner shell (MPa)
S_{ope}: Stress due to internal pressure (MPa)
S_{pre}: Pre (residual) stress (MPa)
S_t: Pre-tension in the ribbons (MPa)
S_{t1}: Pre-tension in the ribbon1 (MPa)
n: Number of ribbon layers on inner shell
t_i: Inner shell thickness (mm)
t_r: Ribbon thickness (mm)
α: Helical winding angle (degrees)
L_1, L_2, L_3, L_4, L_5, L_6: Ribbon layers 1,2,3,4,5 and 6
IS: Inner shell

1 INTRODUCTION

Pressure vessels are used basically for the containment of some fluid under pressure. Pressure Vessels are classified into thin-walled and thick walled vessels according to dimensions. In thin walled vessels, the main stresses are longitudinal and circumferential while in thick-walled vessels, besides the longitudinal and circumferential stresses the radial stress is also important [1, 7]. The FMRWPV is the newest and most emerging technique of multilayer pressure vessels. FMRWPV is introduced in Peoples Republic of China during last few decades. For the 1st time Guo Hui Zhu presented his research on this technology at Zhejiang University China in 1964. A typical model of FMRWPV is shown in below figure.

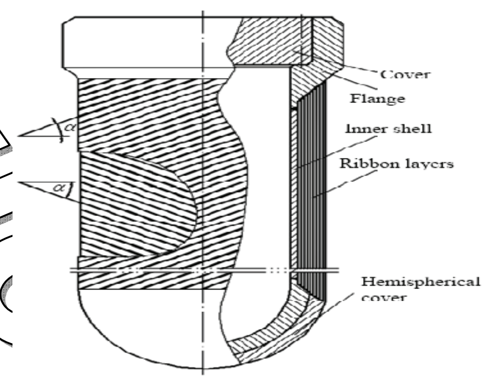

Figure 1 - FMRWPV [4]

Then Hung and Zhu [4] presented their research on this technology. In such technique a single walled pressure vessel is replaced with a thin inner shell and a number of layers of FMRs, wound on the inner shell. The thickness of inner shell is suggested to be 15 to 45 percent of the total thickness and the remaining thickness are compensated by the outer layers of FMRs which are wound under tension. The suggested sizes of FMRs are 50-80 mm in width and 3-8 mm in thickness. The overall thickness will be the same but the manufacturing will be much simplified. Chuan-xiang [5] in his research investigated that how the burst resistance and strength of ribbon-wound pressure vessel varies with winding angle. Axial strength has close relation with winding angle. He found that hoop as well as axial strength of ribbon wound vessel is greater than that of conventional vessel and the strength is optimized for winding angle 15-30 degrees.

Brownell and Young [2] presented their work on thin walled FMRWPV but they had not considered the helical winding angle. They derived hoop stress formula for hoop winding of shell without deriving axial stress formula. Zhu and Zhu [3] in their research proposed formulae for circumferential and axial stress at which bursting will occur. Their work shows that such vessels are safer against axial as well as circumferential stress but have more strength in axial direction. They suggested angle range of 20-25 degrees. Zhu et al [6] in their research have described that a simple lathe type machine is required for winding of pressure vessels with steel ribbons. The ribbon is wound under tension. Zheng et al [8] reviewed the characteristics, design methods, relevant developments and advancements in FMRWPV technology. Ribbon wound vessels have reasonable strength against bursting. Such vessels have property to leak before bursting that gives indication of failure before the time. Zhu and Zhu [9] in their work compared different design methods in pressure vessel technology and described advantages of Chinese flat steel ribbon wound technology over other methods. They used a thin inner core of thickness 1/6-1/4 of the total vessel thickness and compensated the remaining thickness by thin flat steel ribbons wound helically around the inner core. About 80% welding is reduced by using this technology. Hearn [10] in his book presented a basic concept of designing FMRWPV. Vrbka and Suchanek [11], in their work presented an analysis of a vessel wound with orthotropic strips to develop mathematical formulae considering tangential and radial stiffness of the strips.

2 ANALYTICAL FORMULATION

To validate the FEM developed in this research work, analytical formulae have been developed from the theory of thin walled shells using Tailor Series Expansion Method, by considering one, two, three and then n number of layers of FMRs on the inner shell. The basic concept is taken from Brownell and Young's book [2], who have derived formula for hoop stress by considering hoop winding of inner shell (without considering the helical winding angle). In the present work the inner shell has been wound helically with FMRs and formulae for hoop as well as axial stress through the inner shell and ribbons layers around the inner shell have been developed. Following assumptions have been made.

1. There should be no stress concentration.
2. There should be very low stress in inner shell and even negative if possible.
3. Materials of inner shell and ribbon layers have linear and isotropic behavior.
4. Limitations for helical winding angle are $0 < \alpha < 90^{\circ}$.
5. Same tension is applied to each ribbon.
6. All ribbons have same cross-sectional areas.

Winding of FMRs causes residual compressive stress induced in the inner shell of the vessel. The final stress in the inner shell is a combination of residual stress and the stress due to the operating pressure.

Mathematically we can write it as below

$$S_f = S_{pre} + S_{ope} \ldots\ldots\ldots (1.1)\ [2]$$

In first step a single ribbon layer is considered on the inner shell and hoop and axial stress formulae have been developed under the effect of ribbon pre-tension. In the 2nd and 3rd steps two and three ribbon layers are considered on the inner shell and formulae have been developed. At the last the formulae for hoop and axial stress through the inner shell as well as through the ribbon layers are generalized for n layers of FMRs.

The formulae for the hoop and axial stress are given below, for the detailed explanations of derivations please refer to appendix1.

2.1 Hoop stress formulae

$$S_{hfs} = -\left(\frac{nt_r}{t_i}\right).(S_t)cos^2\alpha + \frac{P_iR_i}{t_i + nt_r} \ldots\ldots (1.2)$$

$$S_{hfl} = \left(\frac{nt_r}{t_i}\right).(S_t)cos^2\alpha + \frac{P_iR_i}{t_i + nt_r} \ldots\ldots\ldots (1.3)$$

2.2 Axial stress formulae

$$S_{afs} = -\left(\frac{nt_r}{t_i}\right).(S_t)\cos\alpha\sin\alpha + \frac{P_iR_i}{2t_i + 2nt_r} \ldots\ldots (1.4)$$

$$S_{afl} = \left(\frac{nt_r}{t_i}\right).(S_t)\cos\alpha\sin\alpha + \frac{P_iR_i}{2t_i + 2nt_r} \ldots\ldots\ldots (1.5)$$

3 FINITE ELEMENT ANALYSIS

FEA of inner shell wound with FMRs under tension has been carried out using ANSYS software. The solid model of the problem is created with inner shell, six layers of FMRs and hemispherical heads, symmetric about x-axis and y-axis as shown in figure 2. Layers have been modeled as thin shells with thickness of 2mm as shown in figure 3 and ribbons have been defined in the real constant sets 1 and 2 as shown in Table1. The material used for the problem is given in Table2. The sample data used for creating the model of FMRWPV is given in Table3. After creating the solid model, inner shell and ribbon layers have been meshed separately using mapped 4 to 6 sided mesh. Solid185 element is used for the meshing of inner shell and layered solid46 for ribbon layers. The next step is to create area/surface contacts between layers and between inner shell and 1st layer. After meshing and creating area contacts, boundary conditions are applied in two steps and load step files named as ls1 and ls2 are created. In 1st step symmetric boundary conditions, constraints and imposed displacements are applied. Imposed displacements with values 0.202, 0.200, 0.201, 0.202, 0.203 and 0.204mm are applied on the face areas of layers 1, 2, 3, 4, 5 and 6 which produce equivalent tension of 66MPa in each ribbon layer and load step file ls1 is created and saved. In the 2nd step, keeping all the boundary conditions of step1, internal pressure is applied and ls2 file is created.

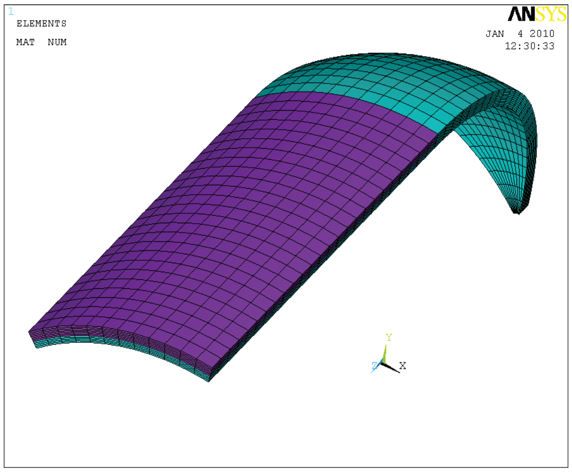

Figure 2

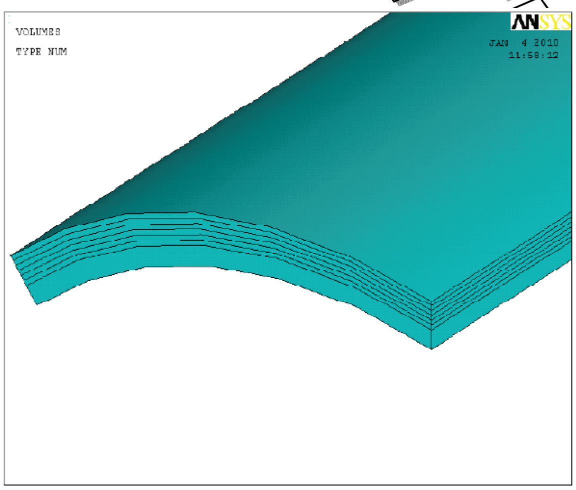

Figure 3

Table 1 - Real constant sets

Real Constant	Material	Angle (deg.)	Thickness (mm)
1	2	20	2
2	2	-20	2

Table 2 - Material properties

Material	Young's Modulus (MPa)	Poisson's Ratio	Density (Kg/mm^3)
1, 2	200	0.3	8

Table 3

Parameter	Description	Value	Units
D_i	Vessel inner diameter	600	Mm
D_o	Vessel outer diameter	640	mm
t_i	Inner shell thickness	8	mm
t_r	Ribbon thickness	2	mm
n	No. of ribbon layers	6	
α	Ribbon winding angle	15~30	degrees
P_i	Internal operating pressure	15	MPa
S_{t1}	Ribbon layer1 pretension	66	MPa
S_{t2}	Ribbon layer2 pretension	66	MPa
S_{t3}	Ribbon layer3 pretension	66	MPa
S_{t4}	Ribbon layer4 pretension	66	MPa
S_{t5}	Ribbon layer5 pretension	66	MPa
S_{t6}	Ribbon layer6 pretension	66	MPa

Solution option is used to the problem. In this work solution is done in two load steps whose files are generated as described above. Post-processing phase is used to review the results of the analysis in terms of final axial and hoop stress.

4 RESULTS COMPARISON

To see the effects of changing helical winding angle on the values of hoop and axial stress, different cases have been discussed below by taking angle 15, 20, 25 and 30 degrees as suggested by Chuan-xiang [5]. Internal pressure is taken equal to 15 MPA and remaining parameters have been kept same. Results of analytical and FEA methods in terms of hoop and axial stress have been compared graphically as shown in below figures.

Case1:

With same sample data as in Table3 and P_i=15 MPa and α=15°.

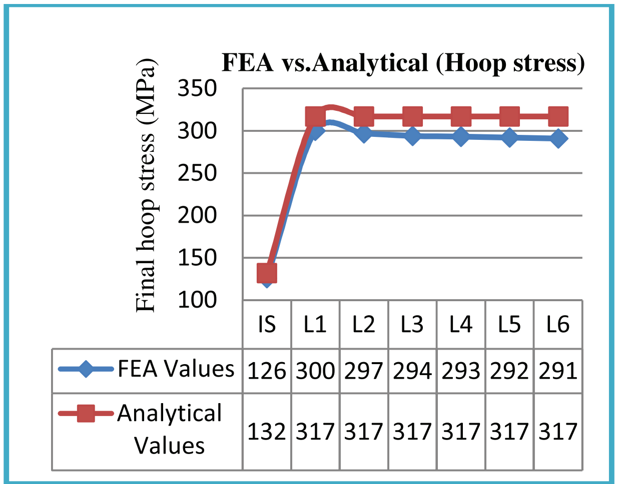

Figure 4

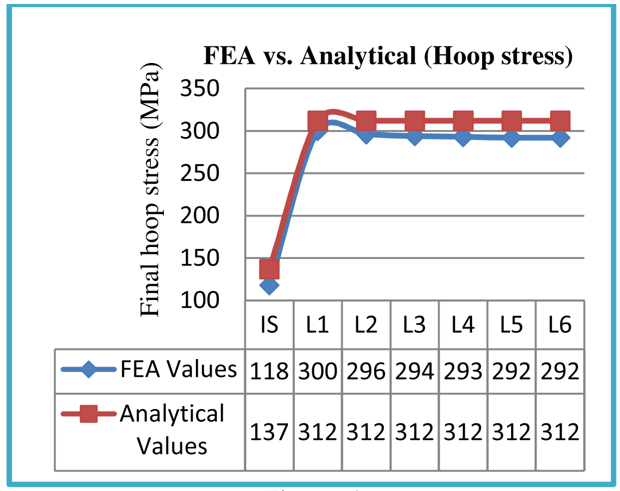

Figure 6

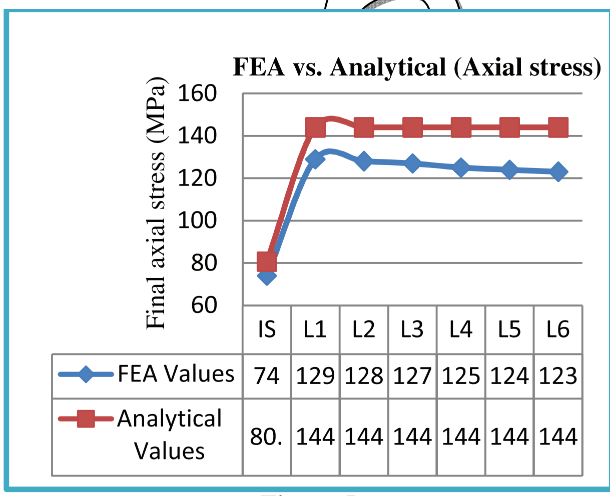

Figure 5

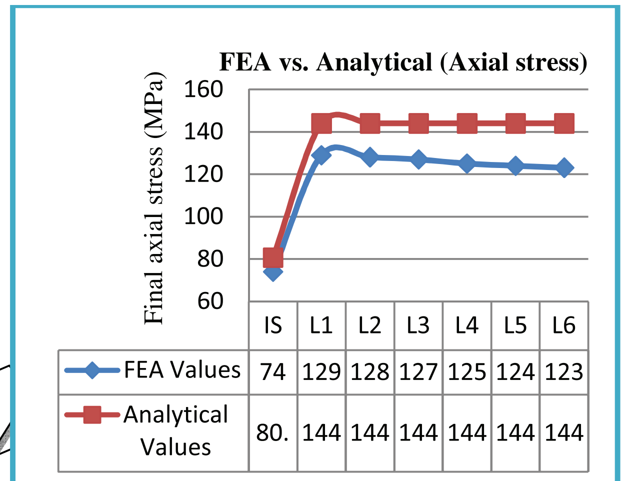

Figure 7

Figure 4 and Figure 5 illustrate that the variation between FEA and analytical values for hoop stress through the thickness of inner shell is very small. There is almost a constant variation between the FEA and analytical values of hoop stress from layer1 and onwards. Axial stress variation through the thickness of inner shell is very small but from layer1 and onwards, there is small but gradually increasing variation.

Case2:

Using same sample data with P_i=15 MPa and α=20°.

Case3:

Using same sample data with P_i=15 MPa and α=25°.

Figure 6 and Figure 7 show that the variation between FEA and analytical values for hoop stress through the thickness of inner shell is very small. There is almost a constant variation between the FEA and analytical values of hoop stress from layer1 and onwards. Axial stress variation through the thickness of inner shell is very small but from layer1 and onwards, there is small but gradually increasing variation.

Case4:

With same sample data as in Table 3 and P_i=15 MPa and α=30°. We can observe from Figure 10 and Figure 11 that the variation between FEA and analytical values for hoop stress through the thickness of inner shell is very small. There is almost a constant variation between the FEA and analytical values of hoop stress from layer1 and onwards. In case of axial stress variation through the thickness of inner shell is very small but from layer1 and onwards, there is small but gradually increasing variation.

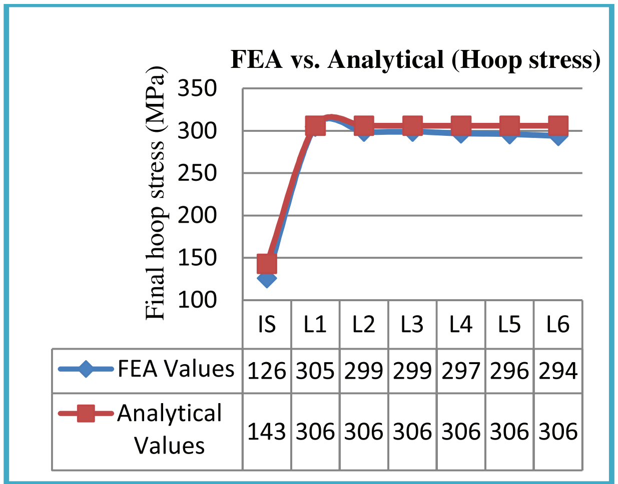

Figure 8

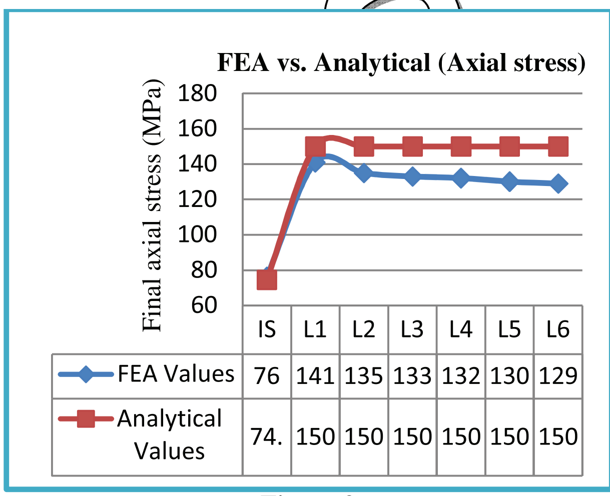

Figure 9

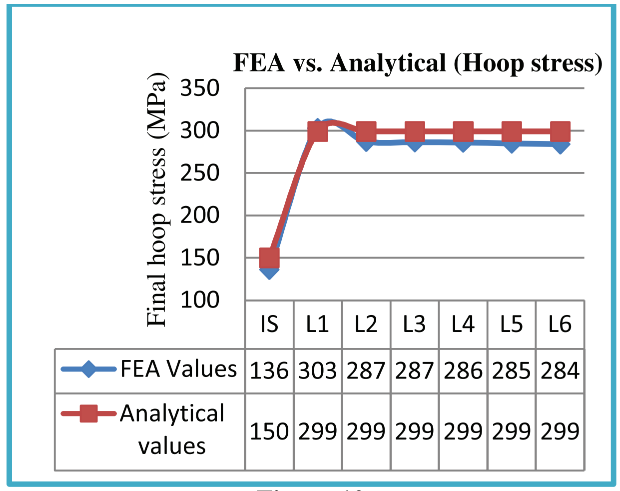

Figure 10

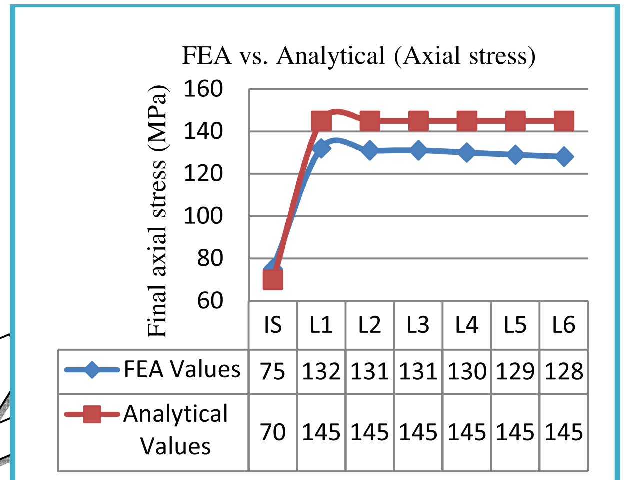

Figure 11

Above figures give a comparison between results from FEA and developed analytical methods in terms of final hoop and axial stress. The FEA and analytical values of hoop stress have a small variation through the thickness of inner shell and there is almost a constant variation between FEA and analytical values of hoop stress from layer1 and onwards as shown in Figure 8. Axial stress through the thickness of inner shell has a small variation between FEA and analytical values and from layer1 and onwards this variation is small but increasing gradually as shown in Figure 9.

5 CAUSES OF VARIATIONS

In reference to above given results and discussions the causes of variation in results are summarized as below.

1. Approximate solutions from numerical methods like FEA using ANSYS software.
2. Highly non-linear behavior of FEA solution by ANSYS software because of the contact pairs created between consecutive layers and between inner shell and ribbon layer1 which make the ANSYS solution highly non-linear.
3. Stress concentration at weld areas.

6 CONCLUSION

After comparing results and discussion on above graphs, it was concluded that

1. Winding of ribbons on the inner shell, under tension produces residual compressive stresses in the inner shell which acts against the internal pressure. The ribbons will be under tension due to pre-tensioning of ribbons during winding and this tension adds up with the tensile stresses due to internal pressure. Finally, ribbons come under higher tension, so a stronger material for ribbons is required as compared to the material for inner shell.
2. For multilayer pressure vessels like the FMRWPV, we have to develop contact pairs between the consecutive layers so that load can be transferred accurately from inner shell to the ribbon layers. The best contact pairs observed in this research were surface/area contacts.
3. ANSYS software does not converge the solution if the pre-tensioning pressure is applied directly on the face areas of ribbons; instead we have to apply imposed displacements on these areas. The reactions

measured on the opposite areas give the pretension equivalent to the applied imposed displacements.

4. Numerical analysis carried out using ANSYS software produces approximate results which are not the exact solutions as in case of analytical methods which give exact values. This causes variations between FEA and analytical results.
5. Analytical formulae were developed by considering a linear behavior of inner shell and ribbon layers material, while in FEA using ANSYS software; we have to create area/surface contact pairs between the consecutive layers. The area/surface contacts show a highly nonlinear behavior which is a major cause of result variations.
6. In analytical formulation it was assumed that there should be no stress concentration but in case of FEA of the model using ANSYS software, we had merged keypoints and nodes of ribbon and inner shell with heads, to define welding which causes stress concentration at the weld areas.

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