

# STATIC STRUCTURAL, FATIGUE AND BUCKLING ANALYSIS OF JET PIPE LINER BY INDUCING CORRUGATION

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**Abstract** - Gas turbines are the power plant used in aero engine applications in both military and civil aircrafts due to easy to start and stop, compact in nature. Gas turbine mainly consists of compressor, combustor, turbine, Afterburner and exhaust modules. Afterburner is used for thrust augmentation purpose to meet the take-off and critical measure requirements. The liner covering the after burner internally serves to shield the jet pipe from the high temperature gases. In the Afterburner, when reheat process takes place a liner is protecting the jet pipe from the hot gases emitting from Afterburner. The liner is known as jet pipe liner which is a thin shell cylindrical member whose primary failure mode is buckling

In this paper, basic conventional jet pipe line is selected for structural analysis to do Stress analysis, Fatigue analysis and buckling analysis. The conventional liner is further optimized by inducing corrugation. The CAD model of the liner was imported from AutoCAD into Hypermesh 13.0 and mapped finite element mesh was done using Hypermesh 13.0. ANSYS Workbench 14.5 is used for carrying out the structural analysis. Theoretical calculations are done to find out Von-Mises stress and fatigue value. The analysis carried out to estimate the Deformation, Von-Mises stress, Fatigue life and buckling load factor. The results are analyzed with suitable graphs.

**Key words:** Liner, Corrugation, Buckling.

## 1. INTRODUCTION

Afterburning (AB) is one of the methods of augmenting the basic thrust of the turbojet and turbofan engines, when required, without having to use a larger engine with its concurrent penalties of increased frontal area and weight. The afterburner increases the thrust by adding thermal energy to the entering gas stream in the liner. The AB liner experiences high variable differential pressure and thermal loads along its longitudinal direction. Structural design analysis of a liner involves various objectives, the important ones being related to strength, buckling, stiffness, natural frequencies through eigen values, fatigue and creep along with inherent challenges to minimise weight and production costs.

Buckling is a dominant mode of failure for an afterburner liner due to its inherent slender geometry and external

pressure loading system in addition to low stiffness due to metal high temperature. Further, the liner is required to withstand severe loads like those occurring during unpredictable engine flameouts. The flameout load conditions though happen rarely in aero-engines, however they are not uncommon during the life cycle of the engine and constitute an extreme load condition for the structural design of AB liner, particularly for buckling mode of failure.

Thin-walled circular cylindrical shells of various constructions are widely used in engineering structures. Because of their high slenderness ratio, the loss of elastic stability is of primary concern. Numerous numerical and experimental studies on the shell stability phenomenon, including critical load, buckling mode as well. These studies were mainly focused on simple shell with specific boundary conditions at free ends to provide accurate and validated analytical formulae for design purposes. In industrial sectors corrugated structures are selected because of their higher buckling strength. Also, the corrugated structures are easier to manufacture compared to stiffened welded structures. Corrugated circular cylindrical shells made of an isotropic and homogeneous material can also be considered approximately orthotropic cylindrical shell provided that the corrugation depth is not too high.

Chethan.E.Kolambe and Kapil.S.Shah [1] done Static structural and buckling mode analysis on thin shell cylinder with and without dent using FEA software to find equivalent stress and buckling. Dr.N.V.Srinivasulu et al., [2] done buckling analysis on thin cylindrical shell considering three different types of geometrical patterns. T.Subramani and Athulya Sugathan [3] done FEM analysis on thin walled shell structures to get Buckling load factor. Gilijith Jacob et al., [4] done buckling analysis on steel cylindrical shells with an elliptical cutout under combined loading.

## 2. PROBLEM DEFINITION

Comparative FEA analysis is carried out to find Deformation, Von-Mises stress, Fatigue life and Buckling load multiplier using ANSYS Workbench. Considering homogeneous material throughout shell element.

### 3. METHODOLOGY

Steps for FEA Analysis:

1. Importing CAD model using AutoCAD
2. Optimizing the model by inducing Corrugation
3. Theoretical calculations to find Von-Mises stress and Fatigue life using correlations.
4. Selecting Material Properties such as material, modulus of elasticity, poisson's ratio, yield stress etc.
5. Meshing both the models using Hypermesh 13.0
6. Applying Boundary Conditions
7. Calculating Total Deformation, Von-Mises stress, Fatigue life and Buckling load multiplier

### 4. GEOMETRY OF MODEL

#### 4.1 Geometry of the conventional Liner

For the present work on the liner, 2D sectional model of liner was imported from AutoCAD. The 2D model was converted into 3D circular model. The length of the liner is 950 mm, Diameter is 880 mm and Thickness is 1.2 mm. The 3D model of conventional liner is as shown in Fig.1

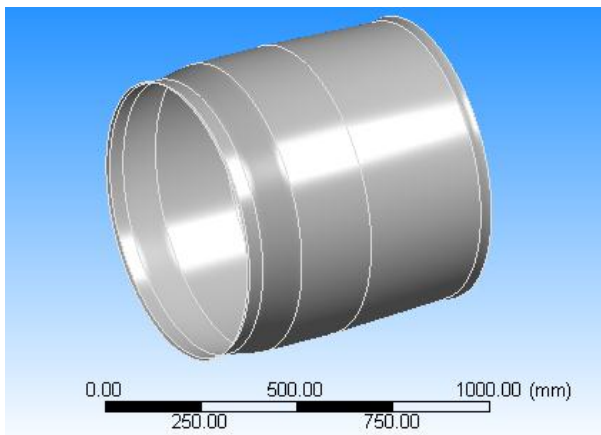


Fig 1 : 3D model of conventional liner

#### 4.2 Optimization of conventional liner by inducing corrugation.

A corrugated Liner has been prepared by considering the length and diameter of the afterburner as decided by the aerodynamic and performance requirements. Suitable parameters are added to get corrugated model. The specifications of corrugated liner model are as shown in the Table I. The major parameters of corrugation viz. amplitude, pitch, inclinations and corner fillet radius are shown in Fig.2. Fig.3 shows the 3D model of corrugated liner

### Specifications of Corrugated liner:

TABLE I: Specifications of corrugated liner

Parameter	Values in mm
Length	950
Diameter	880
Thickness	1.2
P	180
A	12
R1	20
R2	30
$\alpha$	10 degree
$\beta$	20 degree

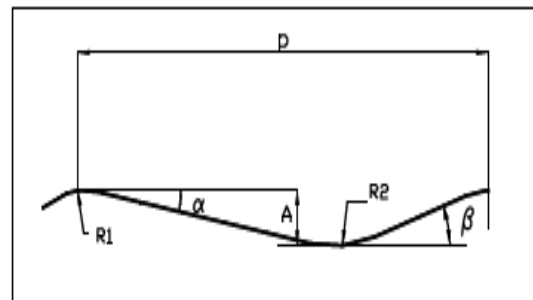


Fig.2: Major parameters of the corrugation

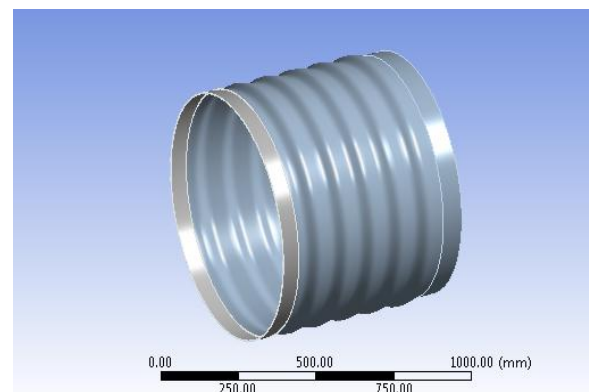


Fig.3: 3D model of corrugated liner

## 5. THEORETICAL CALCULATIONS

### 5.1 Von-Mises stress calculations

Von-Mises stress theory is also called as Maximum distortion strain energy theory. It states that, the given structural material is safe as long as the maximum value of the distortion energy per unit volume in that material remains smaller than the distortion energy per unit volume required to cause yield in a tensile test specimen of the same material.

Since Liner belongs to thin cylinders, the Von-Mises stress is calculated as shown below,

$$\sigma_H = \frac{Pd}{2t} = \frac{0.44 \times 880}{2 \times 1.2} = 161.33 \text{ MPa}$$

$$\sigma_L = \frac{Pd}{4t} = \frac{0.44 \times 880}{4 \times 1.2} = 80.66 \text{ MPa}$$

where,

P = Load (Pressure) applied on liner in MPa

d = diameter of liner in mm

t = thickness of liner in mm

Here,  $\sigma_H = \sigma_1$ ,  $\sigma_L = \sigma_3$ ,  $\sigma_2 = 0$

$$\sigma_V = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}} = \sqrt{\frac{(161.33 - 0)^2 + (0 - 80.66)^2 + (80.66 - 161.33)^2}{2}} = 140 \text{ MPa}$$

### 5.2 Fatigue calculation

#### Soderberg approach:

Soderberg approach is used to determine fatigue life of ductile materials considering yield strength.

$$\frac{\sigma_a}{\sigma_{Nf}} + \frac{\sigma_m}{\sigma_y} = 1$$

Where,

$\sigma_y$  is the yield strength of the material,

$\sigma_m$  is the mean stress,

$\sigma_{Nf}$  is the fatigue life strength and

$\sigma_a$  is the alternating stress.

The endurance limit  $\sigma_e$  is determined using suitable formula. The yield strength of Inconel 188 material is 268 MPa.

The S-N behavior is typically fit using power law of the

form,

$$\sigma = a N^b$$

Where, N is the number of cycles to failure at stress level.

a and b is called the fatigue exponent.

i) For  $N = 10^3$  cycles,  $\sigma = \sigma_e$

$$\sigma_e = 0.9 \sigma_y = 0.9 \times 268 = 241 \text{ MPa}$$

$$241 = a N^{3b} \longrightarrow (1)$$

ii) For  $N = 10^6$  cycles,  $\sigma = \sigma_e$

$$\sigma_e = (K_a \cdot K_b \cdot K_c \cdot K_d) \sigma_e'$$

a) Surface factor

The surface modification factor  $K_a$  depends on the yield strength of the material  $\sigma_y$  and on the part surface finish.

$$K_a = a \sigma_y^b$$

The factor 'a' and the exponent 'b' depends upon the quality of the surface finish

a = {57.7 for hot rolled surface finish.

b = {-0.718 for hot rolled surface finish.

$$\text{Then, } K_a = 57.7 \times 268^{-0.718} = 1.0417$$

b) Size factor

$$K_b = \{0.9 \text{ for bending and torsion.}$$

c) Reliability factor

$$K_c = \{0.753 \text{ for 99. \% reliability.}$$

d) Load factor

$$K_d = \{1 \text{ for bending}$$

The corrected endurance limit  $\sigma_e$  can now be calculated from given equation.

$$\sigma_e = (K_a \cdot K_b \cdot K_c \cdot K_d) \sigma_e'$$

$$\sigma_e = (1.0417 \times 0.9 \times 0.753 \times 1) 241$$

$$= 94.59 \approx 95 \text{ MPa}$$

$$95 = a N^{6b} \longrightarrow (2)$$

By solving equation (1) and (2),

a = 611.46, b = -0.1349

The Soderberg line equation is,

$$\frac{\sigma_a}{\sigma_{Nf}} + \frac{\sigma_m}{\sigma_y} = 1 \longrightarrow (3)$$

The maximum and minimum stress acting on liner is,

$$\sigma_{max} = 140 \text{ MPa}, \sigma_{min} = 0 \text{ MPa}$$

The alternating stress( $\sigma_a$ ) and mean stress( $\sigma_m$ ) are given by

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{140 - 0}{2} = 70 \text{ MPa}$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = \frac{140 + 0}{2} = 70 \text{ MPa}$$

On substituting the value of  $\sigma_a$  and  $\sigma_m$  in equation (3), we get

$$\frac{70}{\sigma_{Nf}} + \frac{70}{268} = 1$$

$$\sigma_{Nf} = 95 \text{ MPa}$$

$$\sigma_{Nf} = a N_f^b$$

On substituting,  $\sigma_{Nf}$ , a and b values,

$$N_f = 9.902 \times 10^5 \text{ cycles}$$

## 6. MATERIAL PROPERTIES

TABLE II: Properties of material

Material type	Inconel 188
Poisson's ratio( $\mu$ )	0.322
Young's modulus(E)	172e3 MPa
Density( $\rho$ )	8980 kg/m <sup>3</sup>
Yield strength( $\sigma_y$ )	268 MPa
Factor of Safety	1.5
Allowable yield stress	178.6 MPa

The material used in the Liner for doing analysis is Inconel 188. Assuming 800 degree celcius, the material properties are shown in Table II

## 7. MESHING

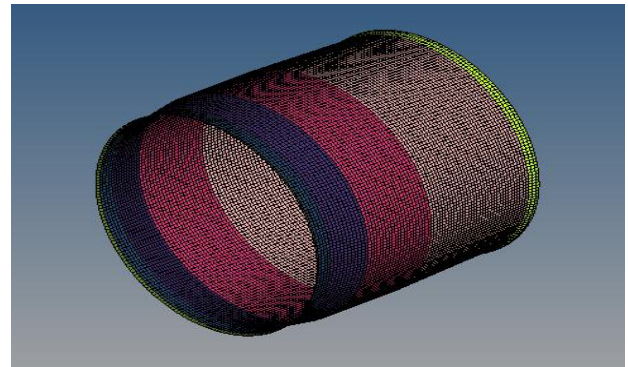


Fig.4: Meshed model of conventional liner

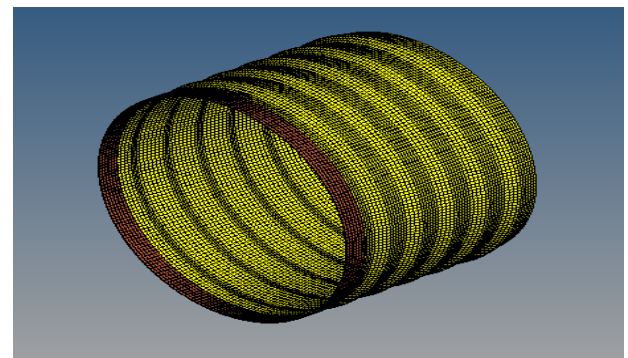


Fig.5: Meshed model of corrugated liner

Fig.4 and Fig.5 shows the meshed model of conventional and corrugated liner respectively. The SHELL181 elements are used for representation of the thin shell structure. The conventional liner model consists of 17550 nodes and 17316 elements and the corrugated liner model consists of 23280 nodes and 221840 elements.

## 8. BOUNDARY CONDITIONS

Boundary conditions play an important role in the finite element analysis. Here cylindrical co-ordinate system is used at free end in both models. Tangential[Y] and Axial[Z] displacement is fixed, and radial[X] displacement is kept free. A Pressure of 0.44 MPa is applied on the outer surface of the body. Fig. 6 and Fig. 7 shows the Boundary conditions applied to the conventional and corrugated liner respectively.

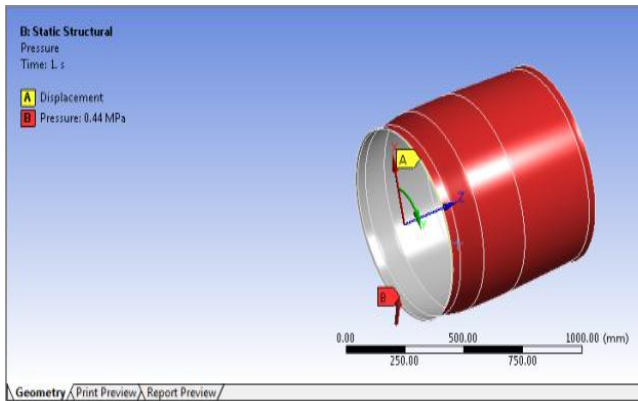


Fig.6: Boundary conditions on conventional liner

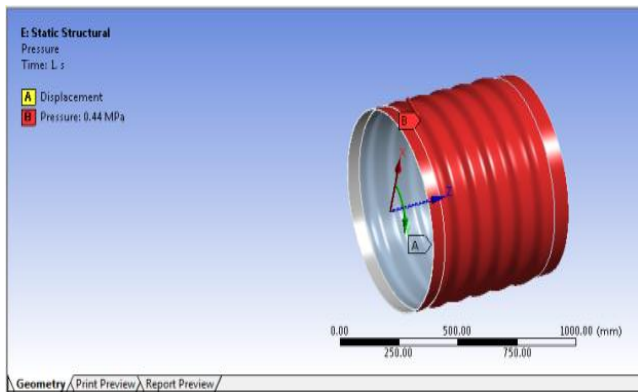


Fig.7: Boundary conditions on corrugated liner

## 9. RESULTS AND DISCUSSION

### 9.1 Static Structural Analysis

#### [a] Conventional liner

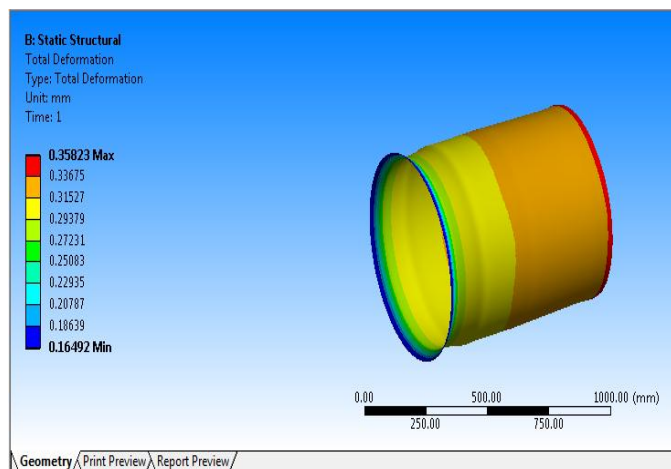


Fig.8: Total deformation of conventional liner

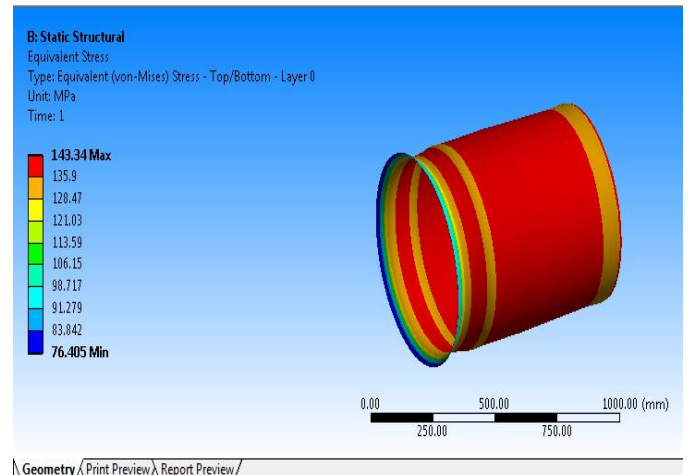


Fig.9: Von-Mises stress of conventional liner

#### [b] Corrugated liner

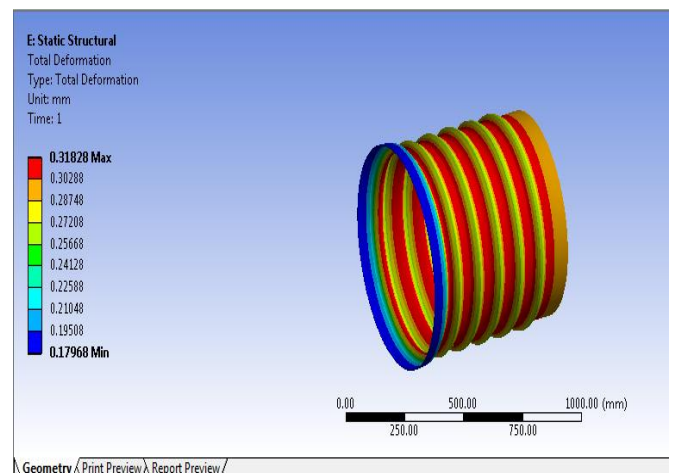


Fig. 10: Total deformation of corrugated liner

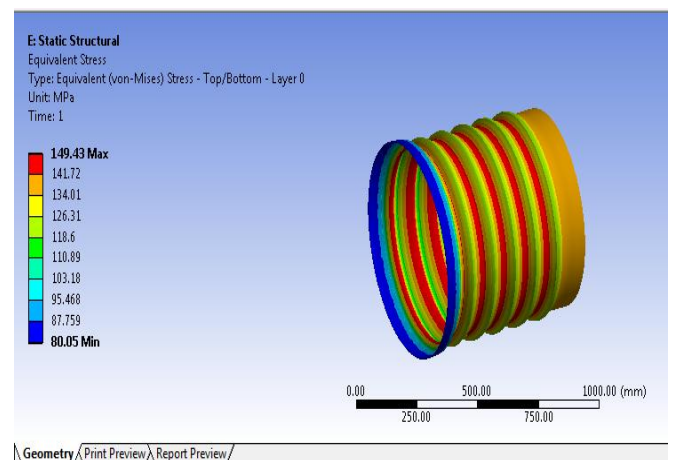


Fig. 11: Von-Mises stress of corrugated liner

Fig. 8, Fig. 9 shows the Deformation and Von-Mises stress results of Conventional liner and Fig. 10, Fig. 11 shows the Deformation and Von-Mises stress results of Corrugated liner respectively. For the corrugated liner, deformation obtained is 0.3182 mm which is less than that of the deformation of conventional liner value 0.3582 mm. Hence the corrugated liner design can be accepted.

Ansys Von-Mises stresses obtained is 143.34 MPa and 149.43 MPa which is nearer to the theoretical Von-Mises stress i.e 140 MPa. By taking FOS of 1.5, the critical yield stress of material obtained is 178.6 MPa. The Von-Mises stress obtained for conventional and corrugated liner is 143.34 MPa and 149.43 MPa respectively, which is less than the critical yield stress. Hence the design is safe.

### 9.2 Fatigue Analysis

#### [a] Conventional liner

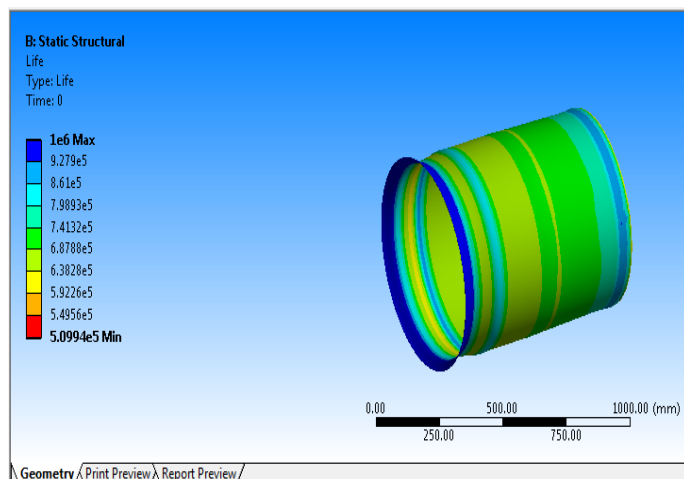


Fig. 12 : Fatigue Life of conventional liner

#### [b] Corrugated liner

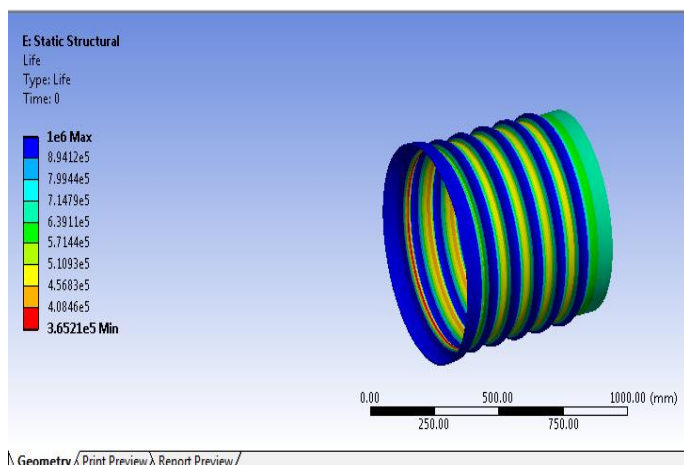


Fig. 13 : Fatigue Life of corrugated liner

Fig. 12, Fig. 13 shows the Fatigue life results of Conventional and corrugated liner respectively. For conventional liner, the life obtained is  $5.099 \times 10^5$  cycles and for corrugated liner, the life obtained is  $3.652 \times 10^5$  cycles. Component under the given loading condition exhibits the maximum fatigue life of  $1 \times 10^6$  cycle and the theoretical value obtained is  $9.902 \times 10^5$  cycle. Here the theoretical fatigue life and ansys fatigue life results corresponds to less than  $10^6$  cycles of life. Hence, the design is safe, it has a infinite life and there is no yielding.

### 9.3 Buckling Analysis

#### [a] Conventional liner

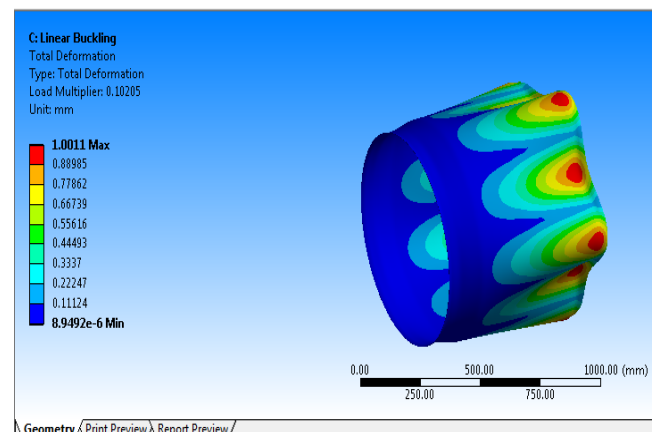


Fig. 14: Buckling mode shape of conventional liner

#### [b] Corrugated liner

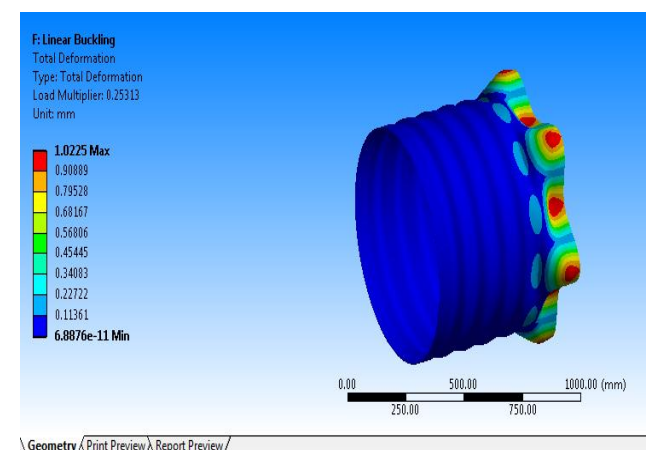


Fig. 15: Buckling mode shape of corrugated liner

Fig. 14, Fig. 15 shows buckling mode shapes of conventional and corrugated liner respectively. From the results, the buckling load multiplier for conventional and corrugated liner obtained is 0.102 and 0.253 respectively. Here buckling load multiplier of corrugated liner is more than that of the conventional liner. Hence corrugated liner is more stiffer than conventional liner in Buckling point of view. Table III provides the consolidation of overall results.

**Table III:** Consolidation of overall results

Liner type	Deformation in mm	Von-Mises stress in MPa	Life in cycles	Buckling load multiplier
Conventional liner	0.3582	143.34	5.099e5	0.102
Corrugated liner	0.3182	149.43	3.652e5	0.253

### 10. CONCLUSION

Some of the following conclusion may be drawn from the FEM analysis.

1. Literature survey was conducted and the problem is defined after consideration of previous research work.
2. A brief description of the design and analysis of the jet pipe liner is presented, FEA procedure involved in the analysis is given.
3. Conventional liner is considered for analysis. The liner is further optimized by inducing corrugation with suitable parameters.
4. Static structural analysis shows that, the Deformation of corrugation liner is less compared to conventional liner. The Von-Mises stress of both liners are below allowable yield stress and also it is nearer to theoretical results.
5. Fatigue analysis has been done on both models considering Stress-Life approach and using Soderberg line. The obtained ansys fatigue life results and theoretical fatigue life corresponds to less than  $10^6$  cycle of life and hence design is safe, it has infinite life and there is no yielding.
6. Buckling analysis has been done on both liners. Results show that, corrugated liner is more stiffer than conventional liner in Buckling point of view.

### REFERENCES

- [1] Chetan.E.Kolambe and Kapil.S.Shah, "A Study of Buckling Mode Analysis of Thin Shell Cylinder Using FEA Software", IJESC, Volume 6 Issue No. 3, 2016
- [2] Dr. N.V.Srinivasulu, B.Suryanarayana, N.Chandra Sekhar, " Experimental Buckling Analysis of Thin Cylindrical Aluminium Shells", International journal of Engineering and technology, Vol-3.

[3] T. Subramani and Athulya Sugathan, "Finite Element Analysis of Thin Walled-Shell Structures by ANSYS and LS-DYNA", International Journal of Modern Engineering Research (IJMER), Vol.2, Issue.4, July-Aug. 2012

[4] Giljith Jacob, Divya K.K, Preetha Prabhakaran, "Buckling Analysis of Cylindrical Shells Subjected to Axial Compression", International Research Journal of Engineering and Technology (IRJET), e-ISSN: 2395 - 0056, Volume: 04, Issue: 04, April -2017

### BIOGRAPHIES



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