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Performance Enhancement of Metal Expansion Bellows for Shell and Tube Type Heat Exchanger

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ABSTRACT

The bellows are the flexible element of an expansion joint consisting of one or more convolutions and the end tangents. The number of convolutions in a bellows is in direct relationship to the amount of thermal or mechanical movement in the piping system or heat exchanger. Metal bellows are elastic vessels that can be compressed when pressure is applied or extended under vacuum. When the pressure or vacuum is released, the bellows will return to its original shape. In this project work, objective is to design & enhance the performance of the metal expansion bellows as per the ASME and EJMA. In this work Johnson Method is used to select the optimum material with objective of minimum weight and cost. As per Johnson Method the material selected for bellows under consideration was SAE 240 321 for minimum weight and cost as compared to materials like Inconal & Hastelloy X. Analytical analysis was done for bellow with 8, 9 and 10 convolution and it was found that Circumferential Membrane stress in bellow tangent (S1) is constant for bellow with 8, 9 and 10 convolution. Whereas End convolution circumferential membrane stress (S2E) and Intermediate convolution circumferential membrane stress (S2I) shows the linear decrease in stress as the number of convolution increases. In order to validate the findings required numerical analysis is also done by using ANSYS software and by performing extensive experimentation using Data Acquisition System. After comparing results it was found that all results are in close agreement.

Keywords— **Metal expansion joint, Johnson Method, FEA, Convolutions.**

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I. INTRODUCTION

Normally shell tube heat exchangers are operated on large temperature difference, internal pressure, vibration and aggressive corrosion environment. Due to the extreme working conditions the shell is always stressed under axial pressure, thermal load and vibration. This may lead to contraction and expansion of shell. If the rate of contraction and expansion increases then there may be chance of shell failure. In order to compensate the deformation an expansion joint is used in shell and tube heat exchanger. Normally metal bellows are used as an expansion joint in shell and tube heat exchanger. Metal bellows are elastic vessels that can be compressed when pressure is applied to the outside of the vessel, or extended under vacuum. When the pressure or vacuum is released, the bellows will return to its original shape.

The bellows is a very unique component of a piping system. It must be designed strong enough to accommodate the system design pressure, as well as, flexible enough to accept the design deflections for a calculated number of occurrences, with a minimum resistive force. The system pressure and deflection create the major stresses in a bellows. Typically the deflection stresses are higher than the pressure stresses and are meridional or longitudinal in direction. These stresses are calculated and evaluated in the Standards of the Expansion Joint Manufacturers Association, Inc. or EJMA. The material used for bellows are normally stainless steel, in rare case Inconel and Aluminium. The bellows are having wide shape verity like U-shaped, semi-toroidal, Toroidal, S-shaped, Flat, Stepped, Single Sweep, Nested Ripple.

II. LITERATURE SURVEY

Expansion joints used as an integral part of heat exchangers or other pressure vessels shall be designed to provide flexibility for thermal expansion and also to function as a pressure containing element. The rules in this Appendix 26 in ASME VIII Div 1 [1] are intended to bellows-type expansion joint geometries subject to internal pressure. And the most comprehensive and widely accepted text on bellows design is the Standards of Expansion Joint Manufactures Association (EJMA) [2] . The study on characteristics of stress can be found in the following papers.

The development of the metal expansion bellows with proper geometry is suggested by Hyun Wook Kang [3]. K. C. Leong, K. C. Toh, Y.C. Leong [4] gives information in their paper about a software for the thermal and hydraulic design of shell and tube heat exchangers with flow induced vibration checks has been developed in a Windows-based Delphi programming environment. Its user-friendly input format and excellent color graphics features make it an excellent tool for the teaching, learning and preliminary design of shell and tube heat exchangers. Design methodology is based on the open literature Delaware method while flow-induced vibration calculations adhere closely to the methodology prescribed by the latest TEMA standards for industry practice.

The forming technology with the mechanical behavior of the metal expansion bellow is also suggested by him. The analysis of buckling of metal expansion joints under internal pressure are suggested by D.E.Newland [5]. The analysis of two types of metal expansion joints movement test are suggested by Jorivaldo Medeiros [6]. Li [7] has investigated the effect of the elliptic degree of Ω-shaped bellows toroid on its stresses. The calculated stress results of Ω-shaped bellows with elliptic toroid correspond to experiments. The elliptic degree of Ω-shaped toroid affects the magnitude of internal pressure-induced stress and axial deflection-induced stress. Especially, it produces agreat effect on the pressure-induced stress. In order to keep the bellows strength and maintain its fatigue life,the toroid elliptic degree should be reduced greatly in manufacturing process, for example, at least lower than15%. The main key parameter responsible in the design and analysis of the metal expansion joint is the effective parameters of the metal expansion bellow are suggested by Gh Faraji [8] .

Kang et al. [9] proposed the forming process of various shape of tubular bellows using a single-step hydro forming process. The conventional manufacturing of metallic tubular bellows consists of four-step process: deep drawing, ironing, tube bulging, and folding. In their study a single step tube hydro forming combined with controlling of internal pressure and axial feeding was proposed. Those reviewed papers show that there are needs for rigorous analysis and forming parameters of bellows. It is stated that the Ω-shaped bellows have much better ability to endure high internal pressure than common U-shaped bellows. He suggested most effective parameters of the bellows are initial length of tube, internal pressure, axial feeding and velocity, mechanical properties and the type of materials by finite element (FE) analysis (LS-Dyna) and experimental tests.

G.Wang et al [10] developed a new technology uses super-plastic forming (SPF) method of applying gas pressure and compressive axial load. It is developed and can be used to manufacture large diameter “U” type bellows expansion joints made of titanium alloys. The forming technology for bellows expansion joints made of titanium alloys is presented to make a two-convolution bellows expansion joint of Ti–6Al–4V alloy thin plate of 1.28mm thickness. Welded pipe bent by a hot bending method with a set of specific dies and welded by PAW was used as a tubular blank in the SPF. During the SPF process the tubular blank is restrained in a multi-layer die block assembly which determines the final shape of convolution. The forming load route is divided into three steps in order to obtain optimum thickness distribution. This technology can also be used to fabricate stainless steel bellows expansion joints. The super-plasticity of Ti–6Al–4V titanium alloy is the best among of them, for instance the elongation can exceed 1000%. G I Broman [14] suggested I – DEAS software for the simulation of the metal expansion bellows before realization of the bellows. At the end of the review work now the author concluding that the remaining part of the metal expansion bellow is the key objective for the next research work. H.Shaikh [15] have performed an experimental work to analyze failure of an AM 350 steel bellows.

As per the literature survey and discussion with the expert in this field , it is very difficult to formulate the perfect design and investigate the selection of materials and shapes, vibration effect, joining of bellows to shell, stresses, flow analysis, fatigue life analysis and prediction of failure .As per the motivation and problem formulation following aim and objective were set.

conference website

III.METHODS

The bellows is a very unique component of a piping system. It must be designed strong enough to accommodate the system design pressure, as well as, flexible enough to accept the design deflections for a calculated number of occurrences, with a minimum resistive force.

The system pressure and deflection create the major stresses in a bellows. Typically the deflection stresses are higher than the pressure stresses and are meridional or longitudinal in direction. These stresses are calculated and evaluated in the Standards of the Expansion Joint Manufacturers Association, (EJMA).

The pressure stresses include circumferential (hoop) stress in the bellows tangent as well as the convolutions. EJMA defines the bellows tangent membrane stress due to pressure as S1. The bellows circumferential membrane stress due to pressure is designated as S2 in the EJMA calculations. The tangent stress (S1) and circumferential bellows stress (S2) must not exceed the maximum allowable stress, which is set as per the specification.

There are also meridional pressure stresses that are evaluated in the design of a bellows. The bellows meridional membrane stress due to pressure is designated as S3 in the EJMA calculations. The other meridional stress that is evaluated in EJMA is the bellows meridional bending stress due to pressure, or S4. If these meridional stresses are exceeded, the convolution sidewall will be overstressed and this will lead to bellows failure.

A. Terminology of Bellows

In order to comprehend the design process of bellows, it is very important that, the designer must have the knowledge of bellows terminology [Fig.1]. In this section different terms used in bellows design are explain in details as shown in Fig.1

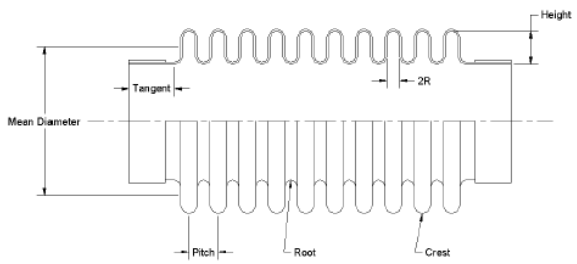

Fig.1. Terminology of Bellow

1) Convolution height: The height of the convolutions measured from the outside- preferably with slide gauge.

2) Convolution pitch: The distance between the convolutions measured from crest to crest.

3) Cycle: One complete cycle is based upon moving the bellows from neutral length to position 1,back through the neutral length to position 2 and then back to the neutral length.

4) Design Temperature: The maximum and minimum design, operating and installation temperatures should be accurately stated. In situations where the ambient temperature is expected to vary significantly during pipe line construction, special care in expansion joint positioning may be necessary.

5) Design Pressure: The system design pressure, operating pressure and test pressure should be specified realistically without the addition of arbitrary safety factors, because this practice necessitates greater bellows material thickness to withstand the overstated pressures. For standard metal expansion joints the PN (nominal pressure) factor can be defined as the allowed positive operating pressure at room temperature.

6) Design Allowable stress: This is the allowable stress for the bellows material at the bellows design temperature.

7) Modulus of elasticity at Design temperature: This is the modulus of elasticity of the bellow material at the design temperature which is used to calculate spring rate and columns squirm pressure.

8) *Modulus of elasticity at ambient temperature:* The room temperature modulus of elasticity is used to calculate the deflection stresses (S5 & S6).

B. Selection of Material

Selection of material is very much important from the design, weight and cost point because reliability and durability of design depends on the material only. The cost of material varies from material to material therefore it is necessary to select the best suitable material for the application for the purpose of design and cost point of view.

The material for our design can be selected on basis of one of the well known method of optimization that is "JOHNSON'S METHOD". For the purpose of minimizing weight and cost of bellow, Let us consider bellow as a thin cylinder with internal pressure. Procedure for finding the optimum material for bellow using Johnson's method is as follows:

1) Primary Design Equation (P.D.E) :

The most significant undesirable effect to be minimized is the weight of the bellow and given by

$$W = \rho g * volume$$

$$W = \rho g * \pi h \left(\frac{d_o^{\,2} - d_i^{\,2}}{4} \right)$$

2) Subsidiary Design Equation (S.D.E) :

$$\tau_{\max} = P_i \left(\frac{d_o^{\,2}}{d_o^{\,2} - d_i^{\,2}} \right)$$

$$\therefore d_o^{\,2} = \frac{\tau_{\max} d_i^{\,2}}{(\tau_{\max} - P_i)}$$

3) Limit Equation :

$$\tau_{\max} \le \frac{0.5 S_{yt}}{N_f}$$

4) Classification Of Parameters:

TABLE I

Classification Of Parameter

	Specified	Limited	Unspecified & Unlimited
Functional Requirement Parameter	P_i		
Undesirable Effect Parameter		τ_{max}	W
Geometrical Parameter	d_o, d_i	h	
Material Parameter	ρ , S_{yt}		

5) *Combining S.D.E & P.D.E :*

$$W = \frac{\rho g \pi h}{4}\left(\frac{\tau_{max} d_i^2}{(\tau_{max} - P_i)} - d_i^2\right)$$

$$W = \frac{\rho g \pi h d_i^2}{4}\left(\frac{P_i}{\tau_{max} - P_i}\right)$$

This is new developed P.D.E

6) *Combining Limit Equation with P.D.E :*

$$\tau_{max} \le \frac{0.5 S_{yt}}{N_f}$$

$$\therefore W = \frac{\rho g \pi h d_i^2}{4}\left(\frac{P_i N_f}{0.5 S_{yt} - P_i N_f}\right)$$

$$W = \frac{\rho * \pi * 9.81 * 0.115 * .156^2}{4}\left(\frac{1.099 * 10^6 * 1.5}{0.5 * S_{yt} + 1.099 * 10^6 * 1.5}\right)$$

$$W = 0.02156\rho\left(\frac{1.6485 * 10^6}{0.5 * S_{yt} + 1.6485 * 10^6}\right)$$

TABLE II

Material Selection Calculation Table

Material	SAE 240 321	Inconal	Hastelloy X
Mass Density (Kg/m^2)	8090	8440	8220
Tensile Strength (N/m^2)	$515*10^6$	$460*10^6$	$380*10^6$
Material Selection Factor	51.45	60.06	70.71
Weight 0.02156*(a) (W)(Kg)	1.1092	1.2948	1.5245
Rate/Kg (R)	270	285	260
Total Material Cost (W*R)	299.8/-	369./-	396.4/-

This is final P.D.E. In order to minimize 'W' , the material selection factor $\rho\left(\frac{1.6485*10^6}{0.5*S_{yt} + 1.6485*10^6}\right)$ should be minimize.

7) *Selection Of Material :*

From Table II it is concluded that the SAE 240 321 has minimum weight and cost as compared to Inconal & Hastelloy X, hence SAE 240 321 is selected for our design purpose.

C. Specification of U – Shaped Bellows

Table.3. shows different input parameters required in designing of U – Shaped Bellows

TABLE III

Different design parameters

Parameters	Symbol	Values
Expansion Joint Material		SA- 240 321
Material UNS Number		S32100
Bellows Design Allowable Stress(N / mm^2)	S	129.65
Bellows Ambient Allowable Stress(N / mm^2)	S_a	137.89
Bellows Yield Stress(N/mm^2)	S_y	157.39
Bellows Elastic Modulus at Design Temp (N / mm^2)	E_b	183090
Bellows Elastic Modulus at Ambient Temp (N / mm^2)	E_0	195121
Poisson's Ratio	υ_b	0.300
Bellows Material Condition		Formed
Design Cycle Life, Required number of cycles	Nreq	7000
Design Internal pressure	P	1.099
Design temperature for Internal Pressure		190 ^{0}C
Bellow Type		U Shaped
Bellows inside diameter(mm)	D_b	156.000
Convolution Depth (mm)	w	10
Convolution Pitch (mm)	q	9.44
Expansion joint opening per convolution (mm)	ΔQ	0.2985
Total number of convolutions	N	9
Nominal thickness of one ply (mm)	t	0.300
Total number of plies	n	3
End tangent Length (mm)	LT	13.000
Fatigue Strength Reduction Factor	Kg	1.500

IV. DESIGN OF BELLOWS

A. *Determination of U-shaped Bellows Parameters:*

In this section different parameter required to perform bellow design are determined.

1. *Mean diameter of bellows convolutions (D):*
   $= D_b + W + n * t$
2. *Thickness of one ply, corrected for Thinning during forming (tp):*
   $= t*\sqrt{D_b / D_m}$
3. *Convolution Pitch (q):*
   $$= \frac{N_q}{N} = \frac{85}{8} = 10.625mm$$
4. *Bellows cross sectional area of One convolution (A):*
   $$= \left\{\frac{(3.141612-2)}{2}*q+2*W\right\}*n*t_p$$
5. *Stiffening effect factor (k):*
   $$= MIN\left\{1, \frac{L_t}{\left[1.5*\sqrt{(D_b*t)}\right]}\right\}$$
6. *Coefficient (C1)*
   $$= \frac{q}{2*W}$$
7. *Coefficient (C2)*
   $$= \frac{q}{2.2*\sqrt{D_m*t_p}}$$

TABLE IV
Parameters Of Bellows

Types Of Stress	Convolutions		
	8	9	10
Convolution Pitch (q) (mm)	10.625	9.44	8.5
Bellows cross sectional area of One convolution (A) (mm^2)	22.676	22.1064	21.62
Coefficient (C_1)	0.5313	0.472	0.425
Coefficient (C_2)	0.6942	0.617	0.5

B. *Analytical Stress Analysis of U-shaped bellows:*

In this section stresses induced in different direction in U-shaped bellow are determined.

1. *Circumferential membrane stress in bellows tangent (S1) :*
   $$\frac{1}{2}\left\{\frac{L_t*E_b*K*P(D_b+n*t)^2}{[n*t*(D_b+n*t)*L_t*E_b+t_c*D_c*L_c*E_c*K]}\right\}$$
2. *End convolution circumferential membrane stress (S2E) :*
   $$\frac{1}{2}\left\{\frac{[q*D_m+L_t*(D_b+n*t)]*P}{(A+n*t_p*L_t+t_c*L_c)}\right\}$$
3. *Intermediate convolution circumferential membrane stress (S2I) :*
   $$\frac{1}{2}\left\{\frac{P*q*D_m}{A}\right\}$$
4. *Meridional membrane stress (S3) :*
   $$\frac{1}{2}\left\{\frac{W*P}{n*t_p}\right\}$$
5. *Meridional bending stress (S4) :*
   $$\left\{\frac{1}{2*n}\right\}*\left\{\frac{W}{t_p}\right\}^2*P*C_p$$
6. *Unreinforced Meridional stress summation :*
   $$S_3+S_4$$
7. *Unreinforced Bellow axial stiffness (spring rate) [Kb]*
   $$\frac{1}{2}\left[\frac{\pi}{(1-\upsilon_b^2)}\right]*\left(\frac{n}{N}\right)*E_b*D_m*\left(\frac{t_p}{W}\right)^3*\left(\frac{1}{C_f}\right)$$
8. *Allowable pressure for unreinforced column stability (Psc) :*
   $$0.34*3.1416*\left(\frac{K_b}{N*q}\right)$$
   $$\Delta = \frac{S_4}{3*S2I}$$
   $$\alpha = 1+2*\Delta^2+\sqrt{\left(1-2\Delta^2+4\Delta^4\right)}$$
9. *Allowable pressure for In- plane stability (Psi) :*
   $$(\pi-2)*A*\left(\frac{SYSTAR}{D_m*q*\sqrt{\alpha}}\right)$$

TABLE V
Summary of Analytical Results

Types Of Stresses	Convolutions			Allowable Streses
	8 (N/mm^2)	9 (N/mm^2)	10 (N/mm^2)	(N/mm^2)
(S1)	95.8	95.8	95.8	129.65
(S2E)	60.95	58.76	57.03	129.65
(S2I)	42.97	39.163	36.06	129.65
(S3 + S4)	139.12	149.82	167.32	388.94
(Psc)	3.84	3.34	3.39	5.153
(Psi)	2.36	2.17	2.04	3.450

S1 = Circumferential membrane stress in bellows tangent.

S2E = End convolution circumferential membrane stress.

S2I = Intermediate convolution circumferential membrane stress.

S3 + S4 = Unreinforced Meridional stress summation.

Psc = Pressure for unreinforced column stability.

Psi = Pressure for In- plane stability

Brief summary of analytically calculated values of stresses (as per EJMA & ASME Code) is given in TABLE V. Calculated stresses i.e, Circumferential membrane stress in bellows tangent (S1), End convolution circumferential membrane stress (S2E), Intermediate convolution circumferential membrane stress (S2I), Unreinforced Meridional stress summation (S3+S4), are within good

agreement when compared with allowable stresses. Hence performed design of bellow with 8,9 & 10 convolution is safe. TABLE V. shows that there Circumferential Membrane stress in bellow tangent(S1) is constant irrespective of increase or decrease of the number of stress. And End convolution circumferential membrane stress (S2E) and Intermediate convolution circumferential membrane stress (S2I) shows the linear decrease in stress as the number of convolution increases. Similarly Pressure for unreinforced column stability (Psc) also decreases as the number of the convolution increases. Whereas Unreinforced Meridional stress summation (S3 + S4) increases as the number of convolution increases.

C. Fatigue Calculation

1. Unreinforced Meridional membrane stress due to axial displacement (S 5) :

$$= E_b * (t_p)^2 * \frac{\Delta Q}{(2 * w^3 * C_f)}$$

2. Unreinforced Meridional bending stress due to axial displacement (S 6) :

$$= 5 * E_b * t_p * \frac{\Delta Q}{(3 * w^2 * C_d)}$$

3. Unreinforced total stress range due to cyclic displacement (St):

$$= 0.7 * (S_3 + S_4) + (S_5 + S_6)$$

$$K_g \text{ Factor} = K_g * \frac{E_0}{E_b} * S_t$$

4. Allowable number of Fatigue Cycles (Nallw) :

$$= \left\{ \frac{K_0}{\frac{K_g * E_0}{E_b * S_t} - S_0} \right\}^2$$

TABLE VI
Summary of Fatigue Results

Types Of Stresses	Convolutions		
	8 (N/mm2)	9 (N/mm2)	10 (N/mm2)
(S5)	1.45	1.44	1.4589
(S6)	206.1	153.6	152.44
(St)	304.93	260	271.02
(Nallw)	25740 Cycles	50978 Cycles	43215 Cycles

S5 = Unreinforced Meridional membrane stress due to axial displacement.
S6 = Unreinforced Meridional bending stress due to axial displacement.
St = Unreinforced total stress range due to cyclic displacement.
Nallw = Allowable number of Fatigue Cycles

TABLE.VI shows that Unreinforced Meridional membrane stress due to axial displacement (S 5) is approximately same for all the bellows with different convolution. Whereas Unreinforced Meridional bending stress due to axial displacement (S6) and Unreinforced total stress range due to cyclic displacement (St) goes on decreasing as the number of convolution increased from 8 to 10. Allowable number of Fatigue Cycles (Nallw) shows the anonymous behavior that is number of fatigue cycle first increases as the number of cycle increases from 8 convolution to 9 convolution but it decreases as the number of convolution is further increased from the 9 convolution to 10 convolution.

V.MODELLING & FEA ANALYSIS

Modeling and analysis of bellows with 8,9 and 10 convolution is done using ANSYS 14.0 software. Modeling of bellow is done using the Design Modular of ANSYS software and analysis is done using Multi-Physics module of ANSYS software. Axis symmetry method is used to do the analysis as bellow is symmetry about the axis and it reduces the number of nodes and element which ultimately the reduces the meshing and solving time. Analysis is done using Couple Field Analysis method as we have to calculate the thermal stresses by giving two boundary condition that is temperature (190 ^{0}C) and pressure (1.099MPa) .Fig.2 ,3 & 4 shows the Von Misses stress distribution for bellow with 8,9 & 10 convolution respectively.

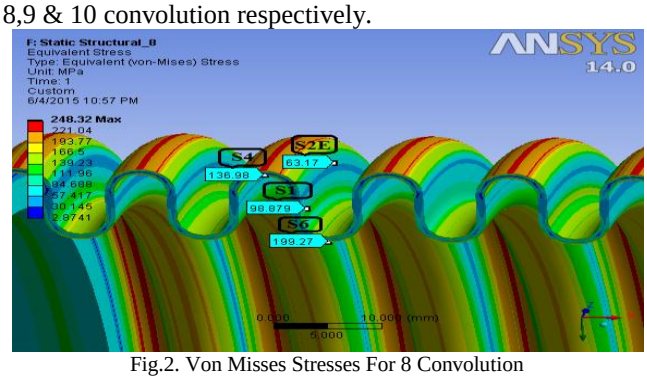

Fig.2. Von Misses Stresses For 8 Convolution

Fig.2.shows the Von Misses Stress for 8 Convolutions. Fig.2. shows the four stresses that is Circumferential membrane stress in bellows tangent (S1), End convolution circumferential membrane stress (S2E), Meridional bending stress (S4) and Unreinforced Meridional bending stress due to axial displacement (S 6) respectively.

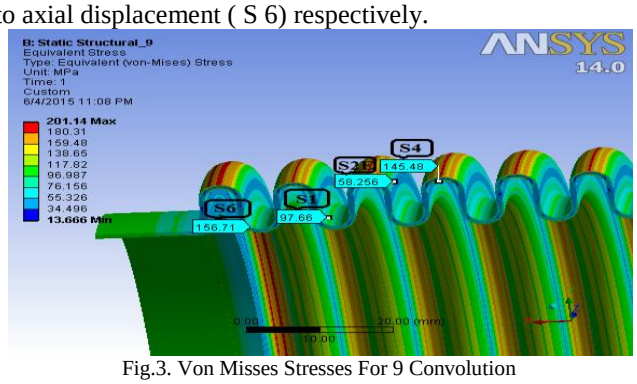

Fig.3. Von Misses Stresses For 9 Convolution

Fig.3.shows the Von Misses Stress for 8 Convolution. Fig.3. shows the four stresses that is Circumferential membrane stress in bellows tangent (S1), End convolution circumferential membrane stress (S2E), Meridional bending stress (S4) and Unreinforced Meridional bending stress due to axial displacement (S 6) respectively.

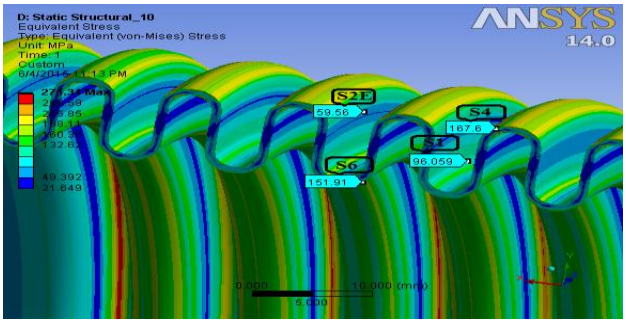

Fig.4. Von Misses Stresses For 10 Convolution

Fig.4.shows the Von Misses Stress for 8 Convolution. Fig.3. shows the four stresses that is Circumferential membrane stress in bellows tangent (S1), End convolution circumferential membrane stress (S2E), Meridional bending stress (S4) and Unreinforced Meridional bending stress due to axial displacement (S 6) respectively.

A. Comparison of Analytical and FEA Results

TABLE VII

.Comparison of Analytical and FEA Results

Types Of Stress	Convolutions					
	8 (N/mm2)		9 (N/mm2)		10 (N/mm2)	
	Analytical	FEA	Analytical	FEA	Analytical	FEA
S1	95.8	98.9	95.8	97.66	95.8	96.0
S2E	60.95	63.17	58.76	58.26	57.03	59.5
S4	132.8	136.9	143.5	145.4	161	167.6
S6	206.1	199.2	153.6	156.7	152.4	151.9

TABLE VII.shows the analytical and FEA stress value for Circumferential membrane stress in bellows tangent (S1), End convolution circumferential membrane stress (S2E), Meridional bending stress (S4) and Unreinforced Meridional bending stress due to axial displacement (S 6) and it found that the FEA results are in close agreement with the analytical results.

VI.EXPERIMENTAL SETUP

a In order to perform the experiment, we had developed the experimental setup. Experimental setup consist of boiler , bellow, strain gauges and NI Data Acquisition System. Boiler is used to give boundary condition that is pressure and temperature and NI Data Acquisition System is used to capture the data that is strain and stress with the help of strain gauges. Fig.6. show the schematic diagram of experimental setup.

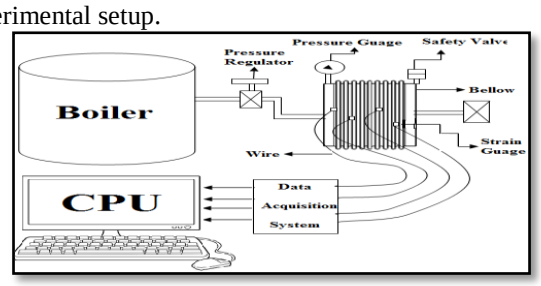

Fig.6.Schematic Diagram of Experimental Setup

In order to study the effect of pressure on bellow stresses and validate the results , the actual experiment was conducted on bellow with 10 convolutions by varying the pressure from 1 bar to 2 bar.Fig.7. and Fig.8. shows the actual picture of the experimental setup and Data acquisition system respectively.

Fig.7.Actual Experimental Setup

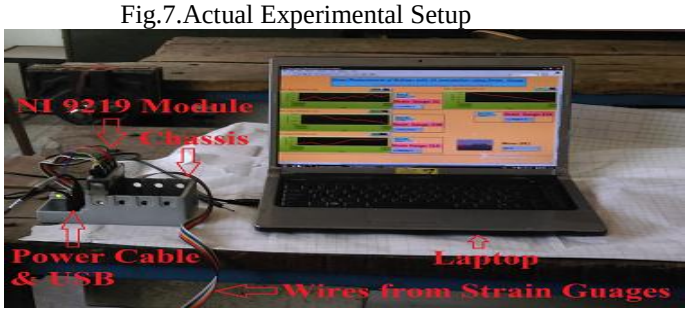

Fig.8. Data Acquisition System

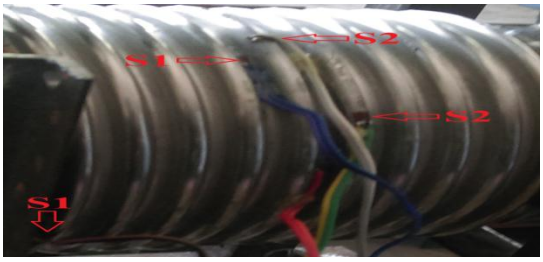

Fig.9.Location of Strain Gauges

S1 = Circumferential membrane stress in bellows tangent

S2 = End Convolution circumferential membrane stress

Fig.9. shows the location of strain gauges. We can get only two types of stresses in experimental setup that are Circumferential membrane stress in bellows tangent (S1) and end convolution circumferential membrane stress (S2) that are shown in Fig.9. Other stresses are meridional stresses which are on the cross sectional area hence cannot be evaluated. Following Figures shows the screen shot of the evaluated results from LabView software with different pressure condition.

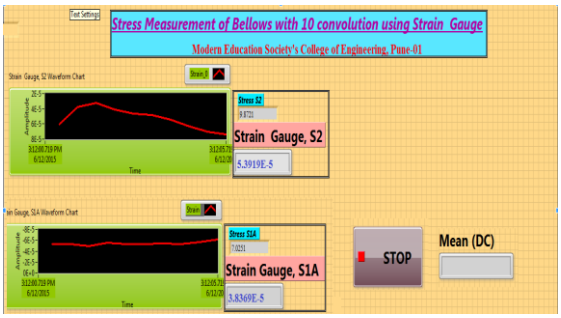

Fig.10.Results with Pressure 1 bar

Fig.11.Results with Pressure 1.25 bar

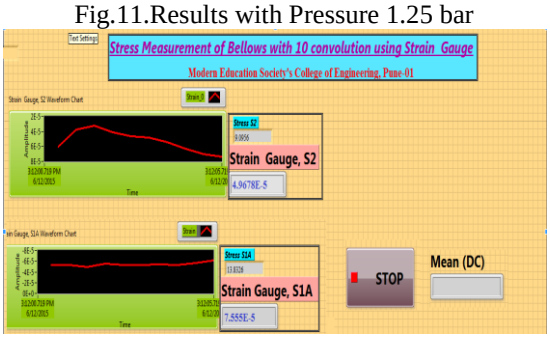

Fig.12.Results with Pressure 1.5 bar

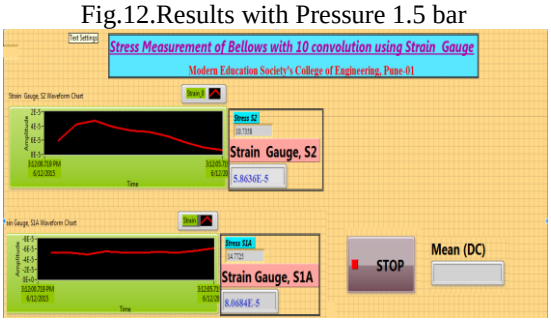

Fig.13.Results with Pressure 1.75 bar

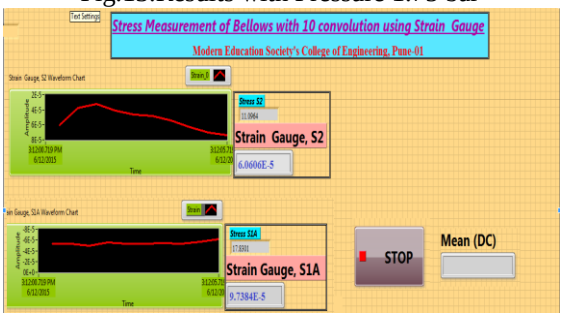

Fig.14.Results with Pressure 2 bar

Fig 10,11,12,13 & 14 shows the results of Labview software of with different pressures that are 1,1.25,1.5,1.75 & 2 bar respectively.

A. *Comparison of Analytical and Experimental Results*

TABLE VIII

.Comparison of circumferential membrane stress in bellows tangent at different pressure

Pressure (Bar)	Description	
	Analytical	**Experimental**
1	7.1611	9.8721
1.25	8.9513	11.4219
1.5	10.7416	13.8326
1.75	12.5319	14.7725
2	14.3222	17.8301

TABLE IX

.Comparison of End Convolution circumferential membrane stress at different pressure

Pressure (Bar)	Description	
	Analytical	**Experimental**
1	4.3889	7.0251
1.25	5.4862	8.9821
1.5	6.5834	9.0956
1.75	7.6807	10.7358
2	8.7779	11.0964

Table VIII & IX shows the value of circumferential membrane stress in bellows tangent and end convolution circumferential membrane stress at different pressure. It shows that as the pressure increases, stresses also increase simultaneously. It is also found that the experimental results are in close agreement with the analytical results.

VII.CONCLUSIONS

Material for bellows was selected on the basis of Johnson method and it was that SAE 240 321 is the optimum in terms of weight and cost. Bellows with 8,9 & 10 convolution is designed as the ASME and EJMA code. All the stresses are calculated and it is found that the stresses are under allowable limits, hence the is safe. Analytical results show that there Circumferential Membrane stress in bellow tangent (S1) is constant irrespective of increase or decrease of the number of stress. And End convolution circumferential membrane stress(S2E) and Intermediate convolution circumferential membrane stress (S2I) shows the linear decrease in stress as the number of convolution increases. Similarly Pressure for unreinforced column stability (Psc) also decreases as the number of the convolution increases. Whereas Unreinforced Meridional stress summation (S3 + S4) increases as the number of convolution increases. Unreinforced Meridional membrane stress due to axial displacement (S 5) is approximately same for all the bellows with different convolution. Whereas Unreinforced Meridional bending stress due to axial displacement (S6) and Unreinforced total stress range due to cyclic displacement (St) goes on decreasing as the number of convolution increased from 8 to 10. Allowable number of Fatigue Cycles (Nallw) shows the anonymous behaviour that is number of fatigue cycle first increases as the number of cycle increases from 8 convolutions to 9 convolutions but it decreases as the number of convolution is further increased from the 9 convolutions to 10 convolutions. In order to validate the findings required numerical analysis is also done by using ANSYS software and by performing extensive experimentation using Data Acquisition System. After comparing results it was found that all results are in close agreement.

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REFERENCES

[1] ASME, ASME Boiler and Pressure Vessel Code-Section VIII, Division 1, Appendix 26 – Pressure Vessel and Heat Exchanger Joints, New York, 2000.

[2] EJMA, Standards of Expansion Joint Manufacturers Association, ninth edition, New York, 2008S.

[3] K. C. Leong, K. C. Toh, Y.C. Leong, "Shell and tube heat exchanger design software for educational applications" , International Journal Engineering Education, Vol. 14, No. 3, Pages. 217-224, 1998.

[4] S.Igi "Evaluation of mechanical behavior of new type bellows with two directional convolutions", Nuclear Engineering and Design, Vol. 197, No. 1–2, Pages 107–114,2000.

[5] D.E.Newland "Buckling of double bellows expansion joints under internal pressure" Journal of mechanical engineering science, Vol 6 ,No 3, 1964.

[6] J.Medeiros "Analysis of FCC expansion joints movement test" Proceedings of the ASME 2009 Pressure Vessels and Piping Division Conference, Vol. 136, 2009M.

[7] T. Li., "Effect of the elliptic degree of Ω-shaped bellows toroid on its stresses", International Journal of Pressure Vessels and Piping, Vol. 75, No. 13, Pages.951-954,1998.

[8] G.H. Faraji, M.M. Mashhadi, and V.Norouzifard, "Evaluation of effective parameters in metal bellows forming process", Journal of Materials Processing Technology, Vol. 209, no. 7, Pages.3431-3437, 2009.

[9] B.H.Kang, M.Y. Lee, S.M.Shon, and Y.H. Moon, "Forming various shapes of tubular bellows using asingle-step hydroforming process", Journal of Materials Processing Technology, vol.194, no. 1-3, pp.1-6, 2007.

[10] G.Wang "Super plastic forming of bellows expansion joints made of titanium alloys" Elsevier Science, Journal of Materials Processing Technology, Vol 178,Pages 24-28, 19 Oct 2005.

[11] J. A Dreister, ''Problems in Developing Highly Efficient Tubular Heat Exchanger'', ISSN 0040-6015, Vol 53 no.4

[12] J. A. Guirao, Silvia Iglesias, Benéitez and S.L. J. PistonoIn, Research on ''New Method for Optimized Design of Heat Exchangers''.

[13] A.D.Andrade, G.Angelo, G.Fainer and E.Angelo, ''Assessment of a Heat Exchanger Inlet Nozzle Flow Using ANSYS-CFX'', Ansys south American conference 2010.

[14] G.I.Broman "Determining dynamic character of bellows by manipulated beam finite element of commercial software" Elsevier Science,Vol.77, Issue No.8,Pages 445-453, 2000.

[15] H. Shaikh, G. George, and H. S. Khatak, "Failure analysis of an AM 350 steel bellows", Engineering Failure Analysis, Vol. 8, no. 6, Pages 571-576, 2001.