Dynamic Buckling Analysis of Composite Shell Under Impact

Milad Abdi¹, Zahra Zamirian², Masoud Mohammadi¹,*

¹Department of Mechanical Engineering, Buinza Zara Branch, Islamic Azad University, Buinza Zara, Qazvin, Iran
²Amir Kabir University of Technology, Faculty of Aerospace

ABSTRACT

This paper discusses the dynamic analysis of thin-walled composite shells under axial load. Analytical and numerical methods are used to analyze and validate the results with experimental results. This paper considers the effects of orthotropic materials, boundary conditions, geometry, geometric imperfections and time course load on the linear and nonlinear analysis of thin-walled composite shells. In the first analysis of the sample without considering geometric imperfections and then the effects of imperfections equal to the actual shell. The critical buckling dynamic is reported in load time that will show the sharp reduction in the buckling load with in increases load time and stiffness of shell is inversely to load time. This effect is more pronounced in the case of layup. With the increase of load time, the dynamic critical load dropped quickly and smaller than the critical buckling load is static. So in general, the choice of the static critical load is wrong as a criterion for dynamic design. In addition to the static buckling load, dynamic buckling load is reduced with increasing geometric imperfections of sample. Sensitivity to geometric imperfections is dependent on the type of the lay up. The geometric imperfections are the same effect on the dynamic critical load and static critical load for unlimited periods.

KEYWORD

Dynamic analysis, Buckling, Composite, Shell, Impact.

INTRODUCTION

For example, Liew and colleagues [1] performed dynamic analysis with mesh - free and Ritz method on a composite cylindrical shell. This paper studies the dynamic buckling of thin composite shells made from CFRP using the finite element method. Numerical finite element method allows different conditions to be studied. The dynamic axial compressive loading of cylindrical shells, considering all effective conditions in the buckling behavior of structures and predicting strength structures have been studied.

METHOD

Buckling analysis of composite shell finite element has been two common methods: ABAQUS/Explicit, ABAQUS/Standard. Buckle studied with three methods such as modes shape of buckling analysis, nonlinear static analysis and dynamic analysis. At first compares results of the three methods of analysis without considering the geometric imperfections and then will be evaluated the effects of geometric imperfections on the critical buckling load. This paper discusses the following important points:

1) Three methods finite element analysis are compared with the static buckling that Including modes shape of buckling, nonlinear analysis (Riks) and dynamic analysis.

2) Effects of geometric imperfections and sequence of lay up on amount of cylindrical shell buckling load analysis under the static load.

3) Static deformation is examined for all layers cylindrical shell and expressed some dynamic buckling behavior under load.

Composite shells used in this study consists of carbon fiber reinforcement that a radius, length and a thickness of the shell is 350, 700 and 1.32 mm and along the radius to thickness ratio is 265.

Two different types of lay up [0/45/−45/0], [45/−45] is studied that angle of fiber along the axis is zero degrees. The sequence of the lay up of shell is from outside to inside. The thickness of layers are 0.33 mm and material properties used are shown in Table 1.

Table 1. The mechanical properties of CFRP layers

<table>
<thead>
<tr>
<th>Elastic modules</th>
<th>Shear modules</th>
<th>Poisson's ratio</th>
<th>Thickness</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E_{11}$ = $E_{22}$ ($N/mm^2$)</td>
<td>$G_{12}$ ($N/mm^2$)</td>
<td>$v_{12}$ (ratio)</td>
<td>(mm)</td>
<td>(kg/m³)</td>
</tr>
<tr>
<td>52000</td>
<td>2350</td>
<td>0.302</td>
<td>0.33</td>
<td>1320</td>
</tr>
</tbody>
</table>

*Corresponding Author: Massoud Mohammadi
E-mail: mohammadi82@yahoo.com
Telephone Number: +989332212330
The finite element analysis were used of the full sample cylindrical shell. Thus, the modes shape of buckling and the buckling load for conditions are symmetric. Specimens was modeled with shell element S4R because of ABAQUS / Explicit is no a element 8 nodes. In order to can be compared analyze the results of ABAQUS / Standard with explicit methods. A 4-node shell element solved using the integral is reduced and is true for a large strain. After convergence of the mesh generator model is divided to 11440 Elements 6 degrees of freedom (220 elements in circumference and 52 elements in the axial direction). Boundary conditions was considered similar to buckling experimental tests [2], where all three components rotation and three components of the displacement of the lower sample was closed. While only the axial component of displacement where the top of sample is free. In this regard, the compressive axial load was applied to all nodes along the circumference of cylindrical shell.

Results and Discussion
At first the three methods described above for the two types of lay up static buckling shell without considering the geometric imperfections was studied and then the effects of geometric imperfections on the critical buckling load was assessed. The use of nonlinear analysis requires much more computing time but it gives more accurate results. To do this analysis was used modified Riks method. Finite element code ABAQUS / Explicit dynamic analysis does with use Lagrangian method and integral equations of motion in terms of time to explicitly that is reached where the critical buckling load with studied the load to displacement curve. Thus, it is very helpful to using the explicit dynamic analysis to study the buckling behavior of cylindrical shells under axial load.

Effects of geometric imperfections on the critical load
Due to the difference between the predicted analytical and experimental buckling load, Effects of geometric imperfections on the buckling load cylindrical shell is important. To simulate the geometric imperfections in cylindrical coordinates, shell considered the radius of $r = R + w^0$ that $R$ represents the radius of the shell and $w^0$ is the average value of the imperfections that provided with using laser-assisted scanning of the actual samples obtained from the references [2, 3, 4]. The amplitude of the mean value of geometric imperfections was calculated as 0.063 mm. Results of nonlinear analysis and dynamic analysis compared with the result of experimental tests that expressed the experimental conditions in reference [2].

The critical buckling load and high sensitivity to geometric imperfections is characterized and buckling load is calculated at this condition is 85% load in the shell without geometric imperfections. Tests and finite element model is shown in Figure 1.

Dynamic buckling analysis
Dynamic buckling of shells under axial load for two type of lay up and the effect of the geometric imperfections was studied varying the duration of impact load and amount of the load. And so the dynamic buckling load became quasi static. Critical load is lowest amount due to sudden large changes in the transient response system. Compressive axial load along the shell environment is applied on the nodes in the time limit interval and fixed amount. In tables 2 and 3, the critical buckling load for four time limit (1, 5, 10 and 15 ms) are presented with and without considering the effect of geometric imperfections.

Table 2. Dynamic buckling load and times limit in lay up

<table>
<thead>
<tr>
<th>Load time(ms)</th>
<th>Dynamic load without geometric imperfections (KN)</th>
<th>Dynamic load with geometric imperfections (KN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>134</td>
<td>122</td>
</tr>
<tr>
<td>5</td>
<td>109</td>
<td>96</td>
</tr>
<tr>
<td>10</td>
<td>87</td>
<td>72</td>
</tr>
<tr>
<td>15</td>
<td>83</td>
<td>69</td>
</tr>
</tbody>
</table>

Table 3. Dynamic buckling load and times limit in lay up

<table>
<thead>
<tr>
<th>Load time(ms)</th>
<th>Dynamic load without geometric imperfections (KN)</th>
<th>Dynamic load with geometric imperfections (KN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>364</td>
<td>248</td>
</tr>
<tr>
<td>5</td>
<td>167</td>
<td>137</td>
</tr>
<tr>
<td>10</td>
<td>133</td>
<td>119</td>
</tr>
<tr>
<td>15</td>
<td>117</td>
<td>109</td>
</tr>
</tbody>
</table>

To achieve good results of the above analysis were studied the axial displacement of shells. Figure 2 where the axial displacement of shell with lay up $[0/45/-45/0]$ without considering the geometric imperfections in the time limit of 10 ms with the effects of three amount of loads that applied to sample are shown. Where the axial displacement of shell curve - time to load 132 KN is logical response and fluctuate around the equilibrium state. From 132 to 134 KN large variations exist between the critical buckling load dynamic response due to the there is 133 KN. This study will show the effect of the dynamic buckling critical load and load
time and in Figure 3, the dynamic deformation of the load is shown at the time of 10 milliseconds for $[0/45/-45/0]$ shell Layup.

![Figure 2 - axial displacement of shell $[0/45/-45/0]$ where the load time of 10 ms](image)

![Figure 3 - deformation in $[0/45/-45/0]$ layup under dynamic load of 10 ms](image)

geometric imperfections is dependent on the type of the lay up. The geometric imperfections are the same effect on the dynamic critical load and static critical load for unlimited periods.

**CONCLUSION**

In the first analyze the sample without to consider geometric imperfections and then the effects of imperfections equal to the actual shell. The critical buckling dynamic is reported in load time that will show the sharp reduction in the buckling load with in increases load time and stiffens of shell is inversely to load time. This effect is more pronounced in the case of $[0/45/-45/0]$ layup. With the increase of load time, the dynamic critical load dropped quickly and smaller than the critical buckling load is static. So in general, the choice of the static critical load is wrong as a criterion for dynamic design. In addition to the static buckling load, dynamic buckling load is reduced with increasing geometric imperfections of sample. Sensitivity to

**REFERENCES**


