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

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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

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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 U-shaped 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. Kazuyuki 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.

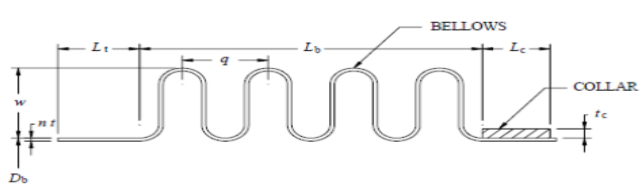

Fig. 1 Geometry of bellow

Design considerations

Normal Working Pressure = 37 bar
Normal Working Temperature = 650ºC
Angular Moment required = 17.5Nm
Maximum Permissible Deflection = ±6º
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
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, 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 (S_1)
2. Primary Collar Circumferential Membrane stress due to pressure (S_1^1)
3. Circumferential Membrane stress is also induced in the convolutions (S_2)
4. Bellows Meridional Membrane stress due to pressure (S_3)
5. Bellows Meridional Bending stress due to pressure (S_4)
6. Bellows Meridional Membrane stress due to deflection(S_5)
7. Bellows Meridional Bending stress due to deflection (S_6)

For Inconel 625 Material

I. Bellow tangent circumferential membrane stress due to pressure (S_1)

$$S_1 = \left[\frac{P \times \left(D_h + n \times t\right)^2 \times L_t \times E_h \times k}{2 \times \left\{ n \times t \times E_h \times L_t \times \left(D_h + n \times t\right) + t_c \times k \times E_c \times L_c \times D_c \right\}} \right]$$

$E_b = E_c$ = 16700 kgf/mm^2
S_1= 3.89kgf/mm^2
S_u = Ultimate tensile strength of Inconel 625 at design temperature (650ºC) = 760Mpa or76 kgf/mm^2
S_{ab} = allowable material stress of Inconel 625 at design temperature = 76/2.5 =30 kgf/mm^2
C_w = Factor accounting for Welding joint efficiency = 0.7
Effective $S_{ab} = C_w \times S_{ab}$ = 21 kgf/mm^2
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 (S_1')

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

$$S_1' = \left[\frac{P \times D_c^{\,2} \times L_t \times E_c \times k}{2 \times \left\{ n \times t \times E_b \times L_t \times \left(D_b + n \times t\right) + t_c \times k \times L_c \times D_c \right\}} \right]$$

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

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

$$S_2 = \frac{P \times D_m}{2 \times n \times t_p}\left[\frac{1}{0.571 + 2 \times {}^{w}\!/_{q}}\right]$$

(1) $S_2 = 3.4\ kgf/mm^2$

$S_2 < S_{ab}$(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 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

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

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)

$S_3 + S_4$ = 1.52 + 31.29 = 32.81 kgf/mm^2

$C_m * S_{ab}$ =3*21=63 kgf/mm^2

$S_3 + S_4 < C_m * S_{ab}$

Hence design is safe.

VI. Meridional membrane stress induced in the bellow convolution due to deflection(S_5)

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/mm^2 (at room temperature)

C_f is a shape factor = 1.38

$S_5 = 0.351 kg/mm^2$

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

(3) Where

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.

i.e.

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

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

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

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

L_b / D_b = 43/42 = 1.02

$$C_z = \text{Transition point factor} = \sqrt{\frac{4.72 \times f_{iu} \times q^2}{S_y \times D_b \times A_c}}$$

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 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 kg/mm per convolution

D_b = 42 mm

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

Substituting the values, C_z =0.776

Since $L_b / D_b \geq C_z$

P_{sc} should be evaluated for elastic region

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

C_θ =column instability pressure reduction factor based on initial angular rotation

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

γ = 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_θ= 0.5713

P_{sc} = 0.236 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

$$K_2 = \frac{D_m}{2 \times n \times t_p}\left[\frac{1}{0.571 + 2\,{}^{w}\!/_{q}}\right] = 9.189$$

α = Inplane instability stress interaction factor

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

Inplane instability stress ratio= $\delta = \dfrac{K_4}{3 \times K_2}$

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

α =38.13

For Inconel 625

S_y= 49 kgf/mm^2
P_{si} = 0.44 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 = 1.86×10^6
Total Stress S_t = 0.7 (S_3 + S_4) + (S_5 + S_6)
S_3 = 1.52 kgf/mm^2, S_4 = 31.29 kgf/mm^2

For Inconel 625

S_5 = 0.351 kgf/mm^2 , S_6 = 29.84 kgf/mm^2
S_t = 53.158 kgf/mm^2=77099.161 psi
N_c (at design temperature) = 3.02×10^6 cycles

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.

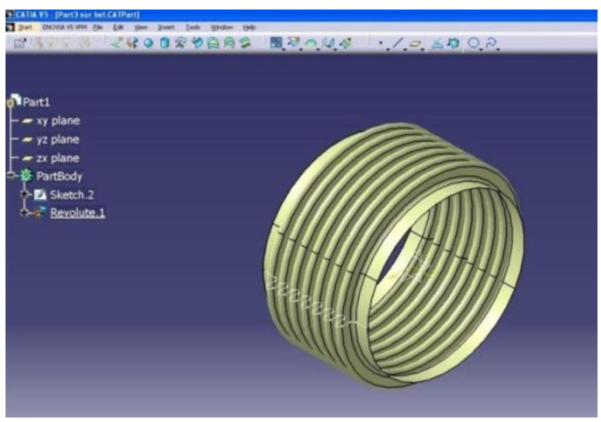

Fig. 2 The surface model

Fig. 3 The meshed model

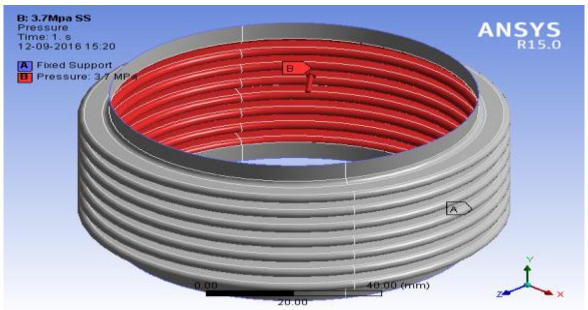

Fig. 4 The loading conditions

2 RESULTS & DICUSSIONS

4.1 Theoretically calculated Values

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 different pressures in Inconel 625

At Pressure	20 Bar	30 Bar	37 Bar	40 Bar
S_1(kgf/mm^2)	2.10	3.152	3.887	4.20
S_1'(kgf/mm^2)	2.30	3.452	4.25	4.602
S_2(kgf/mm^2)	1.84	2.757	3.4	3.676
S_3(kgf/mm^2)	0.822	1.23	1.52	1.645
S_4(kgf/mm^2)	16.91	25.36	31.29	33.81
S_5(kgf/mm^2)	0.351	0.351	0.351	0.351
S_6(kgf/mm^2)	29.84	29.84	29.84	29.84

Table 5.2 Stresses developed at different pressures in Inconel 718

At Pressure	20 Bar	30 Bar	37 Bar	40 Bar
S_1(kgf/mm^2)	2.10	3.152	3.89	4.202
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
S_5(kgf/mm^2)	0.337	0.337	0.337	0.337
S_6(kgf/mm^2)	28.692	28.692	28.692	28.692

4.1.1 Design Stresses induced in Bellow

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

Table 5.4 Fatigue life values (number of cycles N_c)

Pressure (Bar)	20	30	37	40
Inconel 625	1.2×10^8	8.94×10^6	3.022×10^6	2.078×10^6
Inconel 718	2.78×10^8	1.281×10^7	3.908×10^6	2.613×10^6

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

Table 5.5 Theoretical and Numerical stresses of Inconel 625 at different pressures

STRESS	SOURCE	20 Bar	30 Bar	37 Bar	40 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
S_2(kgf/mm^2)	**EJMA**	1.84	2.76	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
S_4(kgf/mm^2)	**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.

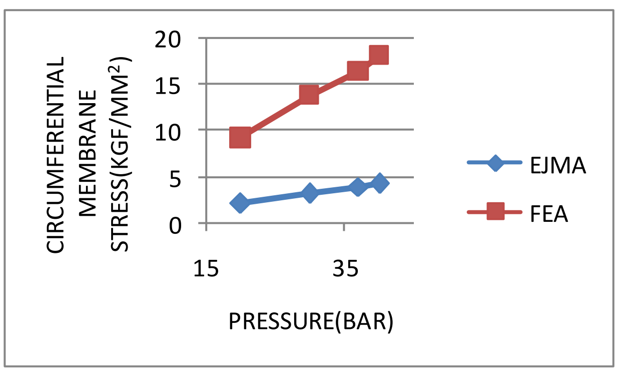

Fig. 5 Circumferential stresses in bellow tangent (S_1) for Inconel 625.$S_1 < S_{ab}$(21 kgf/mm^2)

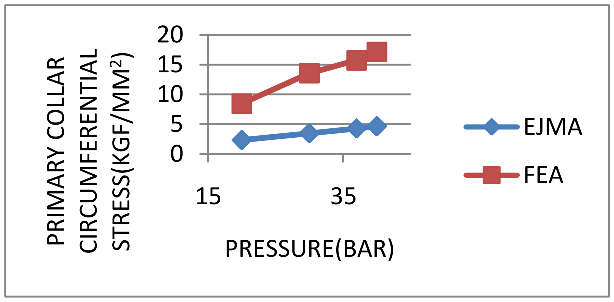

Fig. 6 Primary collar circumferential stress (S_1^1) for Inconel 625.$S_1^1 < S_{ab}$ (21 kgf/mm^2)

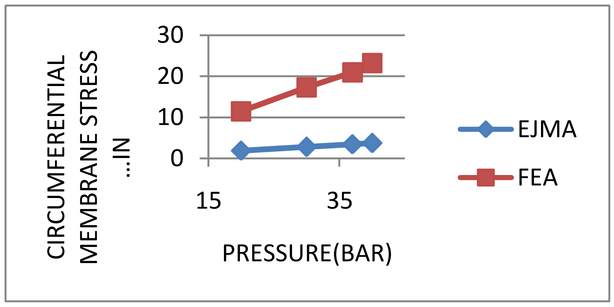

Fig. 7 Circumferential membrane stress induced in convolution (S_2) for Inconel 625 $S_2 < S_{ab}$ (21 kgf/mm^2)

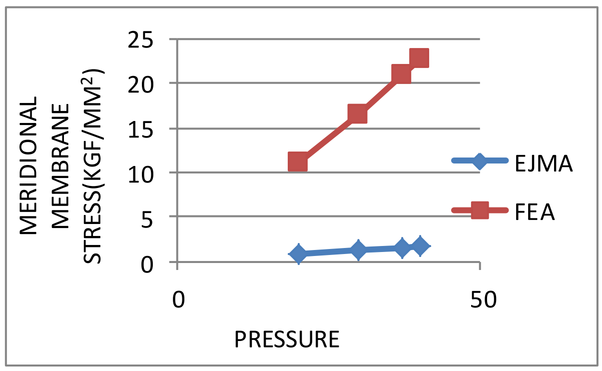

Fig. 8 Meridional membrane stress (S_3) for Inconel 625

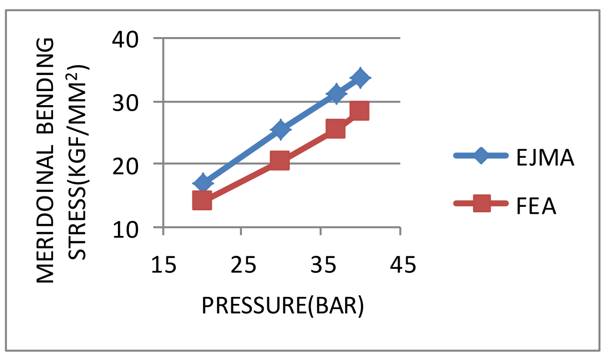

Fig. 9 Meridional bending stress (S_4) for Inconel 625 (S_3+S_4) <C_m×S_{ab}(63 kgf/mm^2)

The allowable stress value (S_{ab}) for circumferential stresses S_1, S_1^1, S_2 is 21 kgf/mm^2. Whereas for meridional stresses (S_3+S_4) <$C_m \times S_{ab}$ i.e.63 kgf/mm^2 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.

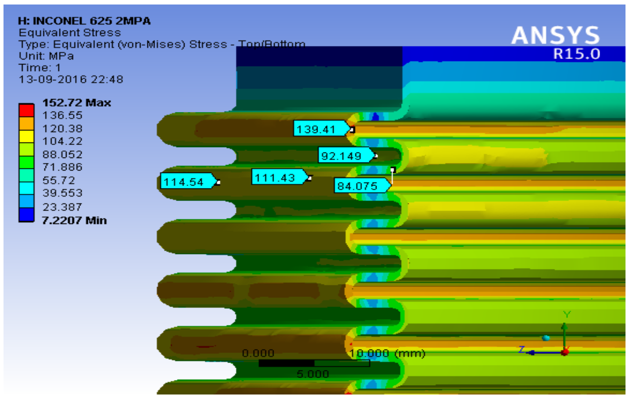

Fig. 10 Stress distribution at pressure 20 Bar

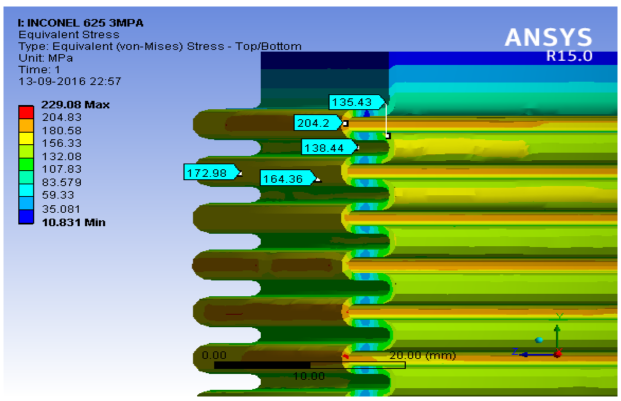

Fig. 11 Stress distribution at pressure 30 Bar

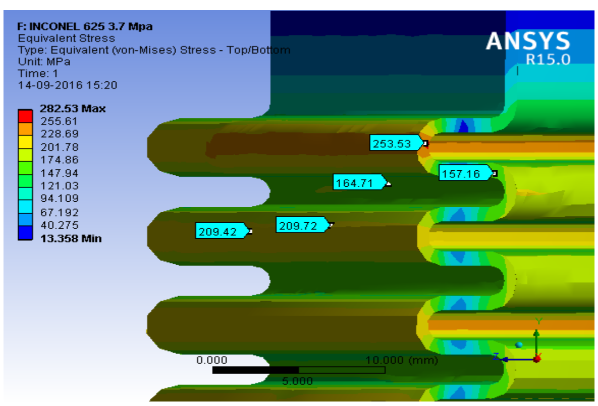

Fig. 12 Stress distribution at pressure 37 Bar

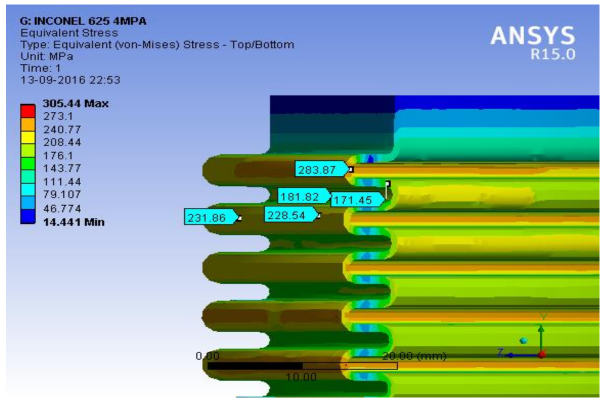

Fig. 13 Stress distribution at pressure 40 Bar

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.

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

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

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.

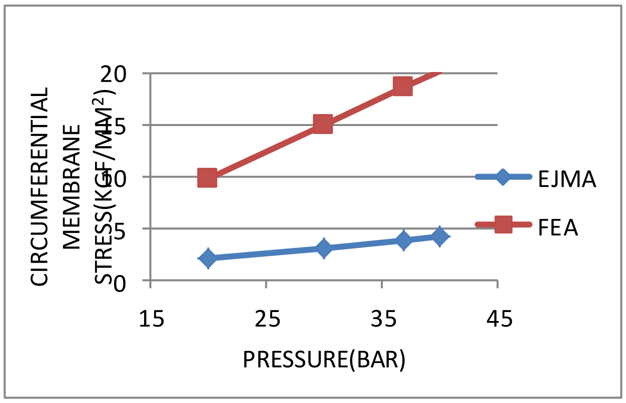

Fig. 14 Circumferential stresses in bellow tangent (S_1)for Inconel 718S_1 <S_{ab}(32.34 kgf/mm^2)

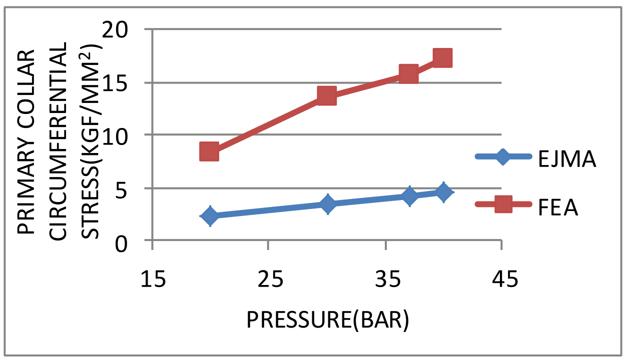

Fig 15 Primary collar circumferential stress (S_1^1) for Inconel 718S_1^1< S_{ab} (32.34 kgf/mm^2)

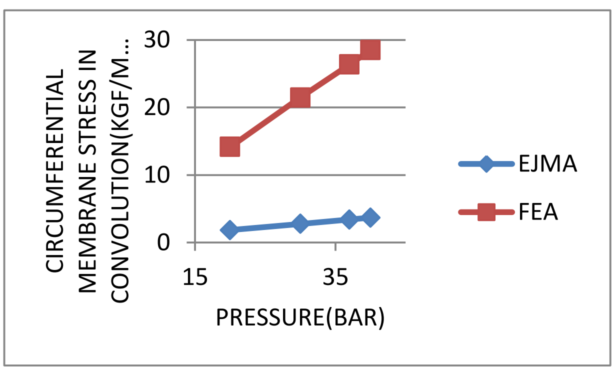

Fig. 16 Circumferential membrane stress induced in convolution (S_2) for Inconel 718 S_2 < S_{ab} (32.34 kgf/mm^2)

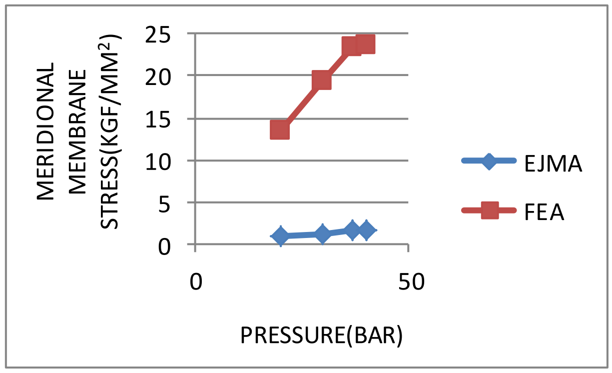

Fig. 17 Meridional membrane stress (S_3) for Inconel 718

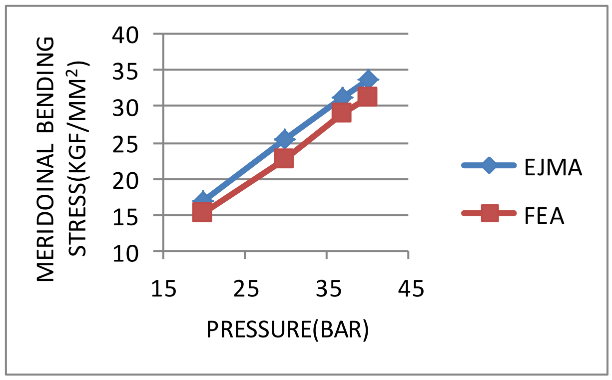

Fig. 18 Meridional bending stress (S_4) for Inconel 718(S_3+S_4) <C_m×S_{ab}(97.02 kgf/mm^2)

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.

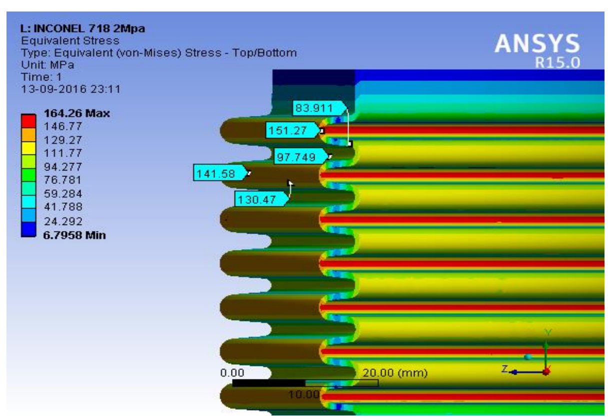

Fig. 19 Stress distribution at Pressure 20 Bar

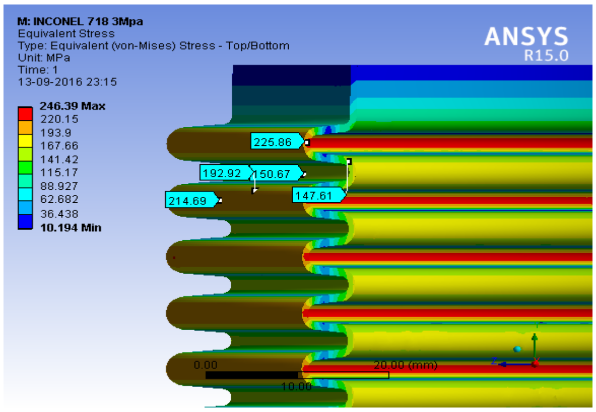

Fig. 20 Stress distribution at Pressure 30 Bar

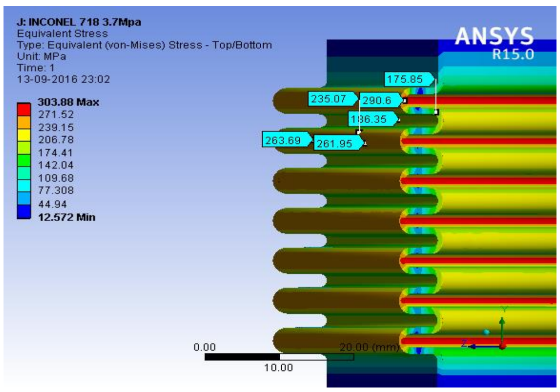

Fig. 21 Stress distribution at Pressure 37 Bar

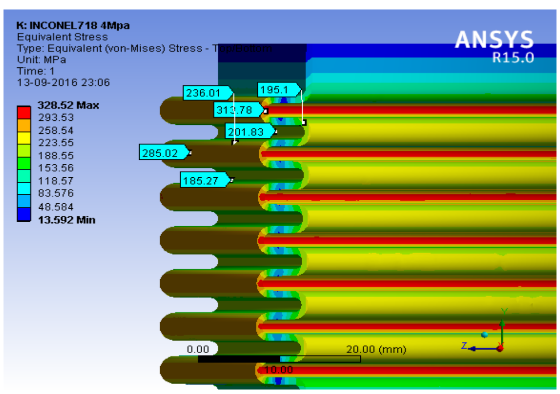

Fig. 22 Stress distribution at Pressure 40Bar

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.

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