Investigation of Stresses in U-Shaped Metal Bellow Using EJMA Standards

Kondapalli Siva Prasad

Department of Mechanical Engineering, Anil Neerukonda Institute of Technology & Sciences, India E-mail: kspanits@gmail.com *Corresponding author

Gudla Pavani

Department of Mechanical Engineering, Anil Neerukonda Institute of Technology & Sciences, India E-mail: pavanireddy.gudla@gmail.com

Received: 5 January 2017, Revised: 1 February 2017, Accepted: 5 February 2017

Abstract: Metal Bellows finds wide application in expansion joints, which are used in aerospace, chemical plants, power system, heat exchangers, automotive vehicle parts, piping system, petrochemical plant, refineries, etc. During service they are subjected to various stresses and exposed to different environments, which leads to failure. Hence there is a need for proper design of metal bellow as per the application. The main objective of the paper is to evaluate the stresses generated in the metal bellow and the cycle life working at different working pressures. In this paper, the stresses are calculated using Expansion Joint Manufacturing Association (EJMA) standards and compared with the results obtained using ANYS software for two different materials namely Inconel 625 and Inconel 718 for the pressure values ranging from 20 to 40 bar.

Keywords: EJMA, Expansion joints, Metal bellow

Reference: Prasad, K. S., Pavanai, G., "Investigation of Stresses in U-Shaped Metal Bellow Using EJMA Standards", Int J of Advanced Design and Manufacturing Technology, Vol. 10/No. 2, 2017, pp. 25-35.

Biographical notes: Kondapalli Siva Prasad obtained PhD from Andhra University, India in 2003. He is currently working as Associate Professor at the Department of Mechanical Engineering, Anil Neerukonda Institute of Technology & Sciences, India. His current research interest includes Manufacturing and product design. **Gudla Pavanai** is PG student of Department of Mechanical Engineering, Anil Neerukonda Institute of Technology & Sciences, India.

1 INTRODUCTION

An expansion joint is an assembly designed to safely absorb the heat-induced expansion and contraction of construction materials, to absorb vibration, to hold parts together, or to allow movement due to ground settlement or earthquakes. Bellow is corrugated part of the expansion joint which is capable of compensating large amount of axial, lateral and angular movements as a single unit. It must be strong enough circumferentially to withstand the pressure and flexible enough longitudinally to accept the deflections for which it is designed, and as repetitively as necessary with a minimum resistance. This strength with flexibility is a unique design problem that is not often found in other components in industrial equipment. Based on the application the material of the bellow is selected. Its present requirement is for the aerospace applications at the bleed air outlet of aircraft engine.

In the field of expansion joints very limited literature is available. Only few technical books and hand books of piping includes about the expansion joints which are used in the piping. But these references are limited up to the working principle of expansion joints. No text or reference books include, design of expansion joints, as this is a specialized area. But all authors are mentioning the reference of standards developed by EJMA. Since major contribution in the design of bellows expansion joint is given by Expansion Joints Manufacturers' Association (EJMA). EJMA has established the codes and guidelines for the design of bellows expansion joints. These codes are available based on membership of EJMA. Jayesh. B. Khunt and Rakesh. Prajapathi [1] studied different types of expansion joints used in industry.

S. H. Gawande et al. [2] performed numerical analysis to find various characteristics of stresses in U-shaped metal expansion bellows as per the requirement of vendor and ASME standards. Lu Zhiming et al. [3] discussed the effects of axial deformation load on Ushaped bellows. Brijeshkumar et al. [4] analyzed the failure of bellows expansion joints made of SS 304. Zhiming Lu et al. [5] analyzed the failure of metal bellow made of austenitic stainless steel. Kazuvuki Tsukimori [6] carried out modeling of creep behavior of bellows. Norton's law is used to study the creep property of bellows. K. Brodzinsko et al. [7] studied the failure mechanism of LHC cryogenic distribution line. Hasan Shaikh et al. [8] analyzed the failure of an AM 350 steel bellows. Jinbong Kim [9] analyzed the effect of geometry on fatigue life for automotive bellows. F. Elshawesh et al. [10] investigated that the expansion joint failed as a result of initiation of fatigue cracks at the corrosion pits that propagated through bellow's circumference. Bijayani Panda et al. [11] discussed the

metallurgical factors responsible for failure of bellows due to stress corrosion cracking. Asril Pramutadi et al. [12] observed the corrosion behavior conducted on the bellows of the bellow-sealed valve used in a lithium circulation loop. Abhay K. Jha et al. [13] observed various metallurgical features in stainless steel bellows. Y.Z. Zhu et al. [14] proposed the effect of environmental medium on corrosion fatigue life. C. Becht IV [15] predicted the fatigue life of bellows by partitioning the bellow fatigue data based on a geometry parameter.

In the above literature review most of the work is done on bellows made of various grade stainless steels which are subjected to different types of corrosion such as fatigue corrosion, liquid droplet erosion, and stress corrosion when used at high temperatures. Fatigue analysis of bellows is less concentrated. Based on these studies, there is a necessity that the materials used should possess great corrosion resistance at elevated temperatures. The fatigue life of the bellows is of great importance as they are subjected repeated loads. Therefore, these two high temperature nickel-chromium alloys Inconel 625 and Inconel 718 are used in our work. These materials have good oxidation resistance, excellent strength and are easily fabricated. As most of the bellows are used in corrosive environment use of these nickel chromium alloys will minimize the failure due to corrosion. So, these material properties are used to calculate the stresses produced in bellows. Hence this work focusses on selection of proper bellow material, design, calculation of stresses both analytically and numerically and finally comparing both with the allowable stress limits.

2 DESIGN OF BELLOW USING EJMA STANDARDS

The design of a bellow is complex and it involves an evaluation of pressure capacity, stresses due deflection and pressure, fatigue life, spring forces and instability. The bellow used in this joint will be tested for two high temperature materials. The design should be based on the actual bellow metal temperature expected during operation. The design values are considered based on conditions available at the bleed air outlet of aircraft engine. Detailed design calculations of bellow used in gimbal joint are shown below.



Fig. 1 Geometry of bellow

Design considerations

Normal Working Pressure 37 bar = Normal Working Temperature 650°C = Angular Moment required = 17.5Nm Maximum Permissible Deflection = $\pm 6^{\circ}$ D_m= Mean diameter of bellows convolution=50.5mm D_{b} = Inside diameter of bellows convolution = 42mm D_c = mean diameter of bellows tangent reinforcing collar=45 mm n = number of plies (Assume initially) = 4 t = Bellows nominal material thickness of one ply (Assume initially) = 0.25mm t_p = Bellows material thickness for one ply corrected for thinning during forming =0.228mm w = convolution height minus bellows thickness = 7.25mm Assume Number of convolutions N = 7 $L_{b} = Bellows convolute length = 43mm$ I. $L_t = Bellows tangent length$ = 6.5mm L_c = Bellows tangent collar length =6.5mm t_c = Bellows tangent reinforcing collar material thickness =1mm q = Pitch = 6.143 mm

e = Total equivalent axial moment per convolution = e_{θ_i} since only rotational movement is allowed= e_{θ} = 0.378 mm

k = A factor which considers the stiffening effect of the attachment weld and the end convolution on the pressure capacity of the bellows tangent and k value is calculated by using formula

$$k = \frac{L_t}{1.5\sqrt{D_b \times t}} = \frac{6.5}{1.5\sqrt{42 \times 0.25}} = 1.337$$

But if k>=1, k should be taken as 1 Hence k =1

2.1 The Stresses induced in bellow

The main causes for the stresses in the bellows are pressure and initial deflection. Pressure and deflection causes circumferential and meridional stresses in the bellows. Stresses due to internal pressure remain largely unaffected by the number of piles except for the convolution meridional bending stress, which are reduced when the total bellows thickness increases. The deflection stresses are reduced due to thinner material per ply resulting in an increase in fatigue life. The equations used below are based on norms followed by Expansion Joint Manufacturer's Association (EJMA) and accepted by ASME (American Society of Mechanical Engineers). The following are the stresses

- 1. Bellows Tangent Circumferential Membrane stress due to pressure (S₁)
- 2. Primary Collar Circumferential Membrane stress due to pressure (S_1^{-1})
- 3. Circumferential Membrane stress is also induced in the convolutions (S₂)
- 4. Bellows Meridional Membrane stress due to pressure (S₃)
- 5. Bellows Meridional Bending stress due to pressure (S₄)
- Bellows Meridional Membrane stress due to deflection(S₅)
- Bellows Meridional Bending stress due to deflection (S₆)

For Inconel 625 Material

I. Bellow tangent circumferential membrane stress due to pressure (S₁)

$$S_{1} = \left\lfloor \frac{P \times \left(D_{h} + n \times t\right)^{2} \times L_{t} \times E_{h} \times k}{2 \times \left\{n \times t \times E_{h} \times L_{t} \times \left(D_{h} + n \times t\right) + t_{c} \times k \times E_{c} \times L_{c} \times D_{c}\right\}} \right\rfloor$$

 $E_{b} = E_{c} = 16700 \text{ kgf/mm}^{2}$

 $S_1 = 3.89 \text{kgf/mm}^2$

 S_u = Ultimate tensile strength of Inconel 625 at design temperature (650°C) = 760Mpa or76 kgf/mm² S_{ab} = allowable material stress of Inconel 625 at design temperature = 76/2.5 =30 kgf/mm² C_w = Factor accounting for Welding joint efficiency = 0.7 Effective S_{ab} = $C_w \times S_{ab}$ = 21 kgf/mm² From the above calculations, it is clear that $S_1 < S_{ab}$. Hence design is safe.

II. Primary collar circumferential membrane stress due to pressure (S₁')

This is the circumference membrane stress induced in the collar directly due to pressure p

$$S_{I}' = \left[\frac{P \times D_{c}^{2} \times L_{t} \times E_{c} \times k}{2 \times \left\{n \times t \times E_{b} \times L_{t} \times \left(D_{b} + n \times t\right) + t_{c} \times k \times L_{c} \times D_{c}\right\}}\right]$$

 $S_1' = 4.25 \text{ Kgf/mm}^2$ Effective $S_{ab} = 21 \text{ kgf/mm}^2$ $S_1' < S_{ab}$ Hence design is safe.

C

III. Circumferential membrane stress induced in the convolutions (S_2)

Int J Advanced Design and Manufacturing Technology, Vol. 10/ No. 2/ June - 2017

$$S_{2} = \frac{P \times D_{m}}{2 \times n \times t_{p}} \left[\frac{1}{0.571 + 2 \times \frac{w}{q}} \right]$$
⁽¹⁾

$$S_{2} = 3.4 \text{ kgf/mm}^{2}$$

$$S_{2} < S_{ab}$$
(Thus design is safe)

IV. Meridional membrane stress in the bellow convolution is induced due to pressure (S_3)

It is a primary stress that follows the longitudinal axis of the bellows at the crest and root of the convolutions

$$S_{3} = \left[\frac{P \times W}{2 \times n \times t_{p}}\right]$$
$$S_{3} = 1.52 \text{ kgf/mm}^{2}$$

V. Meridional bending stress induced in the bellow convolution due to pressure (S_4)

It is a primary stress that follows the longitudinal axis of the bellows across the convolutions

$$S_4 = \left\lfloor \left(\frac{p}{2 \times n} \right) \times \left(\frac{w}{t_p} \right)^2 \times C_p \right\rfloor$$

The factor $C_p = 0.625$

 $S_4 = 31.29 kg / mm^2$

 C_m = Material strength factor at temperatures below the creep range

From EJMA standards, $C_m = 3.0$ for bellows in the formed condition (with cold work)

$$\begin{split} S_3 + S_4 &= 1.52 + 31.29 = 32.81 \text{ kgf/mm}^2 \\ C_m^* S_{ab} &= 3*21 = 63 \text{ kgf/mm}^2 \\ S_3 + S_4 < C_m^* S_{ab} \\ \text{Hence design is safe.} \end{split}$$

VI. Meridional membrane stress induced in the bellow convolution due to deflection(S₅)

It is a secondary stress since the applied load is limited by the deflection. It follows the longitudinal axis of the bellows.

$$S_{5} = \left[\frac{E_{b} \times t_{p}^{2} \times e}{2 \times w^{3} \times C_{f}}\right]$$

 E_b = 20800 kgf/mm² (at room temperature) C_f is a shape factor = 1.38 S_5 = 0.351kg/mm²

VII. Meridional bending stress in the bellow induced due to deflection (S_6)

It is a secondary stress and follows the longitudinal axis of the bellows. To find the value of the maximum moment, the convolution is modeled as a fixed guided strip beam with a concentrated load and a length w.

(2)
$$S_6 = \left[\frac{5 \times E_b \times t_p \times e}{3 \times w^2 \times C_d}\right]$$

$$C_d$$
 is shape factor = 1.78

$$S_6 = 29.84 kg / mm^2$$

The Stresses S_5 and S_6 are used in the evaluation of bellows fatigue life.

2.2 Column Squirm (Calculation of P_{sc})

Column Squirm is defined as a gross lateral shift of the centre section of the bellows. It results in curvature of curvature of the bellows centre line. This condition is mostly associated with bellows which have a relatively large length to diameter ratio and is analogous to the buckling of a column under compressive load. The bellows have to be designed for either elastic or inelastic condition based on length to diameter ratio.

For $L_b / D_b > = C_z$, the squirm pressure P_{sc} is evaluated as $\begin{bmatrix} 0.34 \times \Pi \times C & f \end{bmatrix}$

$$P_{sc} = \left\lfloor \frac{0.34 \times \Pi \times C_{\theta} \times f_{iu}}{N^2 \times q} \right\rfloor$$

For $L_b/D_b < C_z$, the squirm pressure P_{sc} is evaluated as

$$P_{sc} = \left\lfloor \frac{0.87 \times A_c \times S_y}{D_b \times q} \right\rfloor \left[1 - \frac{0.73 \times L_b}{C_z \times D_b} \right]$$

L_b /D_b = 43/42 = 1.02

$$C_{z} = Transition \ point \ factor = \sqrt{\frac{4.72 \times f_{iu} \times q^{2}}{S_{y} \times D_{b} \times A_{c}}}$$

This indicates the value of length to Diameter ratio where the critical instability pressure transitions to a maximum value at the length of one convolution which represents purely inelastic behavior.

Where

 S_y = Yield Strength at room temperature of bellows

For Inconel 625

 $S_y=49 \text{ kgf/mm}^2$ (at room temperature for Inconel 625)

$$f_{iu} = \left[\frac{1.7 \times D_m \times E_b \times t_p^3 \times n}{w^3 \times C_f}\right] = 116.75 \text{ kg/mm per}$$

convolution

 $D_b = 42 \text{ mm}$

 A_c =cross-sectional area of one bellows convolution = $(0.571 \times q + 2 \times w)t_n \times n = 16.88 \text{ mm}^2$

Substituting the values, C_z =0.776 Since L_b / $D_b \geq C_z$

$$P_{sc} = \left\lfloor \frac{0.34 \times \prod \times C_{\theta} \times f_{iu}}{N^2 \times q} \right\rfloor$$

 C_{θ} =column instability pressure reduction factor based on initial angular rotation

$$C_{\theta} = 1 - 1.822 \times \gamma + 1.348 \times \gamma^2 - 0.529 \times \gamma^3$$

$$\gamma =$$
 Ratio of initial to final angular rotation
= $\frac{\theta \times D_m}{\theta \times D_m + 0.3 \times L_b} = 0.2907$

b) $C_{\theta} = 0.5713$ $P_{sc} = 0.236 \text{ kgf/mm}^2$

2.4 In-plane Squirm (Calculation of P_{si})

This is defined as a shift or rotation of the plane of one or more convolutions such that the plane of these convolutions is no longer perpendicular to axis of the bellows. It is characterized by tilting or warping of one or more convolutions. The stress induced due to this squirm is evaluated as follows

$$P_{si} = \left[\frac{0.51 \times S_{y}}{K_{2}\sqrt{\alpha}}\right]$$

where

 P_{si} = Limiting Design Pressure based on Inplane instability (both ends rigidly supported)

b) $K_2 =$ Inplane Instability factor

$$K_{2} = \frac{D_{m}}{2 \times n \times t_{p}} \left[\frac{1}{0.571 + 2\frac{w}{q}} \right] = 9.189$$

 α = Inplane instability stress interaction factor

$$= 1 + 2 \times \delta^{2} + (1 - 2 \times \delta^{2} + 4 \times \delta^{4})^{0.5}$$

Inplane instability stress ratio= $\delta = \frac{K_{4}}{K_{4}}$

Inplane instability factor $K_4 = \frac{C_p}{2 \times n} \left[\frac{w}{t_p} \right]^2 = 84.53$

$$\alpha = 38.13$$

For Inconel 625

$$S_y = 49 \text{ kgf/mm}^2$$

 $P_{si} = 0.44 \text{ kgf/mm}^2$

For all the materials a factor of safety for limiting stress of 2.25 is used in the relation for P_{sc} , P_{si} . As P_{sc} , P_{si} < Normal Working pressure (37 bar), theoretically it is required to go for higher thickness, however there bellows were tested for burst pressure of 125 bar 'g' and found satisfactory.

2.3 Fatigue Life

Fatigue life of a bellow is a function of the sum of the meridional pressure stresses range and the total meridional deflection stresses range. The equation for fatigue life is as follows.

$$N_c = \left(\frac{c}{S_t - b}\right)^a$$

where a, b and c are material and manufacturing constants. These constants are derived from the graph of total stress range S_t versus number of cycles N_c . From EJMA standards, the values of a, b and c are as a=3.4, b=54,000, $c=1.86 \times 10^6$ Total Stress $S_t = 0.7$ ($S_3 + S_4$) + ($S_5 + S_6$) $S_3 = 1.52$ kgf/mm², $S_4 = 31.29$ kgf/mm²

For Inconel 625

$$\begin{split} S_5 &= 0.351 \ \text{kgf/mm}^2 \ , \ S_6 &= 29.84 \ \text{kgf/mm}^2 \\ S_t &= 53.158 \ \text{kgf/mm}^2 \\ = 77099.161 \ \text{psi} \\ N_c \ (\text{at design temperature}) &= 3.02 \times 10^6 \ \text{cycles} \end{split}$$

3 NUMERICAL INVESTIF\GATION OF STRESSES

At first the bellow surface model is designed with the help of CATIA V5 software, which is one of the leading design software, and after that the surface model is saved in IGES format and the geometry is imported to ANSYS software. After importing the geometry, the material properties are given with a thickness of 1 mm. The bellow part is analyzed with the help of ANSYS WORKBENCH 15.0. Figure 2 shows the surface model and figure 3 shows the meshed ANSYS model with the thickness given. The loading conditions are given such that one edge of the bellow is fixed and internal surface is subjected to pressure varying from 20 bar to 40 bar for the materials Inconel 625 and Inconel 718. Figure 4 shows the loading conditions of bellow.



Fig. 2 The surface model

29

 $S_5(kgf/mm^2)$

 $S_6(kgf/mm^2)$



Fig. 3 The meshed model



Fig. 4 The loading conditions

2 RESULTS & DICUSSIONS

4.1 Theoretically calculated Values

20 Bar

2.10

2.30

At Pressure

 $S_1(kgf/mm^2)$

 S_1 (kgf/mm²)

Theoretical values of design stresses, squirm values and fatigue life values for the two materials at 4 different pressures are calculated using EJMA standards.

Table 5.1	Stresses	developed at	t different	pressures	in Inconel
625					

30 Bar

3.152

3.452

37 Bar

3.887

4.25

40 Bar

4.20

4.602

At Pressure	20 Bar	30 Bar	37 Bar	40 Bar	
$S_1(kgf/mm^2)$	2.10	3.152	3.89	4.202	
$\hat{S}_1(kgf/mm^2)$	2.30	3.452	4.25	4.602	
$S_2(kgf/mm^2)$	1.838	2.757	3.4	3.675	
$S_3(kgf/mm^2)$	0.822	1.234	1.52	1.645	
$S_4(kgf/mm^2)$	16.91	25.36	31.29	33.81	

0.337

28.692

0.337

28.692

0.337

28.692

Table 5.2 Stresses developed at different pressures in Inconel

4.1.1 Design Stresses induced in Bellow

0.337

28.692

In the above tables 5.1, and 5.2, the stresses developed in two high temperature metals at four different pressures are calculated and it is observed that the stresses S_1 , S_1^{-1} , S_2 , S_3 , S_4 are increasing within increase in pressure whereas the deflection stress S_5 , S_6 are calculated by using room temperature material properties which does not have any significant change with the change in pressure and vary according to the material used. These stress values are checked with the allowable stress values and the design is found to be safe.

4.1.2 Column and In-plane Squirm

A factor of safety for limiting stress of 2.25 is used in the relation for P_{sc} and P_{si} . As P_{sc} , P_{si} < 2.25 times of working pressure, theoretically it is required to go for higher thickness but these bellows were tested for burst pressure of 125 bar and are found satisfactory.

Table 5.3 Squirm values for different materials						
INSTABILITY	Inconel	Inconel				
Column Squirm	0.236	0.23				
Inplane Squirm P _{si}	0.44	1.056				

4.1.3 Fatigue Life

$S_2(kgf/mm^2)$	1.84	2.757	3.4	3.676	Table 5.4 Fatigue life values (number of cycles N_c)				
S ₃ (kgf/mm ²)	0.822	1.23	1.52	1.645	Pressure (Bar)	20	30	37	40
$S_4(kgf/mm^2)$	16.91	25.36	31.29	33.81	Inconel	1.2×10 ⁸	8.94×10 ⁶	3.022×10 ⁶	2.078×10 ⁶
S ₅ (kgf/mm ²)	0.351	0.351	0.351	0.351	625	2.78×10^8	1.281×10^{7}	3 908×10 ⁶	2.613×10^{6}
S ₆ (kgf/mm ²)	29.84	29.84	29.84	29.84	718	2.70/10	1.201/10	5.900/10	2.015/(10

From theoretical calculations, it is observed that fatigue life values depend upon the working pressure as the pressure increases fatigue life values decreases. Inconel 718 has better fatigue life at all the pressures.

4.1 Comparison of Analytical and Numerical Stresses due to Internal Pressure for Inconel 625 and Inconel 718

The analytical stresses obtained from EJMA standards and numerical stresses obtained from FEA are compared and tabulated below.

4.2.1 For Inconel 625

STRESS SOURCE 20 30 37 40 Bar Bar Bar Bar $S_1(kgf/mm^2)$ EJMA 2.10 3.152 3.887 4.20 FEA 9.21 13.84 16.47 18.18 $S_1^1(kgf/mm^2)$ EJMA 2.30 3.45 4.25 4.602 FEA 8.41 13.54 15.72 17.14 2.76 S₂(kgf/mm²) EJMA 1.84 3.4 3.675 FEA 11.45 17.29 20.942 23.18 $S_3(kgf/mm^2)$ EJMA 0.822 1.23 1.52 1.644 FEA 11.14 16.43 20.97 22.85 S₄(kgf/mm²) EJMA 16.91 25.36 31.29 33.81 FEA 13.94 20.42 25.35 28.38

From the table 5.5 it is observed for Inconel 625 that all the stresses obtained by both the approaches are within the allowable limit and are increasing with increase in the pressure. The difference in the stress profile is comparatively large and is due to variation of the approach methods.



Fig. 5 Circumferential stresses in bellow tangent (S₁) for Inconel $625.S_1 < S_{ab}(21 \text{ kgf/mm}^2)$



Fig. 6 Primary collar circumferential stress (S_1^{-1}) for Inconel 625. $S_1^{-1} < S_{ab}$ (21 kgf/mm²)



Fig. 7 Circumferential membrane stress induced in convolution (S₂) for Inconel 625 $S_2 < S_{ab} (21 \text{ kgf/mm}^2)$



Fig. 8 Meridional membrane stress (S₃) for Inconel 625



Fig. 9 Meridional bending stress (S₄) for Inconel 625 $(S_3+S_4) < C_m \times S_{ab}(63 \text{ kgf/mm}^2)$

© 2017 IAU, Majlesi Branch

Int J Advanced Design and Manufacturing Technology, Vol. 10/ No. 2/ June – 2017

The allowable stress value (S_{ab}) for circumferential stresses $S_1,\ S_1{}^1, S_2$ is 21 kgf/mm². Whereas for meridional stresses $(S_3+S_4) < C_m \times S_{ab}$ i.e.63 kgf/mm² for the design to be safe. Figures 5 to 9 show the comparison graphs of the stresses in the bellow when subjected to internal pressure.



Fig. 10 Stress distribution at pressure 20 Bar



Fig. 11 Stress distribution at pressure 30 Bar



Fig. 12 Stress distribution at pressure 37 Bar



Fig. 13 Stress distribution at pressure 40 Bar

When compared to the meridional bending stress (S_4) all the other stresses have considerable variation, but as per design criteria they are within the allowable limit. The circumferential membrane stress induced in convolution (S_2) for the pressure 40 bar is slightly above the allowable stress which states that withstanding the pressure more than 40 bar there is a necessity to go for higher thickness of the bellow. Figures 10 to 13 show the stress distribution for Inconel 625 at different pressures. It is also incurred from the diagrams that the maximum and minimum values of stresses developed in bellow when compared to theoretically calculated stresses show a close match.

4.2.2 For Inconel 718

From the table 5.6 it is observed for Inconel 718 that all the stresses obtained by both the approaches are within the allowable limit and are increasing with increase in the pressure. The difference in the stress profile is comparatively large and is due to variation of the approach methods.

STRESS	SOURCE	20 Bar	30 Bar	37 Bar	40 Bar
S ₁ (kgf/mm ²)	EJMA	2.10	3.152	3.89	4.202
	FEA	9.77	15.06	18.63	20.18
$S_1^{1}(kgf/mm^2)$	EJMA	2.301	3.452	4.25	4.602
	FEA	8.39	14.76	17.58	19.51
$S_2(kgf/mm^2)$	EJMA	1.838	2.757	3.4	3.676
	FEA	14.15	21.46	26.36	28.50
S ₃ (kgf/mm ²)	EJMA	0.822	1.234	1.52	1.645
	FEA	13.47	19.29	23.51	23.60
S ₄ (kgf/mm ²)	EJMA	16.907	25.361	31.29	33.814
	FEA	15.12	22.58	29.06	31.37

 Table 5.6 Analytical and Numerical stresses of Inconel 718 at different pressures

Int J Advanced Design and Manufacturing Technology, Vol. 10/ No. 2/ June – 2017

Figures 14 to 18 show the comparison graphs of the stresses in the bellow when subjected to internal pressure in Inconel 718. When compared to the meridional bending stress (S_4) , all the other stresses have considerable variation, but as per design criteria, they are within the allowable limit.



Fig. 14 Circumferential stresses in bellow tangent (S₁)for Inconel 718S₁ <S_{ab}(32.34 kgf/mm²)



Fig 15 Primary collar circumferential stress (S_1^{-1}) for Inconel 718S₁¹< S_{ab} (32.34 kgf/mm²)



 $\begin{array}{ll} \mbox{Fig. 16} & \mbox{Circumferential membrane stress} \\ \mbox{induced in convolution } (S_2) \mbox{ for Inconel 718 } S_2 \\ & < S_{ab} (32.34 \mbox{ kgf/mm}^2) \end{array}$



Fig. 17 Meridional membrane stress (S₃) for Inconel 718



 $\begin{array}{lll} \mbox{Fig. 18} & \mbox{Meridional bending stress} (S_4) \mbox{ for Inconel} \\ & 718(S_3{+}S_4) \ {<}C_m{\times}S_{ab}(97.02 \ kgf/mm^2) \end{array}$

Figures 19 to 22 show the stress distribution for Inconel 718 at different pressures varying from 20 bar to 40 bar. It is also incurred from the diagrams that the maximum and minimum values of stresses developed in bellow at these pressures show a close match.



Fig. 19 Stress distribution at Pressure 20 Bar

© 2017 IAU, Majlesi Branch



Fig. 20 Stress distribution at Pressure 30 Bar



Fig. 21 Stress distribution at Pressure 37 Bar



Fig. 22 Stress distribution at Pressure 40Bar

6 CONCLUSIONS

In this paper the design of the metal bellow and theoretical evaluation of the stresses is done using EJMA standards and the numerical evaluation is done by using ANSYS WORKBENCH software. The results obtained are compared and there is slight variation in the stress values and is due to different approaches followed. From the theoretical calculations it can be said that the maximum stress value is Meridional bending stress and this value is checked with the allowable stress value so that the bellow does not fail.

REFERENCES

- [1] Jayesh. B. Khunt, Rakesh. P. Pajapathi, "Design and Thermal Analysis of Thermal Expansion Joint in Industrial Application", International Journal for Innovative Research in Science & Technology, Volume 1, Issue 6, 2014, pp.63-65.
- [2] S. H. Gawande, N. D. Pagar, V. B. Wagh, and A. A. Keste, "Numerical Investigations on Characteristics of Stresses in U-Shaped Metal Expansion Bellows", International Journal of Metals, Volume 2015, Article ID 957925, 2015, pp.1-9.
- [3] Lu Zhiming, Jin Jie, Gao Zengliang, and Qin Yi, "Effects of Axial Deformation Load on In-Plane Instability of U-Shaped Bellows", Journal of Pressure Vessel Technology, Volume.125, 2003, pp.475-477.
- [4] Brijeshkumar. M. Patel, V. A. Patel, "Design, Manufacturing and Analysis of Metal Expansion Bellows", International Journal of Engineering Science and Innovative Technology (IJESIT) Volume 2, Issue 3, 2013, pp. 266-273.
- [5] Zhiming Lu, Zhaoai Li, Chenhui Zhu and Honggang Gao, "Failure Analysis of Expansion Joint Bellows of Gas Pipelines", ASME 2011 Pressure Vessels and Piping Conference Volume 6: Materials and Fabrication, Parts A and B, Paper No. PVP2011-57199, 2011, pp. 891-894.
- [6] Kazuyuki Tsukimori, "Theoretical Modeling of Creep Behavior of Bellows and Some Applications", Journal of Pressure Vessel Technology, Volume.123, 2001, pp.179-190.
- [7] K. Brodzinski, P. Cruikshank, J.L. Fournel, L. Tavian, N. Veillet, "Failure Mechanism and Consolidation of the Compensation Bellows of the LHC Cryogenic Distribution line", Physics Procedia Volume. 67, 2014, pp. 129 – 134.
- [8] Hasan Shaikh, Geogy George, Hawa Singh Khatak, "Failure Analysis of an AM 350 Steel Bellows", Engineering Failure Analysis, Volume. 8, 2000, pp. 571-576.
- [9] Jinbong Kim, "The Effect of Geometry On Fatigue Life for Bellows Advanced Materials Development and Performance (AMDP2011)", International Journal of Modern Physics: Conference Series, Volume. 6, 2012, pp.343-348.
- [10] F. Elshawesh, A. Elhoud, W. Zeglam, K. Abusowa, A. Mesalem, "Corrosion Fatigue of Inconel 825 Flare Gas Line Bellows of Expansion Joints", Journal of Failure Analysis and Prevention, Volume. 15, Issue. 1,2014, pp. 7 -14.
- [11] Bijayani Panda, M. Sujata, M. Madan, S.K. Bhaumik, "Stress Corrosion Cracking in 316L Stainless Steel

Bellows of a Pressure Safety Valve", Engineering Failure Analysis, Volume.36, 2013, pp. 379–389.

- [12] Asril Pramutadi, Andi Mustari, Minoru Takahashi, "Metallurgical Analysis of Corroded Bellows of Bellow-Sealed Valve in Lithium Flow System", Fusion Engineering and Design, Volume. 88, 2013, pp. 202– 208.
- [13] Abhay K. Jha, V. Diwaker, K. Sreekumar, "Metallurgical Investigation on Stainless Steel Bellows Used in Satellite Launch Vehicle", Engineering Failure Analysis, Volume. 13, 2005, pp. 1437–1447.
- [14] Y.Z. Zhu, H.F. Wang, Z.F. Sang, "The Effect of Environmental Medium on Fatigue Life for U-Shaped Bellows Expansion Joints", International Journal of Fatigue, Volume.28, 2004, pp.28–32.
- [15] C. Becht IV, "Fatigue of Bellows, a new Design Approach", International Journal of Pressure Vessels and Piping, Volume.77, 2000, pp.843-850.
- [16] EJMA, "Standards of Expansion Joint Manufacturers Association, Expansion Joint Manufacturers Association", New York, NY, USA, 9th edition, 2008.
- [17] www.haynesintl.com