Experimental Versus Numerical Investigation into the Effects of Hoop Welding in Metallic Pressure Vessels with Spherical Caps

J . Mirzaei
Department of Mechanical Engineering,
Tarbiat Modares University, Iran
E-mail: mirzaei.jaber@yahoo.com

G. H. Rahimi*
Department of Mechanical Engineering,
Tarbiat Modares University, Iran
E-mail: rahimi_gh@modares.ac.ir
*Corresponding author

S . Yazdani
Department of Mechanical Engineering,
Tarbiat Modares University, Iran
E-mail: yazdani.saleh@yahoo.com

Received: 19 June 2012, Revised: 27 June 2012, Accepted: 15 August 2012

Abstract: In this paper, experimental versus numerical investigation into the effects of hoop welding in a metallic pressure vessel is presented. In experimental manufacturing, hoop welding has been used to attach the cylindrical body of the vessel. Hence, the effect of welding, on stress and strain distributions has been studied. Experimental strains are obtained using strain gauges attached to predetermined places and stresses are calculated using Hook's law in composite materials. Results obtained from the two methods are compared.

Keywords: Stress Analysis, Pressure Vessel, Finite Element, Hoop Welding

Reference: Mirzaei, J., Rahimi, G. H., and Yazdani, S., "Experimental versus numerical investigation into the effects of hoop welding in metallic pressure vessels with spherical caps", Int J of Advanced Design and Manufacturing Technology, Vol. 5/No.4, 2012, pp. 19-26.

Biographical notes: J. Mirzaei is graduate student in Aerospace Engineering, from Tarbiat Modares University. His current research focuses on Pressure Vessels and composites. **G. H. Rahimi** is Professor of Mechanical engineering at Tarbiat Modares University. His current research focuses on plates and shells, FGMs, and composite materials. **S. Yazdani** is graduate student in Mechanical Engineering, from Tarbiat Modares University. His current research focuses on stiffened shells and composite materials.

1 INTRODUCTION

CNG pressure vessels are generally in cylindrical shape and feature two lens-shaped lids which are called domes. CNG pressure vessels are divided into four groups according to the ISO-11431 standard:

1) Metallic pressure vessels (Type 1)
2) Composite pressure vessels with metallic liner in which fibers wound only around the cylinder (Type 2)
3) Composite pressure vessels with metallic liner in which fibers wound around the whole cylinder (Type 3)
4) Composite pressure vessels with a plastic liner (Type 4)

These structures are frequently used in aerospace industries as in air battles or fuel tanks, so they must be designed accurately. In 1973, Kissmaul and Kraegeloh investigated the imperfection of welding in pressure vessels under internal pressure [1]. In 1991 Varga studied plastic pressure vessels [2]. The effect of welding on the occurrence of fatigue in metallic pressure vessels has been studied by Rading in 1993 [3]. Finite element analysis of the welding effect at the interface of the dome and the cylindrical part of the pressure vessel has been performed by Dekker and Brink [4]. Balasubramanian and Guha investigated the effect of welding on the crack growth of pressure vessels [5]. The stress analysis of pressure vessels under internal pressure is investigated by Khan in 2010 [6]. In 1995, Rahimi and Bitarafan investigated manufacturing methods of pressure vessels [7]. In the present paper a metallic pressure vessel (Air Battle) is being analyzed. The pressure vessel is seamless at the interface of the cylindrical shell and the domes, but environmental welding is being used at the middle of the cylinder to attach the two parts.

2 EXPERIMENTATION

Geometrical dimensions of the pressure vessel are presented in Table 1. Among different methods of the liner manufacturing, Cold Spinning method has been chosen to reduce manufacturing costs. This method, being one of the earliest methods in manufacturing pressure vessels, includes manual procedures and is implemented when manufacturing with precise tolerance is at hand.

Table 1 Geometrical dimensions of the pressure vessel

Length of the pressure vessel	380 mm
External diameter of the pressure vessel	158 mm
Nozzle's diameter	41 mm
Weld width	15 mm
Weld thickness	1.5 mm

In industry, the initial cross sectional tapering is performed at a high temperature, whereas final cross sectional tapering is performed at lower temperatures. By applying the two mentioned procedures, the specimen can benefit from both. In this method, a metal sheet covers the spinning mandrel in order to shape the mandrel. In manufacturing procedure, first, two semi-pressure vessels are manufactured and next these two parts are environmentally welded and are attached to each other. The specimen is made from Steel 304 and the thickness of the liner is calculated using Eq. (1):

$$t \geq \frac{D}{250} + 1 \tag{1}$$

where "D" is the pressure vessel's diameter. According to the pressure vessel diameter which is 158 mm, the minimum thickness of the liner is calculated to be 1.6 mm. Fig. 1 shows the manufactured liner.

Fig. 1 Manufacturing procedure

In the cold spinning process the material properties of the metal is changed. Therefore two standard specimens are cut from the material according to the ASTM-E8 standard and are tested by INSTRON 5500R universal machine. Fig. 2 shows the tensile testing of the specimens.

Fig. 2 Tensile testing of standard samples

The results of tensile testing procedures are presented in Fig. 3 and Fig. 4.

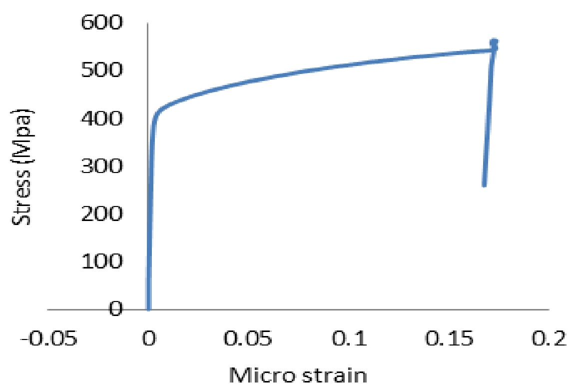

Fig. 3 Stress-strain diagram of specimen 1

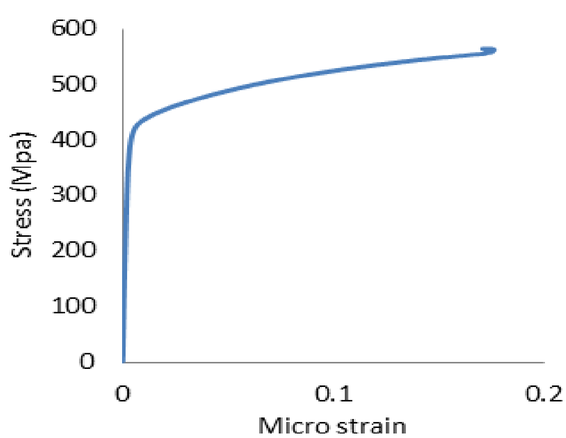

Fig. 4 Stress-strain diagram of specimen 2

Considering the results of tensile testing methods and using the ASTM E-8 standard, the obtained yield stress is found to be 420 MPa. In addition, the ultimate stress of the specimens is calculated to be 520 MPa. The significant difference between the yield stress and the ultimate stress indicates the strong effect of machining on the specimen. It can also be concluded that the steel material has approached to its elastic-perfect plastic region. Strain gauge technique is used for obtaining the strain gradient. For the purpose of data acquisition of strain values, an appropriate Data Logger device is used. Strain gauges are attached on the outer side of the specimen in two directions. Because of plane stress condition, strain gauges are attached in axial and hoop direction and the shear strain is being neglected. The testing procedure is performed under hydrostatic pressure. Fig.5 shows the position of strain gauges on the specimen, and schematic view of the testing setup is shown in Fig. 6.

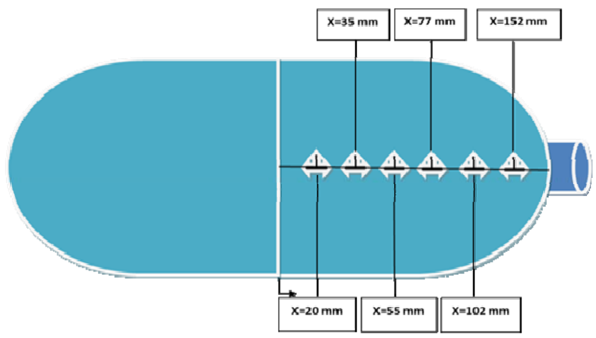

Fig. 5 Position of strain gauges

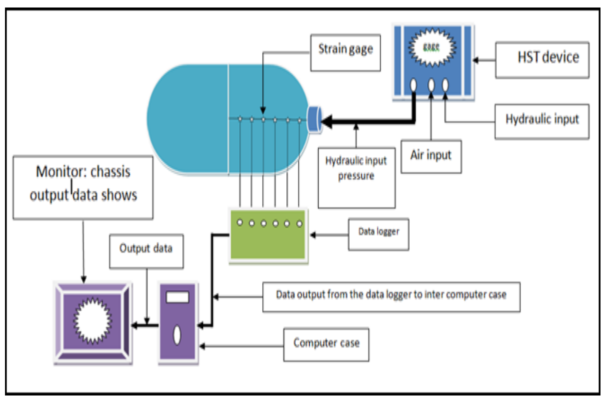

Fig. 6 Schematic diagram of test setup

Experimental strain distributions in axial and hoop directions are shown in Fig. 7 and Fig. 8. The stress distributions are obtained through experimental strain distributions and as well as the stress-strain equations in homogenous materials. Eq. (2) and Eq. (3) are used for obtaining the stress distributions.

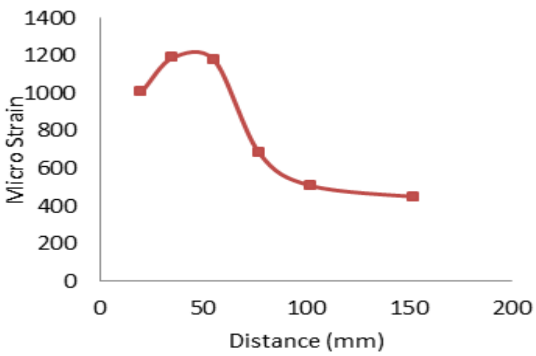

Fig. 7 Hoop strain distribution

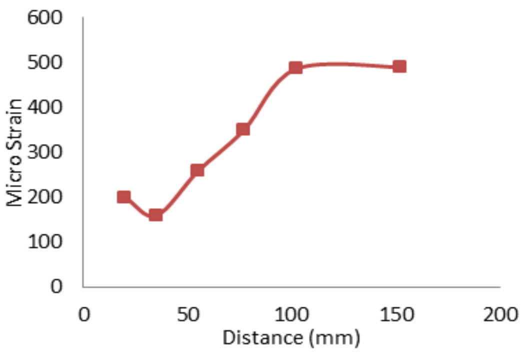

Fig. 8 Axial strain distribution

$$\sigma_{11} = \frac{E}{(1-v^2)}(\varepsilon_{11} - v\varepsilon_{22}) \quad (2)$$

$$\sigma_{22} = \frac{E}{(1-v^2)}(\varepsilon_{22} - v\varepsilon_{11}) \quad (3)$$

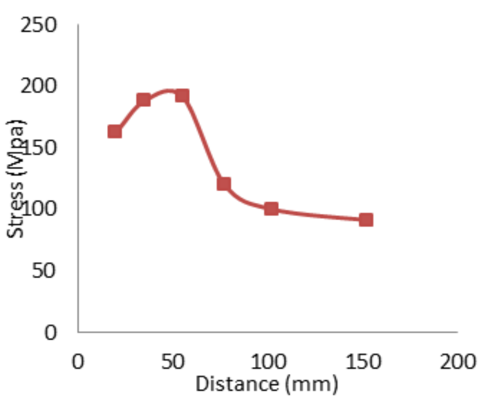

Fig. 9 Hoop stress distribution

Fig. 9 and Fig. 10 show the stress distributions in axial and hoop directions, respectively.

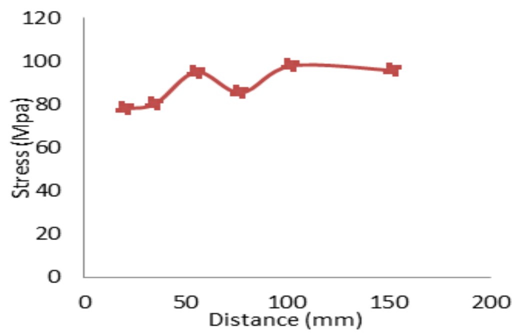

Fig. 10 Axial stress distribution

3 FINITE ELEMENT ANALYSIS

Finite element methods are among powerful prevalent methods for analysis of thin-walled structures. In this paper, ABAQUS commercial software is used for the purpose of numerical analysis. The specimen is modelled completely in the ABAQUS software. In the experimental manufacturing, the pressure vessel contains a nozzle and hoop welding at the intersection. In this paper the aim is to find the strain and stress distributions and the effect of hoop welding on changing the stress and strain distributions. Mechanical properties obtained during the tensile testing, are used for the modelling purpose. Table 2 shows the material properties of this special specimen.

Table 2 Mechanical properties of the specimen

Young modulus of liner	155 GPa
Young modulus of welding	190 GPa

The model was constructed using thick parabolic shell elements (S8R) which allowed a linear variation of stress across the element and by doing so, stresses through thickness brought into account. The pressure vessel is partitioned to lay the nodes of the shell and the weld part on each other.

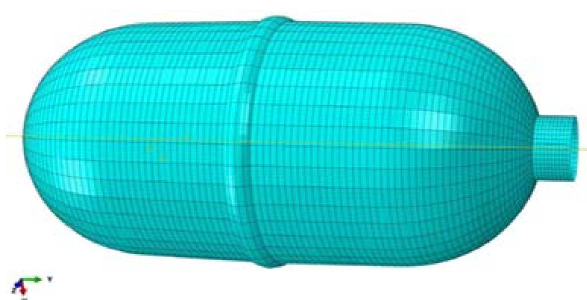

Fig. 11 Mesh model of the specimen

The arrangement and type of element scheme are changed several times to achieve the best element distribution. Fig. 11 shows the mesh model of the specimen.
The model does not need any constraint, except the elements on the outer surface of the nozzle, because it was modeled completely. The model was subjected to internal pressure of 40 bars. This pressure value has been chosen according to the ultimate stress of the specimen. For obtaining the stress and strain distributions, all the nodes from the middle of the cylindrical shell to the end of the dome were selected. Strain distributions in the hoop and axial direction of this path are shown in Fig. 12 and Fig. 13, respectively.

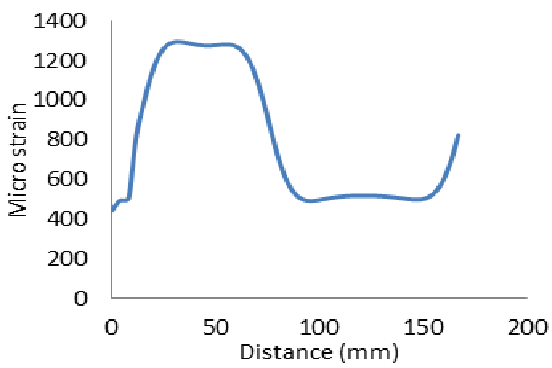

Fig. 12 Hoop strain distribution

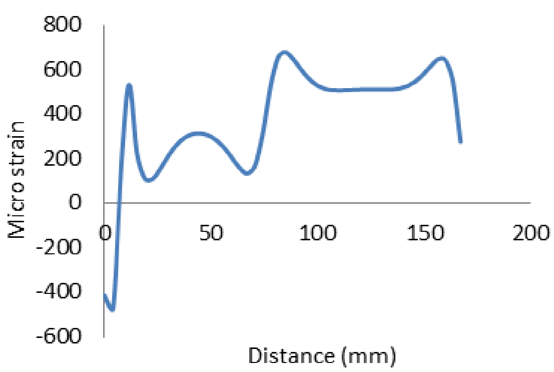

Fig. 13 Axial strain distribution

Stress distributions of the selected nodes are also shown in Fig. 14 and Fig. 15. As it is shown, the maximum changes in stress and strain distribution are related to the pressure vessel imperfection which is caused by welding, and at the interface of the cylindrical shell with the domes.

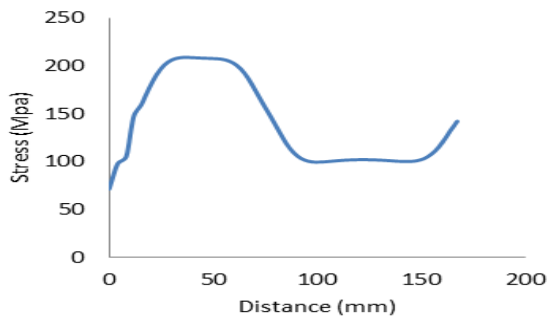

Fig. 14 Hoop stress distribution

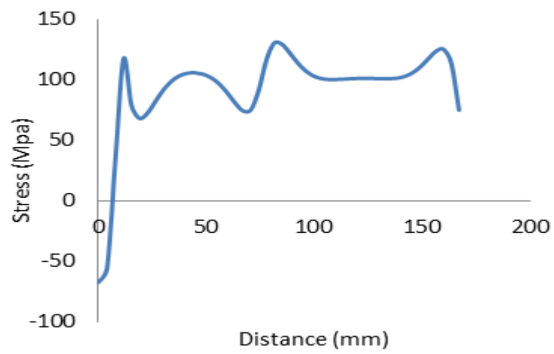

Fig. 15 Axial stress distribution

4 RESULTS AND DISCUSSION

The strain distributions in the metallic pressure vessel are obtained by two different approaches; namely, experimental and numerical analysis. In the experimental analysis, strain gauge technique is used, subsequently; numerical analysis is performed by ABAQUS commercial software. Strain gradient on the pressure vessel in experimental and numerical analysis are compared at the special pressure of 40 bars.

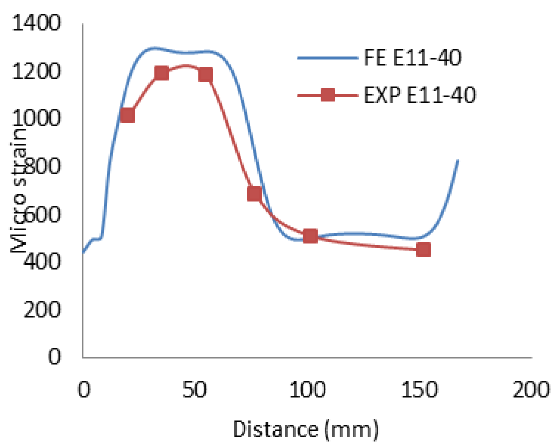

Fig. 16 Experimental and FE hoop strain

Fig. 16 and Fig. 17 show the strain gradient.

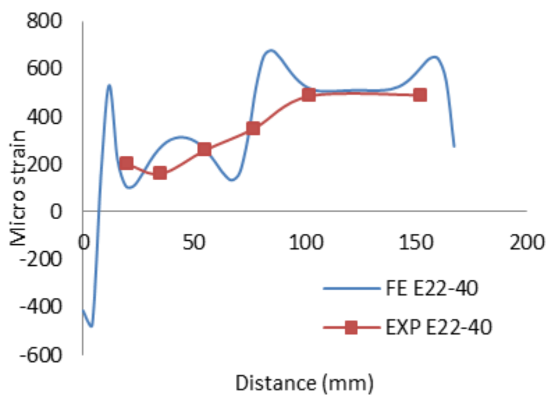

Fig. 17 Experimental and FE axial strain

Similar behaviour in both experimental and FE analysis is observed. It can be concluded that the maximum values of strain correspond to the interface of the shell and the domes as well as the welding region. The strain distribution in axial direction in experimental and FE analysis is irregular; this irregularity may be attributed to the effect of welding. Fig. 18 and Fig. 19 show the stress distribution in axial and hoop directions in the two approaches. Results indicate that the calculated stress distribution is close to the strain distribution. These two approaches indicate that the imperfection, caused by welding the pressure vessel, consequently affects the distribution of strain and stress. The next reason may be attributed to the change in cross sectional shape of the specimen at the imperfect welded region of the shell and the dome interface.

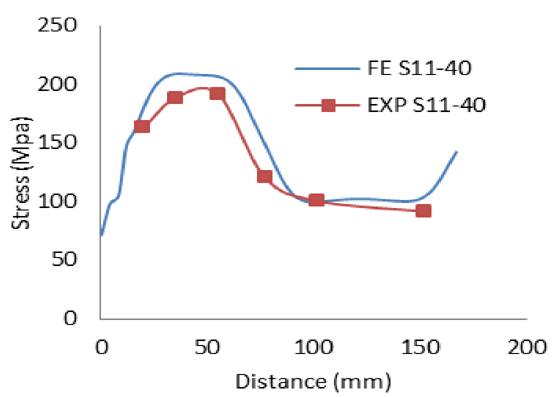

Fig. 18 Experimental and FE hoop stress

Contrary to the pressure vessels equipped with a nozzle, in which the axial pressure force is pushed to one end of the vessel, in nozzle-less pressure vessels; such axial force does not exist. Therefore, in analysing nozzle-less vessels, the stress distribution is regular. The stress and strain diagrams follow the same trend.

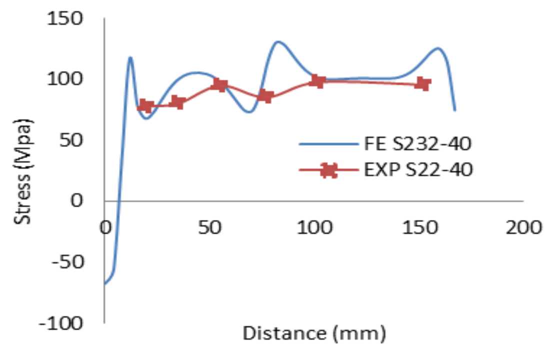

Fig. 19 Experimental and FE axial stress

In Fig. 20 the surface of the pressure vessel is divided into five regions and stress distribution in these regions are investigated.

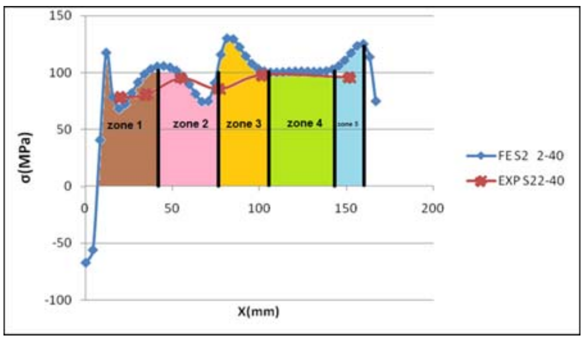

Fig. 20 Axial stress chart on the pressure vessel's surface

In the first zone, the change in stress distribution is related to the imperfection between the welding region and the shell. This imperfection causes the shell to confront bending forces at the interface consequently causing the stress gradient to change. In the absence of this imperfection, the change in the stress distribution could be neglected.

In the second zone, the changes in the stress distribution correspond to the geometrical imperfection between the dome and the cylindrical shell. According to the theoretical formulas, the effect of the spherical dome being placed on the cylindrical part, contributes to the maximum inconstancy of the distribution.

In the third zone, the differences between the values of stress distribution are related to the effect of the cylindrical part on the spherical domes. Based on the classical equations, the difference between these two values will be exceeding by 30 percent.

In the fourth zone, stress values of the experimental and theoretical formulation are the same. In the fifth zone, the geometrical imperfection between the nozzle and the dome causes the shell to be exposed to bending moment and, therefore, the changes are obvious. In this section the surface of the shell is divided into six regions. Fig. 21 shows the hoop stress distributions in these regions.

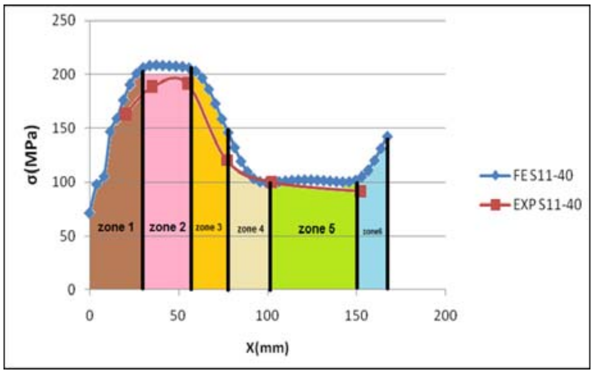

Fig. 21 Hoop stress chart on the pressure vessel's surface

In the first zone, the differences between the stress values are related to the imperfection between the pressure vessel and the welding regions. This imperfection causes the shell to be exposed to bending moment; thus the stress distribution is subject to inconstancies. In the second zone, the pressure vessel is exposed to membrane forces, so, the differences between strain and stress can be neglected. In the third zone, the difference between stress and strain is related to the imperfection between the dome and the cylindrical part. In the theoretical equations, it was observed that this difference is just 2 percent and the experimental results approve this issue.

In the fourth zone, the differences are related to the imperfection between the dome and the cylinder. The cylindrical part changes the stress gradient in the dome. In theoretical formulation, it was proved that the difference of minimum stress compared to maximum stress, exceeds 50 percent. In the fifth zone, membrane forces are much greater than other forces, so differences can be neglected.

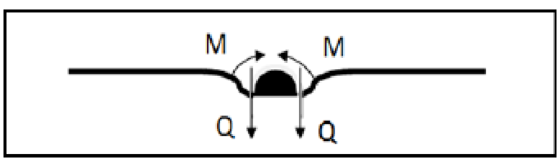

Fig. 22 Shear force and bending moment due to welding at the intersection

In the sixth zone, the geometrical imperfection between the nozzle and the pressure vessel causes the vessel to be exposed to bending moments. Therefore, the differences are significant. According to the imperfection at the welding region, by subjecting the pressure vessel to internal pressure, the imperfection causes the pressure vessel to be exposed to bending moment and shear forces (Fig. 22).

These two forces and moment cause a compression force at the upper section of the welding region and a tensile force at the intersection of the shell welding. The negative values of hoop stress in the FE results depict this problem. Moreover, the FE axial stress distribution which is shown in Fig. 23 shows this problem in the axial stress distribution.

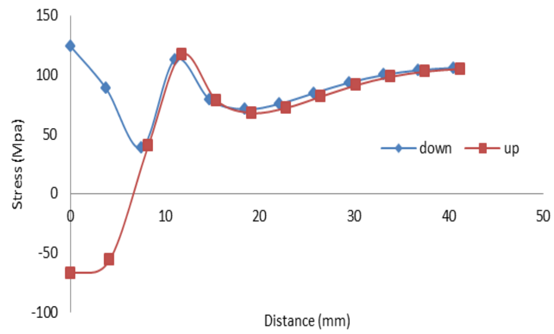

Fig. 23 FE axial stress distribution at upper and lower side of the welded region

5 CONCLUSION

In this research a metallic pressure vessel with hoop welding was designed, manufactured, and tested. The pressure vessel was subjected to internal pressure of 40 bars. The strains were obtained by electrical strain gauges. Classic laminate theory and netting analysis were used for obtaining the stress distributions. The finite element method is also used to compare the results. A good agreement between experimental and numerical results has indicated the correctness and capability of ABAQUS in modelling composite pressure vessels. Results are classified as below:

- Maximum amounts of strain and stress correspond to the cylindrical shell, independent of the pressure vessel length.
- The geometrical imperfection causes irregularities in the stress and strain distributions.
- Behaviour of the strain and stress are similar.

- Before experimental testing, finite element modelling is necessary.
- In the cylindrical part of the pressure vessel, the imperfection at the interface of the shell and dome causes the pressure vessel to be exposed to bending moments and this problem must be noticed before manufacturing the specimen.
- Cold Spinning method is a reliable and cost effective method for manufacturing these kinds of structures.

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