BUCKLING ANALYSIS OF CYLINDRICAL SHELL STRUCTURES USING FINITE ELEMENT ANALYSIS

Neelesh Ashok, Ben Austin B Alapatt

Abstract— **Thin shell structures find wide applications in many branches of engineering. Examples include aircraft, space shuttles, launch vehicle tankages, pressure vessels, pipelines etc. The design involves the determination of stresses and deformations produced by imposed loads on the structure. The structure is subjected to buckling, a physically observed failure mode that is associated closely to load carrying capacity, when it is in service. The buckling strength of thin walled cylinders are to be determined and depending upon the material and geometry, the compressive stresses may lead to any of the failure modes like elastic buckling, elastic plastic buckling and plastic collapse. The purpose of this work is to review the entire field of buckling of cylindrical shell structures to find the allowable stress and deformation under given load which is primarily accomplished by referring to reviews, scholarly articles, journals, books and study the load carrying capacities of shells under different loading mechanisms like torsion, axial compression and uniform lateral external pressur.**

Index Terms— **Buckling of cylindrical shells, elastic buckling, abaqus, finite element analysis.**

I. INTRODUCTION

Buckling refers to the loss of stability of a component and is usually independent of material strength. This loss of stability usually occurs within the elastic range of the material.

Buckling phenomena occurs when most of the strain energy which is stored as membrane energy can be converted to bending energy requiring large deformation resulting in catastrophic failure.

Cylindrical shells have been extensively used in all types of structures. They are subjected to various loadings both static and dynamic in nature. Amongst the most common types of loading are axial compression, torsion, lateral pressure and combination of both axial and lateral pressure which is seen commonly in shell structures that cause deformation of unacceptably large amplitude and could lead to loss of stability and collapse of the whole structure.

The paper is organized into following sections. Section 1 covers introduction followed by literature review and introduction to abaqus finite element software in section 2 and Section 3 Section 4 will cover the objective of the work and entire analysis with result validations is mentioned in section 5. Section 6 which covers summary and conclusion and references will cover in section 7.

II. LITERATURE REVIEW

The researchers have amassed knowledge about elastic buckling of cylindrical shells and gave results based on experiments and numerical methods. Some of them are Jan Błachut (2008) , gave experimental and numerical study of cylindrical shells . K. Athiannan ,R. Palaninathan (2004), gave an investigation on buckling of cylindrical shells under axial compression. K. Magnucki ,M. Mackiewicz (2006), defined effects of lengths, sector angle and different boundary conditions on buckling load and behavior of cylindrical panels have been done by experimental and numerical methods. Jin Guang Ten (2009), presented a review of recent research advances and trends in the area of thin shell buckling. He gave emphasis on imperfections in real structures and their influence, buckling of shells under local/non-uniform loads and localized compressive stresses and the use of computer buckling analysis in the stability design of complex thin shell structures.

From the literature review, it was observed buckling analysis has to be performed using finite element methods and should be validated numerically. Therefore, this paper attempts to explore the different aspects of buckling of cylindrical shells under axial compression, torsional loads, uniform external lateral pressure, combined loads of both axial and external pressure. The numerical solutions are compared for these loading conditions.

III. INTRODUCTION ON ABAQUS FE PACKAGE

ABAQUS is a general-purpose simulation tool, and can solve a wide range of engineering problems, including structural analysis and heat transfer problems.

The Abaqus suite has several parts, but only three were used in the current study: Abaqus/CAE, Abaqus/Standard, and Abaqus/Explicit. Abaqus/CAE is a graphical user interface (GUI) for pre-processing and post-processing a finite element analysis, Abaqus/Standard and Abaqus/Explicit are two different analysis solvers. These three parts are briefly described below

Abaqus/CAE is a graphical tool which allows an analyst to create and prepare a model for analysis and then view the analysis results. CAE is an abbreviation for Complete Abaqus Environment. Abaqus/CAE provides an analyst with tools to create a geometric model of the structure to be analyzed, give the model material properties and a mesh, and to setup an analysis in a way which allows the analyst to make small changes quickly without much hassle. The output database can be viewed in Abaqus/CAE, where pictures and graphs can be created.

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Abaqus/Standard and Abaqus/Explicit are the two solvers in the Abaqus suite. Both solvers can handle nonlinearities in the materials, loads, and geometry. This is one of the primary strengths of Abaqus. However, each has its own uses, and particular features, so it is up to the analyst to decide which is best suited to his particular problem. Abaqus/Explicit determines the solution without iterating by explicitly advancing the kinematic state from the previous increment. Even though a given analysis may require a large number of time increments using the explicit method, the analysis can be more efficient in Abaqus/Explicit if the same analysis in Abaqus/Standard requires many iterations.

IV. OBJECTIVE OF WORK

Buckling analysis of a cylindrical shell under the action of various loads like Uniform axial load, torsion , Uniform External Lateral Pressure and combined action of axial and lateral pressure. These loading conditions are applied to a standard model of a cylinder using ABAQUS software and numerical results are validated by the buckling load which is found analytically.

V. FINITE ELEMENT MODELING AND ANALYSIS

This modeling is done using a certain package which is the numerical method of this project and is handled by ABAQUS version (6.10-1).This package has the ability to model both material and geometrical linearity's and non-linearity as per the need of the project. This package is used to study the buckling characteristics of shells at certain types of loading conditions. The loads acting are: (i) Axial loading (ii) Torsional loading (iii) Uniform lateral pressure loading (iv) Combined axial and uniform lateral pressure loading These are the loading conditions that will be modeled and analyzed in the package. This modeling and analysis is done to verify the theoretical values obtained from the equations. In this FEA a standard shell is chosen for our analysis purpose. Over this shell all the loading conditions are modeled and analyzed. A comparative study also we have done using the package is that when the size of the standard shell is extended and the buckling load variation is accounted. The specifications or the geometric details of the perfect shell models that will be used in the analysis for the axial and torsional loading conditions are given below.

1. Young's modulus =70000 MPa
2. Poisson's ratio = 0.3
3. Thickness = 1mm
4. Mesh Element size = 6.32mm

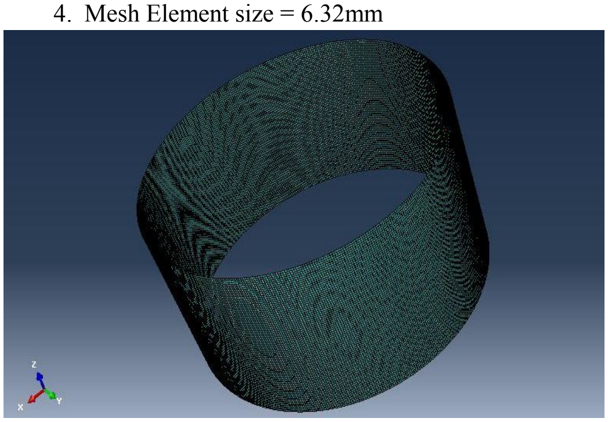

Figure 1: Finite Element Model

5.1 AXIAL BUCKLING

In this type of loading, one side of the shell is in a clamped condition, which means all the degrees of freedom on that side are restricted. At the other (loading) side of the shell except for the degrees of freedom along the length of the shell or the restriction about the axial the rest degrees of freedom are restricted. On this loading side concentrated load is applied at each of the node, total load acting is 1000N.

Buckling stress is given by

$\boldsymbol{\sigma CR} = \boldsymbol{Eh\ a}\ \mathbf{3(1{-}\vartheta 2)}$

Calculation of FEM buckling load:

Eigen buckling load , Pcr =λ* Papp where λ is the eigen value and Papp is the applied load

$$\sigma_{CR} = \frac{Eh}{a\sqrt{3(1-\vartheta^2)}}$$

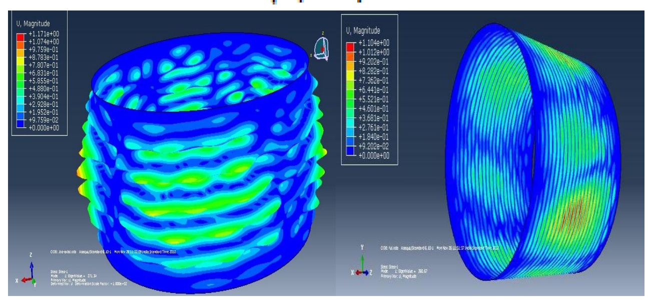

Figure 2: Axially buckled FE results

Model	Std	Full (2* Length)
Buckling Load (Analytical)	266193N	266193N
Buckling Load (FEM)	271340N	268670N

Table 5.1. Comparison between FEM values and theory for axial load

From the above table of comparison between the shells under axial loading it is evident that the buckling loads is getting reduced within a small range, eventhough they are found to be almost similar. This shows that the increment in length of the shell the buckling load of the shell is found to decrease or fall down. We can infer that the shell is more venerable to buckling when the length is increased.

5.2 TORSIONAL BUCKLING

Torsional loading is modeled using the cylindrical co-ordinate system. In this loading the boundary conditions on one side of the shell was completely restricted and on the other side of the shell where the load is applied except for the freedom along the radial displacement and rotation about the radial direction, the rest of the degrees of freedom is unrestricted. The load is applied on the loading side in a tangential direction with respect to the axial direction of the shell. Then the model is analyzed in ABAQUS for buckling analysis.

Buckling stress is given by

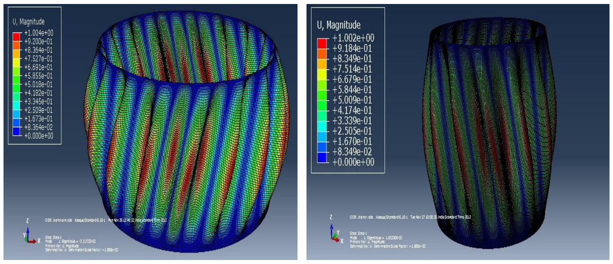

Figure 3: Torsionally buckled FE results

Model	Std	Full (2* Length)
Buckling load (Analytical)	22.96 MPa	15.8387 MPa
Buckling load (FEM)	23.1793MPa	16.5358 MPa

Table 5.2. Comparison between FEM values and theory for torsional load

From the above table of comparative study we can infer that on torsional loading of a standard shell of different lengths in the axial direction the buckling load is likely to vary in large quantity. It can be seen that as the length of the shell increased the buckling load of the shell starts to decrease considerably, from which we can infer that the shell is more venerable to buckling when the length is increased.

5.3 UNIFORM EXTERNAL LATERAL PRESSURE BUCKLING

In this loading scenario the cylindrical co-ordinate system is utilized while modeling the shell for the ease in specifying the boundary conditions and analyzing purpose. Here in this type of loading, on one side of the shell the freedom along the radial displacement and rotation about the angular direction is allowed free but restricting all other degrees of freedom in the side. On the opposite side of the shell along with the previously mentioned two freely allowed degrees of freedom, in this side of the shell the freedom of displacement in the axial direction is also allowed free.
Buckling stress is given by

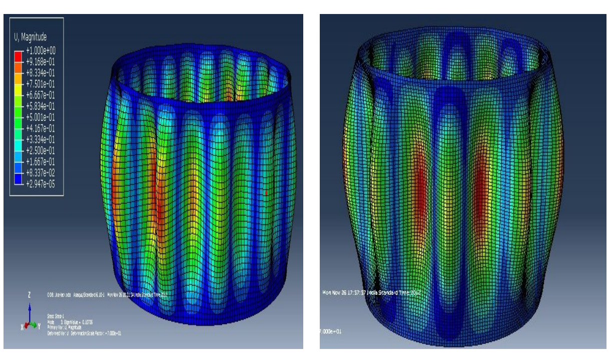

Figure 4: Laterally buckled FE results

Model	Std	Full (2* Length)
Buckling load (Analytical)	0.0928MPa	0.0486 MPa
Buckling load (FEM)	0.106 MPa	0.052 MPa

Table 5.3. Comparison between FEM values and theory for lateral pressure

From the above table we can infer that the buckling load variation with respect to change in the length of the shell has considerable effect in the uniform lateral loading case.

5.4 COMBINED LOADED BUCKLING- BOTH AXIAL AND EXTERNAL PRESSURE

This is a combined state loading where the shell is treated on two loading conditions simultaneously. The modeling is done in the cylindrical co-ordinate system. The boundary conditions, is that on one side of the standard shell the freedom of rotation about the angular direction is let free while arresting all the other degrees of freedom. On the opposite side or axial loading side of the standard shell, except for the freedom of displacement about the axial direction and the rotation about the angular direction, rest of the other degrees of freedom are arrested.
Buckling stress is given by

In this study of comparison between the buckling loads between the shells we see a considerable change on buckling load. The increment in length of the shell the buckling load of the shell is found to decrease or fall down considerably. This helps us to understand that the susceptibility of a shell to buckle under the same loading conditions is more as it length increases.
Along with this inference if we cross study the buckling due to uniform lateral pressure and buckling due to combination of both the uniform lateral pressure and axial loading, we can see that the buckling loads of each shell of respective lengths are found to be similar. This similarity in buckling loads helps us to understand that in buckling scenarios the buckling caused due to axial loads are found to be negligible when compared to buckling loads due to the uniform lateral pressure on the same shell

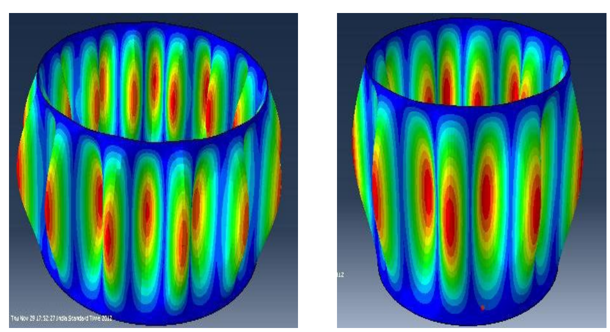

Figure 5: Combined loaded buckled FE results

Model	Standared	Full (2* Length)
Buckling load (Analytical)	0.089	0.042
Buckling load (FEM)	0.109	0.059

In this study of comparison between the buckling loads between the shells we see a considerable change on buckling load. This shows that the increment in length of the shell the buckling load of the shell is found to decrease or fall down considerably. This study helps us to understand that the susceptibility of a shell to buckle under the same loading conditions is more as it length increases.

SUMMARY AND CONCLUSIONS

The buckling load of the various models is analyzed and the analytical calculations of buckling loads for all the four loading conditions are discussed. In the case of axial buckling, buckling loads are almost same irrespective of variation in the length shows that the axial buckling loads are independent of the length of the shell and this matches with the conclusion made by Timoshenko[11] . It may also be concluded that the boundary conditions have negligible influence on the buckling load and hence it need not be considered as a contributing factor for reduction in the buckling loads. In the case of torsional buckling, it can be concluded that the shell is more venerable to buckling when the length is increased.

If we cross study the buckling due to uniform lateral pressure and buckling due to combination of both the uniform lateral pressure and axial loading, we can see that the buckling loads and FE results of each shell are found to be similar. This similarity in buckling loads helps us to understand that in buckling scenarios the buckling caused due to axial loads are found to be negligible when compared to buckling loads due to the uniform lateral pressure on the same shell. This helps to infer that in combined loading scenario the lateral buckling is dominant over axial buckling.

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