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Analysis of Expansion Joint - A Case Study

Prof. P R Subramaniam[1], G. Rakshan Kumar[2]
[1]Associate Professor, Department of Mechanical Engineering, LENDI Institute of Engineering &Technology, Vizianagaram, India
[2]Department of Mechanical Engineering, LENDI Institute of Engineering &Technology, Vizianagaram, India

Abstract*: Automotive, Aerospace, Pipeline industries widely use Bellows. Different types of bellows are used in these industries. The bellows are used for contraction or expansion applications. Repeated variable pressure loading and displacement on Metallic bellows joints results in bellows failure. This paper is a comprehensive modeling and analysis of an axial type exhaust metallic bellow due to varying pressure load and circumferential and radial displacement. All analysis completed using ANSYS software considering variable pressure load and cylindrical displacement as a boundary condition and perused the consequences. Stress distribution in the conditions of Case (i) variable pressure load and Case (ii) displacement are obtained.
Keywords: ANSYS, FE Bellows, Finite Element Analysis, Bellow Failures

I. INTRODUCTION

A flexible connection (Bellow) is as shown in Figure 1. Some displacement takes place because of the skewed path of the connection system and considerable def lections must be allowed. The usage of a rigid joint would result in severe system vibration, noise, and early failure owing to material strength exceeding the limit. This was my experience with off-the-shelf products: they rarely fit unique applications. Failures occurred after very short operating times (Ref. Figure 2), and the problem was not solved by substituting stronger and significantly more expensive materials.

An Externally Pressurized Expansion Joint, unlike a traditional expansion joint, takes pressure from the outside of the bellows element rather than the inside. Due to the squirm associated with the longer bellow's length, excessive axial movements cannot be contained in typical expansion joints with internally pressured bellows.

To give the flexibility needed to absorb mechanical movements, the flexible tube is made up of a thin walled shell with corrugations. To analyze the behavior of the flexible tube is challenging because of its geometric complexity.

II. LITERATURE REVIEW

Several publications address various characteristics of flexible tubes, including internal pressure and axial deflection stresses, fatigue life estimations [2] and instability.

The problems of axially symmetric deformation of flexible pipes have been discussed by Chien WZ, Ming WU. and Qian H.[4,5] and were examined using the finite difference method by Hamada M, Nakagawa K, Miyata K.[6]

A good knowledge of bellow tube research can also be obtained from the conference proceedings of the 1989 ASME Pressure Vessels and Piping Conference [3].

Correction factors relating the behavior of the bellow convolution was derived by Anderson WF. [7] This method has eventually been the premise of requirements and guidebook furnishing formulae for hand-calculation of bellow specifications

Figure-1: Expansion Bellows

Figure-2: Failed Expansion Bellow

A few formulae were covered in ASME regulations on pressure vessel design, which is widely recognized. However, "The Standards of the Expansion Joint Manufacturers" is the complete and extensively accepted textual content on Bellow design. [1]
The EJMA requirements had been compared, with finite element and experimental analyses in different papers. An evaluation of the ASME standards and the EJMA requirements given by Hanna, conclude that the two conform pretty nicely in maximum aspects.

III. GEOMETRY

The Geometry for the FEA consists of the Expansion Joint i.e. Bellow. Figure 3 displays the geometry profile for the analysis. It was modeled; meshed in ANSYS. The mesh consists of 18,802 nodes and 18836 Elements. Figure 4 refers the pressure load variation from 0.01MPa to 0.16MPa applied at the cap end as boundary condition within time span 0-1sec.
Physical dimensions and Material properties used in analysis are described in Table 1. ANSYS is used as the solver for stress analysis and results

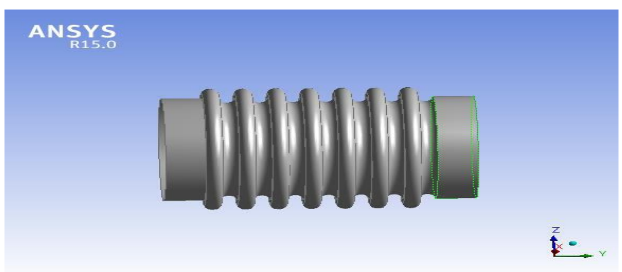

Figure-3: Model used for FEA

(a) Physical dimension				
Radius of Convolution (mm)	Pitch (mm)	Number of Convolutions	Thickness (mm)	Inner Diameter (mm)
7.75	15	7	0.5	156

b) Material properties			
Tensile Yield Stress (MPa)	Ultimate Tensile Stress (MPa)	Young's Modulus (GPa)	Poision's Ratio
240	515	193	0.29

Table-1: Physical dimensions and material properties

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IV. RESULTS AND DISCUSSIONS

A. Case I- Effect Variable Pressure Load on Cap End

Pressure variation from 0.01MPa to 0.16Mpa as shown in figure 5 is acting on the cap end resulting in maximum stress around convolution root from the end cap within 1 second. Stress distribution and deformed shape are as shown in Figure 4. Max. and Min Values of equivalent stress, directional deformation and total deformation are given in table-2.

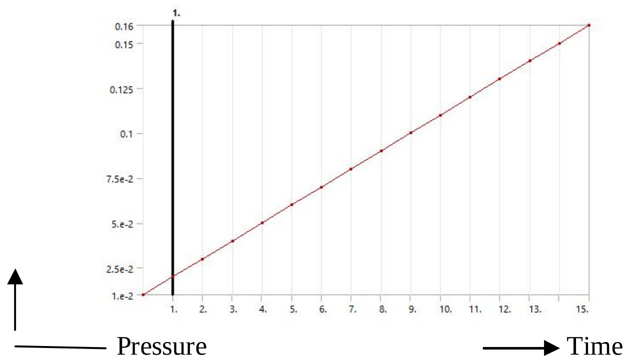

Fig- 4: Pressure load variation over time

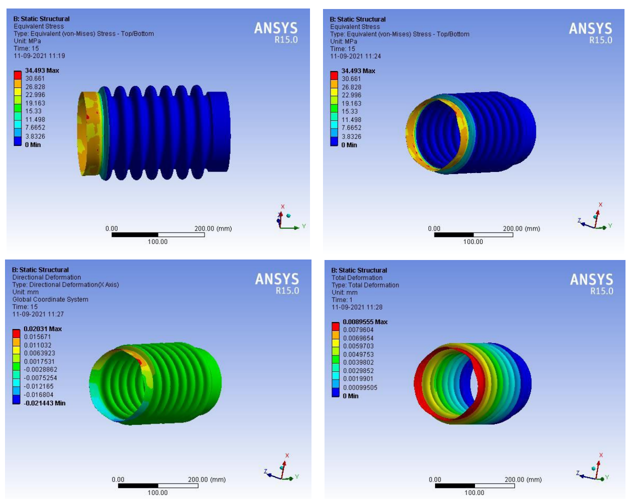

Figure-5: Stress distribution and deformed shape

	Results		
	Equivalent Stress (MPa)	*Directional Deformation* (mm)	*Total Deformation* (mm)
Minimum	0.	-2.1443e-002	0.
Maximum	34.493	2.031e-002	8.9555e-003
	Minimum Value Over Time		
Minimum	0.	2.1443e-002	0.
Maximum	0.	2.6804e-003	0.
	Maximum Value Over Time		
Minimum	4.3117	2.5388e-003	8.9555e-003
Maximum	34.493	2.031e-002	7.1644e-002

Table-2: Equivalent (Von-Mises) Stress, Directional Deformation and Total Deformation

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Volume 9 Issue X Oct 2021- Available at www.ijraset.com

Variation of equivalent stress, Directional Deformation and Total Deformation with respect to time are graphically shown in figures 6 to 9

It can be seen from the analysis that when the Bellow is undergoing expansion and contraction, with momentary pressure fluctuations also stress arises and as the pressure increases from 0.1MPa to 0.16MPa, stress and the deformation at the corners increases as shown in Figure-5 leading to its failure.

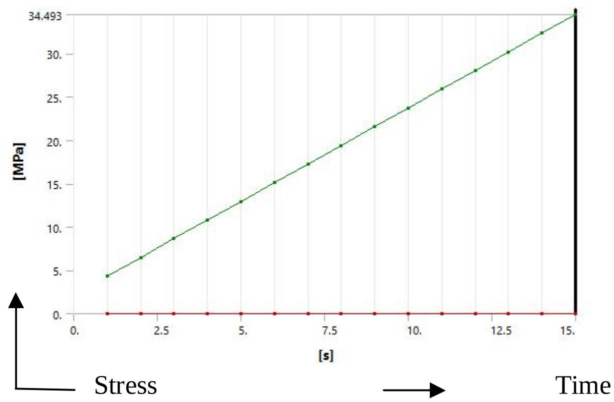

Figure-6: Equivalent Stress

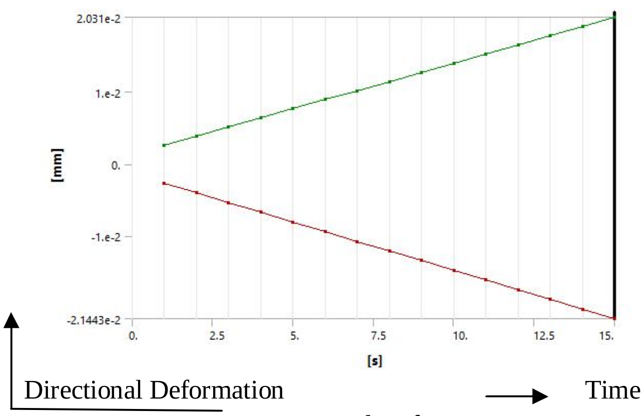

Figure-7: Directional Deformation

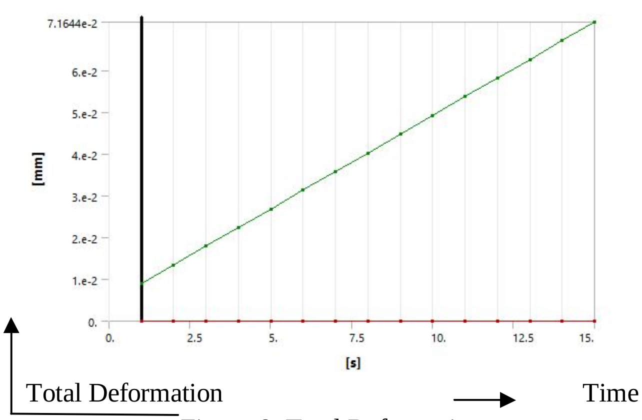

Figure-8: Total Deformation

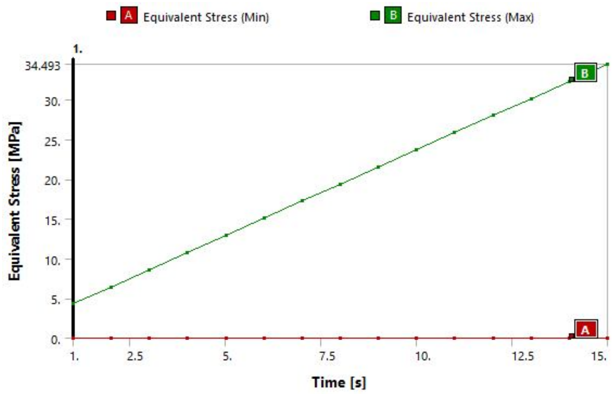

Figure- 9: Equivalent Max and Min Stress

B. Case II- Effect of Displacement Load

Figure 10 refers cylindrical displacement varying from 0.0 to 0.1 mm applied as boundary condition.

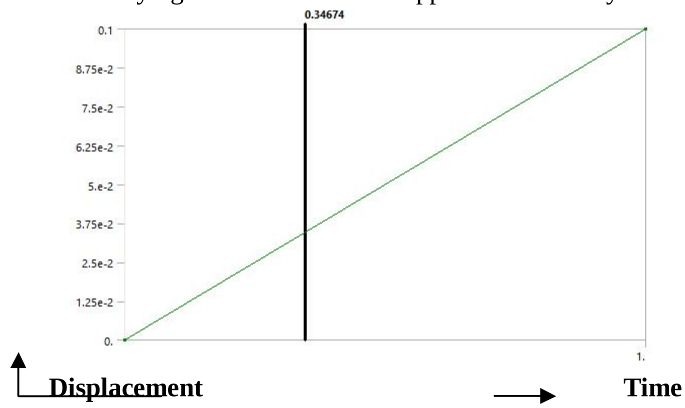

Figure 10 Applied Displacement Load

Figure 10 shows stress distribution and deformed shape. The maximum deformation occurs around convolution root near the end cap for a displacement from 0.1mm. Values of Total Deformation, Equivalent Stress Equivalent Elastic Strain are given in Table-3 below. The equivalent stress increases linearly as the cylindrical displacement increases and as it crosses 0.025mm failure takes place at 0.346 sec

The stress, deformation and elastic strain exhibits an upward trend with the increase of deflection.

	Results		
	Total Deformation (mm)	*Equivalent Stress* (MPa)	*Equivalent Elastic Strain* mm/mm
Minimum	0	0	0.
Maximum	0.18741	1337.7	8.3872e-003
	Minimum Value Over Time		
Minimum	0.	0	0.
Maximum	0.	0	0.
	Maximum Value Over Time		
Minimum	0.18741.	1337.7	8.3872e-003
Maximum	0.18741.	1337.7	8.3872e-003

Table-3 Total Deformation, Equivalent Stress, Equivalent Elastic Strain

From the analysis displacement variation during assembly with corresponding mating combined with pressure fluctuations during operation appears to be the cause of premature failure of Bellows.

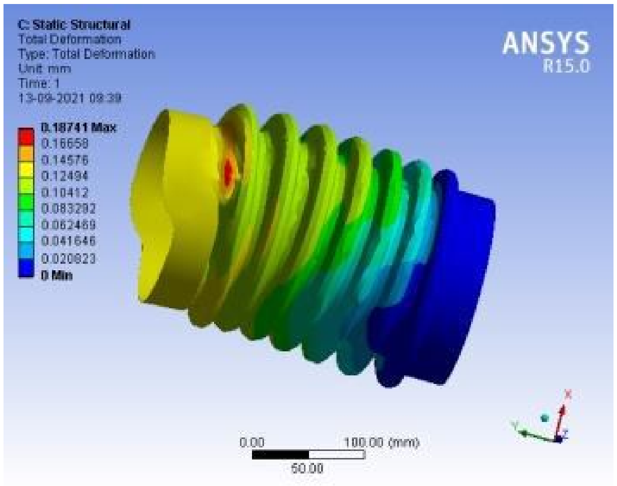

Figure 9 Total Deformation

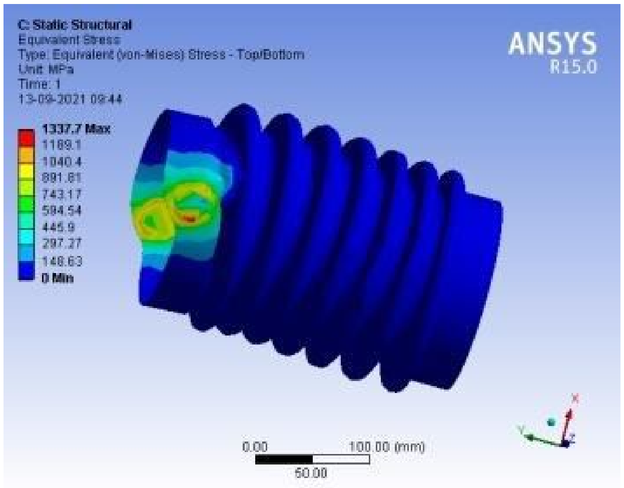

Figure 10 Equivalent Stress

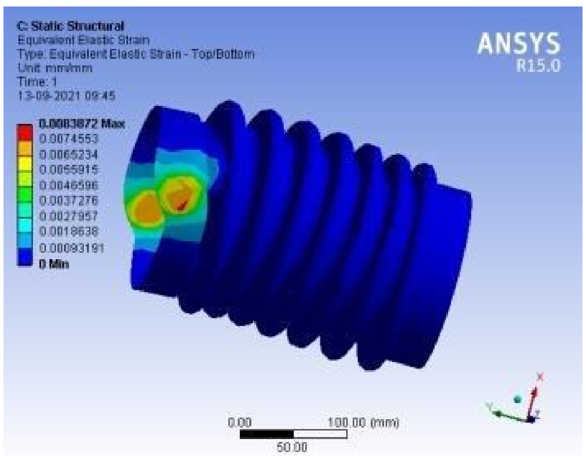

Figure 11 Equivalent Elastic Strain

V. CONCLUSION

The effects of the pressure and displacement variation on the failure of Bellows examined in this study are summarized as follows;

A. The equivalent stress increases linearly and the stress exhibits an upward trend with an increase of pressure at the bellow's cap end.
B. The stress increase is leading to failure with the increase of deflection at the Bellow's cap end.
C. Displacement above 0.09mm and Pressure above 0.11Mpa has a significant effect in contributing towards failure of the Bellow.

VI. ACKNOWLEDGEMENT

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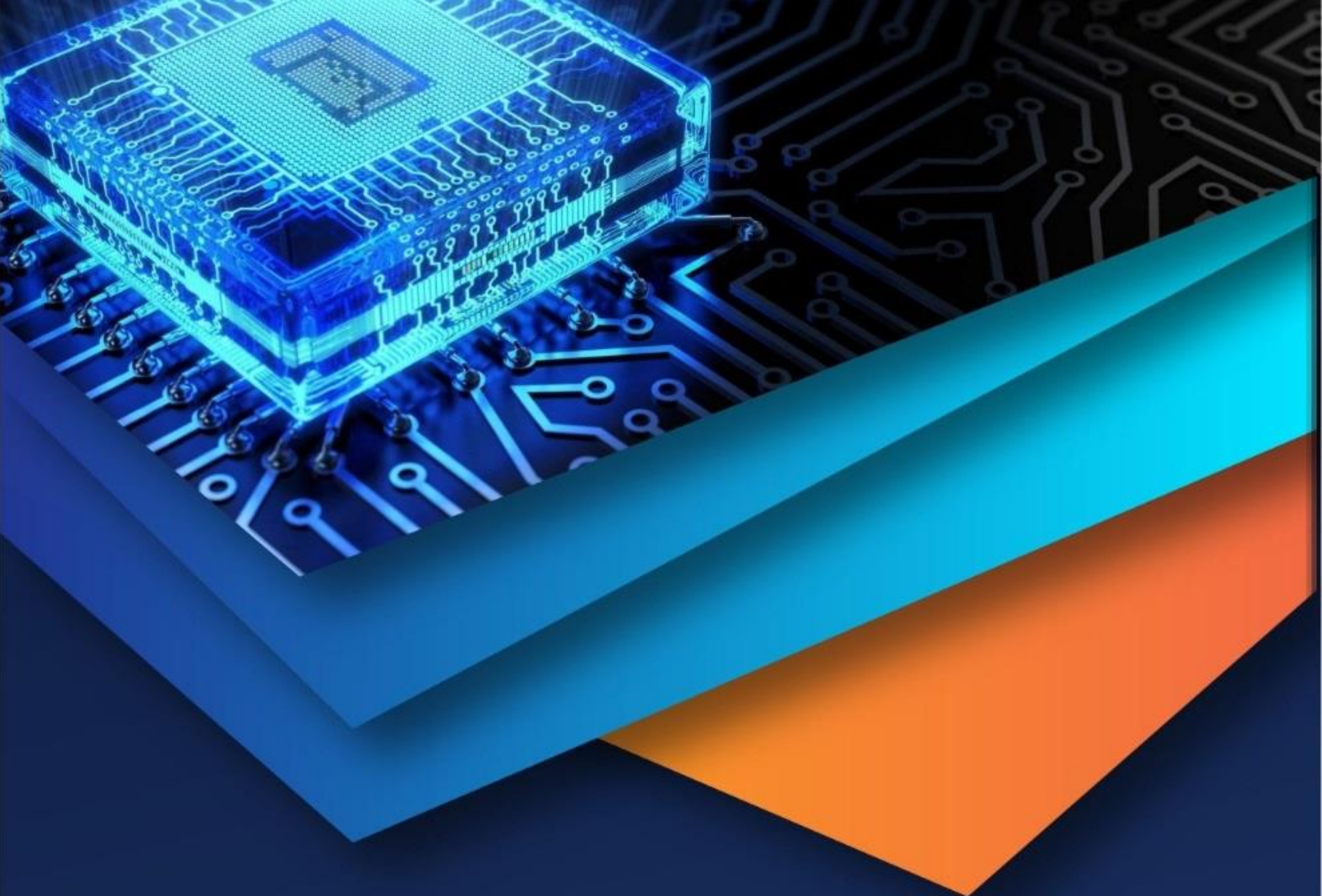

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