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STUDY THE STABILITY OF COMPOSITE LAMINATED THIN CYLINDRICAL SHELL WITH STIFFENERS

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Abstract

The determination of critical buckling load of stiffened composite cylinder is an important factor in determining the structural stability, which was done by ANSYS program and experimental investigation. Buckling of a stiffened composite cylinder is a very complex phenomenon that involves complex interactions between the skin and the stiffeners. Depending on different configurations of the skin and stiffener, different buckling failure modes and failure loads are observed in stiffened cylinders. In this work failure modes and buckling loads of stiffened composite cylinders under uniaxial loading condition is investigated by using analytical and experimental approaches.

A three dimension finite elements models was built which takes into consideration the exact geometric configuration and the orthotropic properties of the stiffeners and the shell. Based on the finite element model the different buckling failure modes are observed.

An experimental analysis was also performed to compliment the analytical method used to determine the buckling load of the stiffened cylinder. Good agreement in the results are presented .

دراسة الاستقرارية على القشريات الاسطوانية النحيفة المصنوعة من المواد المركبة مع الدعامات

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الخلاصة

ان حساب احمال الانبعاج للاسطوانات المركبة المدعمة بالدعامات هو عامل مهم في حساب استقرارية الهياكل ، حيث تم تنفيذ الحسابات في برنامج ANSYS و تم اجراء الدراسة عمليا، حيث أن ظاهرة الانبعاج في الاسطوانات المركبة تتضمن تداخل معقد بين جدار الاسطوانة والدعامات . وحسب اختلاف جدار الاسطوانة والدعامة فان طور فشل الانبعاج سوف يختلف وحمل الانبعاج يكون ملحوظا في الاسطوانات المدعومة. في هذه الدراسة فان اطوار الفشل وحمل الانبعاج للاسطوانات المدعومة بدعامات تحت ظروف احمال احادية قد تم دراسته عمليا ونظريا ضمن برنامج ANSYS . لقد تم اعداد موديل بالابعاد الثلاثة وحيث تم الاخذ بنظر الاعتبار الابعاد الاصلية والخواص غير المنتظمة للدعلامات وللجدار .وحسب طريقة العناصر المحددة فان اطوار فشل الانبعاج قد تم ملاحظتها . لقد تم انجاز التحليل عمليا لايجاد حمل الانبعاج الحرج للاسطوانة مسندة بالدعامات . وقد تم الحصول على توافق مقبول بالنتائج .

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Introduction

Grid stiffened cylinders are cylinders having a certain kind of stiffening structures either on the inner, outer or both sides of the shell. Cylindrical shells are subjected to any combination of in plane, out of plane and shear loads during application. Due to the geometry of these structures, buckling is one of the most important failure criteria. Buckling failure mode of a stiffened cylindrical shell can further be subdivided into global buckling, local skin buckling and stiffener crippling. Global buckling is collapse of the whole structure, i.e. collapse of the stiffeners and the shell as one unit. Local skin buckling and stiffeners crippling on the other hand are localized failure modes involving local failure of only the skin in the first case and the stiffener in the second case .

A grid stiffened cylinder will fail in any of these failure modes depending on the stiffener configuration, skin thickness, shell winding angle and type of applied load . Over the past four decades, a lot of research has been focused on the buckling , collapse, and post buckling behavior of cylindrical shells [1]. A work by Graham [2] presents analysis method for determining the buckling loads of ring and stringer stiffened cylinders. Wang et. al.[3] used the discrete method models stiffeners as lines of axial bending and torsional stiffness on the skin. Hilburger [4] predict the buckling load of a geometrically perfect compression loaded cylinder, and the prebuckling deformation and stress in the cylinder have an insignificant effect on the predicted bifurcation buckling load of the shell. A lot of work has been done in finite elements analysis pertaining to the investigation of buckling of stiffened cylinders [4,5] . Riddick and Hyer [5] study the effect of imperfections on the buckling load of cylinders. Optimization of grid stiffened composite cylinders is also an area of interest to many researchers. Jaunky et.al. have worked on optimizations of grid stiffened composite panels [6] as well as general stiffened composite circular cylinders[7]. In this study a three dimensional finite elements model was built using ANSYS finite elements software and the experiments are used to study the effect of the stability of composite laminated thin cylindrical shell with stiffeners .

Mathematical Model of Buckling Analysis

In the present study the displacement, velocities, stresses and loads are all time-dependent. The procedure involved in deriving the finite element equations can be stated by assuming the displacement model of elements as [8]

$$\vec{U} = (x,y,z,t) = \begin{Bmatrix} u(x,y,z,t) \\ v(x,y,z,t) \\ w(x,y,z,t) \end{Bmatrix} = [N(x,y,z)]\vec{Q} \text{ …………….. (1)}$$

Then deriving the element characteristic (stiffness and mass) matrices and characteristics load vector.

Hence , from Eq.(1) , the strains can be expressed as

$$\vec{\varepsilon} = [B]\vec{Q} \text{ …………………………………………………..(2)}$$

And the stresses as $\vec{\sigma} = [D]\vec{\varepsilon} = [D][B]\vec{Q}$ ………………..(3)

By differentiating Eq(1) with respect to time , the velocity can be obtained as

$$\dot{\vec{U}}(x,y,z,t) = [N(x,y,z)\dot{\vec{Q}} \text{ ………………………………..(4)}$$

The dynamic equation of motion of structure derived from using Lagrange equation [9]

$$L = T - \Pi \text{ ………………………………………………….(5)}$$

The kinetic and potential energies can be expressed :

$$T = \frac{1}{2}\iiint_V \rho \dot{\vec{U}}^T dV \text{ ……………………………………………(6)}$$

$$\Pi = \frac{1}{2}\iiint_V \vec{\varepsilon}^T \vec{\sigma} dV - \iint_{S_1} \vec{U}^T \vec{\Phi} ds_1 - \iiint_V \vec{U}^T \vec{\phi} dV \text{ ……………….(7)}$$

And the dissipation function of an element can be expressed as

$$R = \frac{1}{2}\iiint_V \mu \dot{\vec{U}}^T \dot{\vec{U}} dV \text{ ....(8)}$$

Where μ can be called the damping coefficient . by using Eq.(1) to (3) , the expressions for T, Π and R can be written as :

$$T = \sum_{e=1}^{E} T = \frac{1}{2}\dot{\vec{Q}}^T \left[\sum_{e=1}^{E} \iiint_V \rho [N]^T [N] dV \right] \dot{\vec{Q}} \text{ ....(9)}$$

$$\Pi = \sum_{e=1}^{E} \Pi = \frac{1}{2}\vec{Q}\left[\sum_{e=1}^{E} \iiint_V [B]^T [D][B] dV \right] \overline{Q} - \left(\sum_{e=1}^{E} \iint_{S1} [N]^T \vec{\Phi} ds_1 + \iiint_V [N]^T \vec{\phi} dV \right) - \vec{Q}^T \vec{P}$$

$$\text{............(10)}$$

$$R = \sum_{e=1}^{E} R = \frac{1}{2}\vec{Q}\left[\iiint_V \mu [N]^T [N] dV \right] \dot{\vec{Q}} \text{ ....(11)}$$

Where $\vec{Q}$ is the global nodal displacement vector , $\dot{\vec{Q}}$ is the global nodal velocity vector , and $\vec{P}$ is the vector of concentrated nodal forces.
By defining the matrices involving the integrals as :

Element mass matrix = $[M] = \iiint_V \rho [N]^T [N] dV$ ....(12)

Element stiffness matrix= $[K] = \iiint_V [B]^T [D][B] dV$ ....(13)

Element damping matrix = $[C] = \iiint_V \mu [N]^T [N] dV$ ....(14)

Hence , the dynamic equation of the body as:

$$[M]\ddot{\vec{Q}} + [C]\dot{\vec{Q}} + [K]\vec{Q} = \vec{P} \text{ ....(15)}$$

Linear buckling analysis in ANSYS finite – elements software is performed in two steps. In the first step a static solution to the structure is obtained. In this analysis the prebuckling stress of the structure is calculated. The second step involves solving the eigenvalue problem given in the form of the Eq.(15) without damping and assuming the displacement to be harmonic as

$\vec{\phi} = \vec{\phi} e^{i\omega t}$, following equation can be deduced [8]:

$$([K] + \lambda_i [M])\{\phi_i\} = \{0\} \text{ ....(16)}$$

Where

λ_i = ith eigenvalue and ψ_i = ith eigenvector of displacements

Buckling analysis by ANSYS finite element techniques[10]

A three dimensional model was built for stiffened composite cylinder using ANSYS9 finite elements software (Fig.(1)). The modeled cylinder has generated by element shell99 (Fig.(2-a)).

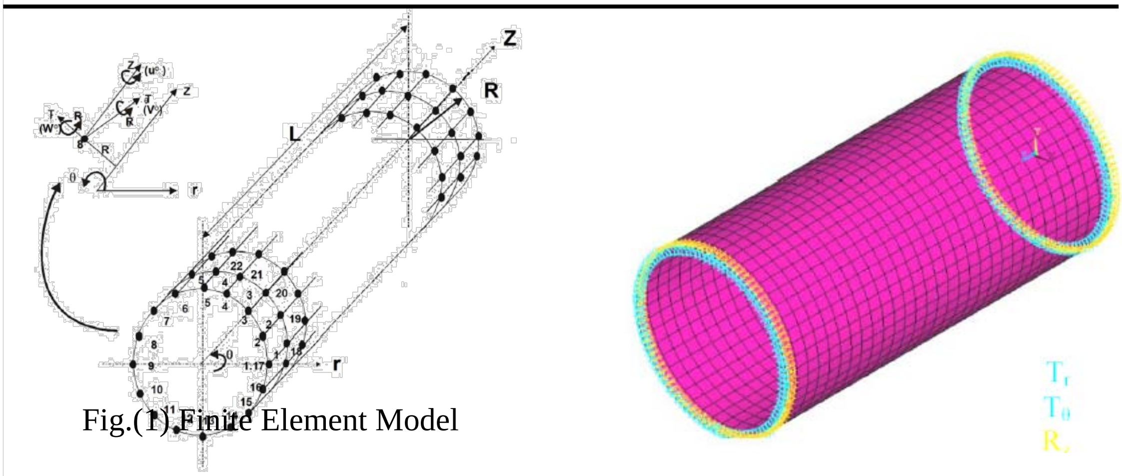


Fig.(1) Finite Element Model

The $\pm 60°$ stiffeners in the model is generated using element Beam188 as shown in Fig.(2-b) , with circular cross section where outer diameter equal to the inner diameter of the shell. The crossing over points of the stiffeners were modeled by matching the displacement of the corresponding stiffeners at these points. This was accomplished by merging the nodes of the crossing over stiffeners at the crossover points.

Fibers in the stiffeners are oriented along the length of the stiffeners. Hence, three different real constant tables were defined for the three stiffener orientations of 0°, 60°, and –60°. A local cylindrical coordinate system was then defined for each element and corresponding orthotropic properties aligned property. The stiffeners were modeled using element beam188, the skin was modeled by four layers .

The shell and stiffeners were glued at the interface, which upon meshing automatically merges the nodes of the shell elements and the beam element on the interface area. The shell was modeled using 8-node layered shell element (SHELL99).

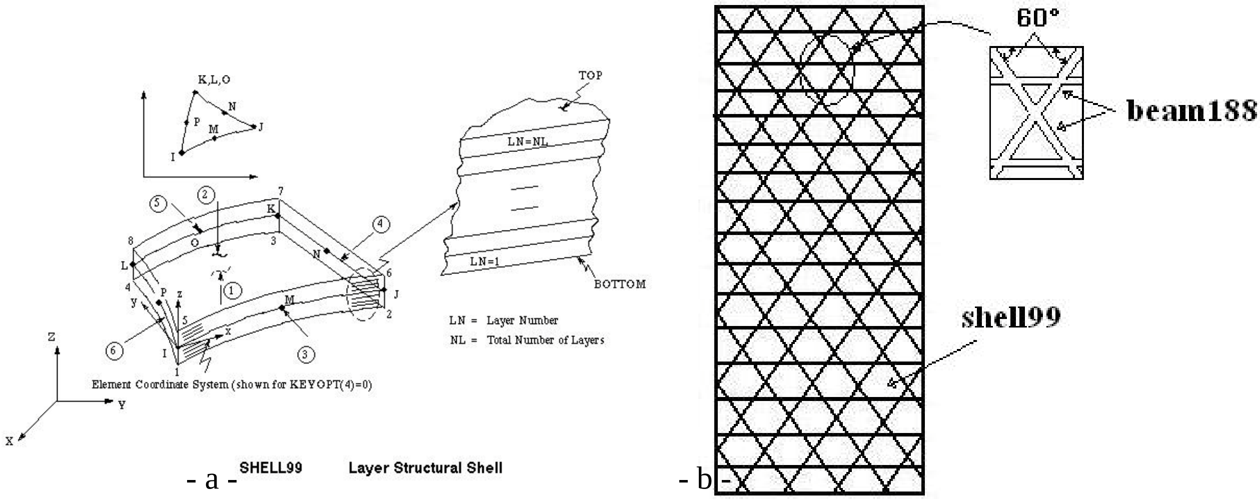


Fig.(2) a- Element shell99 b- Section of arrangement of the stiffeners

Two techniques are available in the ANSYS/Mechanical, programs for predicting the buckling load of a structure: nonlinear buckling analysis and eigenvalue (or linear) buckling analysis. Since these two methods frequently yield quite different results.

Eigenvalue buckling analysis predicts the theoretical buckling strength (the bifurcation point) of an ideal linear elastic structure. This method corresponds to the approach of

elastic buckling analysis: for instance, an eigenvalue buckling analysis of a column will match the classical Euler solution. However, imperfections and nonlinearities prevent most real-world structures from achieving their theoretical elastic buckling strength.
The buckling analysis is defined as the analysis type (ANTYPE, Buckle) and the analysis option (Bucopt) in which the solution methods is chosen either subspace iteration method (Subspac) which is generally recommended for eigenvalue buckling because it uses the full system matrices, and the other method is the Householder method (Reduced).
After that, the no. of eigenvalue to be extracted is chosen by activating the no. of modes to expand, the command (MXPAND).
Finite elements analysis was performed for stiffened composite cylinder having the dimensions shown in table(1). To study the buckling failure modes, different analyses were run by varying the skin thickness of the shell while maintaining the same configuration of stiffeners. The skin thickness was varied from 0.3 mm to 4mm .

Table (1) Dimensions of the model

Cylinder height	170 mm
Stiffeners orientation	0°,+60°,–60°
Horizontal stiffener spacing	36.5mm
Cross stiffeners spacing	40.5 mm
Shell thickness	0.3- 4 mm
Stiffener cross section	5x2.6 mm^2

Preparation of Testing Specimens:

The composites cylinder that prepared for this study were consisted of E-glass fiber plies in a thermosetting polyester matrix. This was manufactured by hand lay-up technique. The thicknesses of the glass fabrics were approximately (2.75 mm) with a real density of (400g/cm^2).
The resin matrix employed was a low viscosity thermosetting polyester resin commonly used for hand lay-up at room temperature. This resin is cured at 70°C and designed to wet easily the reinforcement fabrics employed in hand lay-up construction. The catalyst used was methyl ethyl keton polymer (MEKP)
After hand laying-up construction is done, the specimens were heated to 70°C in an oven with sufficient pressure to get rid of the excess resin and entrapped air bubbles. Fig(3) shows the tensile test results for composite laminated cylinder.

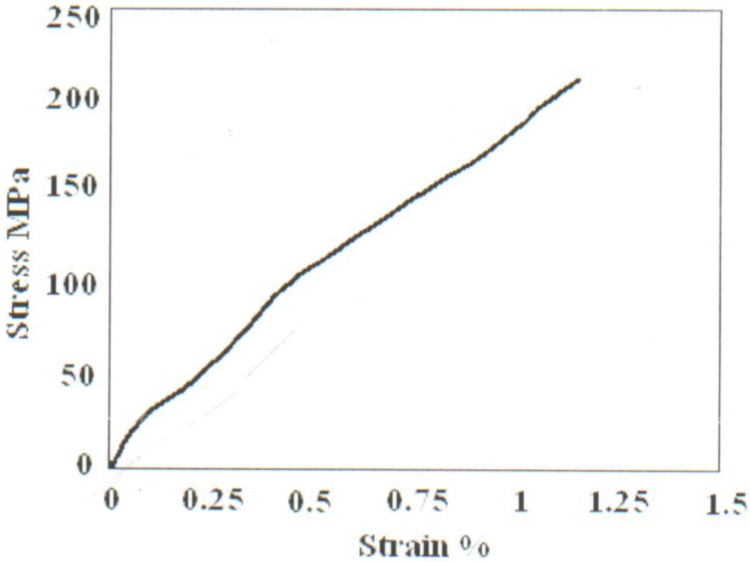


Figure (3) Tensile test results for Glass/ Polyester composite

Filament Winding : Process Technology [11]

To begin with , a large number of fiber roving is pulled from series of creels into bath containing liquid resin, catalyst and other ingredients such as pigments and UV retardants. Fiber tension is controlled by the guides or scissor bars located between each creel and resin bath. Just before entering the resin bath, the roving are usually gathered into a band by passing them through a textile thread board or stainless steel comb.

At the end of the resin tank. the resin-impregnated roving are pulled through a wiping device that removes the excess resin from the roving and controls the resin coating thickness around each roving. The most commonly used wiping device is a set of squeeze rollers in which the position of the top roller is adjusted to control the resin content as well as the tension in fiber roving.

Another technique for wiping the resin-impregnated roving is to pull each roving separately through an orifice. The latter method results in better control of resin content. Once the roving have been thoroughly impregnated and wiped, they are gathered together in a flat hand and positioned on the mandrel. Band formation can be achieved by passing through a stainless steel comb and later through the collecting eve. The transverse speed of the carriage and the winding speed of the mandrel are controlled to create the desired winding angle patterns.

After winding. the filament wound mandrel is subjected to curing and post curing operations during which the mandrel is continuously rotated to maintain uniformity of resin content around the circumference. After curing. product is removed from the mandrel, either by hydraulic or mechanical extractor as shown in Fig.(4) and Fig.(5) .

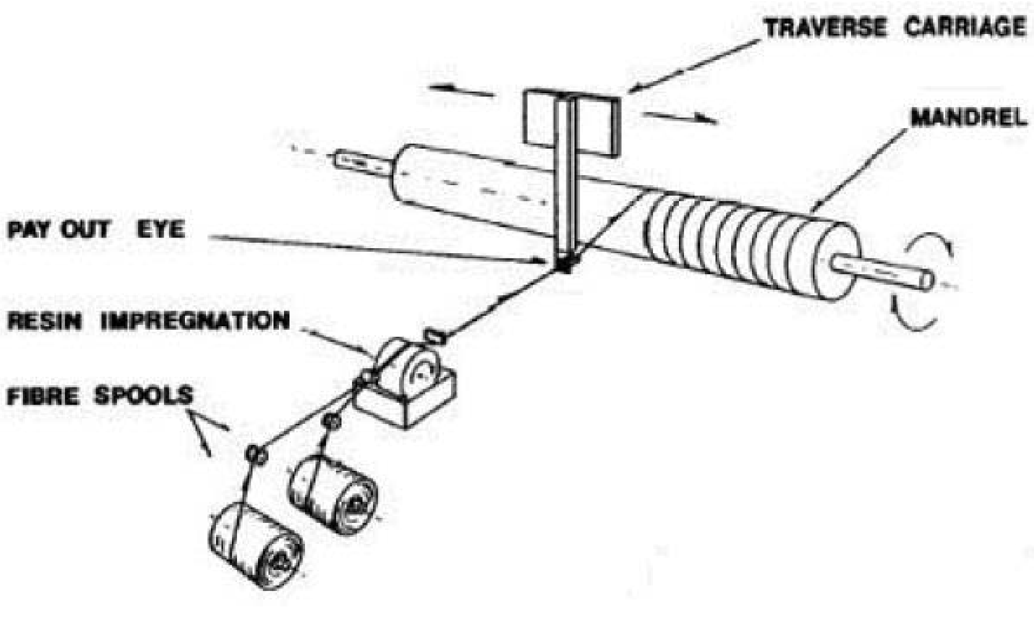


Fig (4) Schematic representation of the wet filament winding process

Fig.(5) Winding machine used in the present work

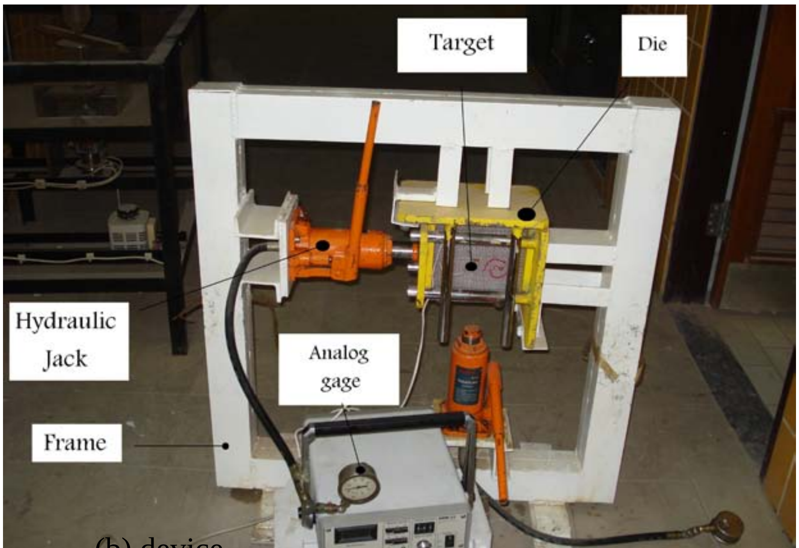
Compression device and its components[12]

The compression testing device (Fig.(6)) was fully designed and manufactured locally. The instrument was tested to verify its proper functioning during applying uniaxial or biaxial compression results. The device is consisted of:

1- Frame: The frame made from double steel channel (30 x 80 x 30 mm).
2- Die: The die is made of steel and divided into two parts:
3- Hydraulic Jacks: two hydraulic jacks with maximum pressing force are used to press the two plates of the die.
4- The two 600 bars analog pressure gages to read the applied pressure by the hydraulic jacks.

This device is manufactured so that the compression force can be applied either uniaxial or biaxial.

(a) specimen



(b) device

Fig. (6) The compression device and specimen

Results and Discussion

The comparison of the two different approaches used to calculated buckling load is presented. The comparison is based on analysis performed on the model having the properties given in table(1). These dimensions and configurations were chosen based on the stiffened composite specimen used for experimentation. The buckling load variation with the skin thickness for both finite- element analysis and experimental techniques is presented in table(2). All parameters are kept the same for both the Experiment model and the finite element model, with cross stiffeners oriented at $\pm 60°$. This result shows that the two methods almost the same values of buckling load in the global buckling failure mode range. While in the two local failure regions, the experiment model predicts different buckling loads compared to the finite elements model. This occurs because the equivalent orthotropic cylindrical shell developed using the experiment method will only fail in global buckling failure mode as opposed to the distinct three buckling failure modes occurring in the actual stiffened cylindrical structure.

Table(2) Experimental and finite elements result comparison

Skin thickness [mm]	Buckling critical Load [kN] Experimentally	Buckling critical Load [kN] ANSYS	Descripency %
0.3	200	100	50
0.5	190	310	63
1	500	620	24
1.5	720	810	12.5
2	920	1000	8.6
2.5	1260	1260	0
3	1500	1410	6
4	1600	1500	6.25

Fig.(7) Shows different mode shape of the cylinder. The cylinder with the thinnest shell thickness of 0.3mm was observed to fail purely due to local skin buckling (Fig.(8)). When the skin thickness was increased, the failure mode gradually changed to global buckling at about 1.5 mm skin thickness. At this point in addition to local buckling of the skin, the adjacent stiffeners started to buckle as well. With further skin thickening of the shell, the localized skin and stiffener failure spread to adjacent cells and gradually transformed to a more global buckling failure mode (Fig.(9)). At about a skin thickness of 3 mm, the shell was observed to be relatively stronger than the stiffeners and hence localized stiffener crippling started to occur. For any skin thickness more than 3 mm the local stiffener crippling failure mode prevailed (Fig.(10)). It should be noted that the global buckling failure mode observed is not fully developed as would result from a monocoque (unstiffened) cylinder. The failure is hence somewhat localized to a certain portion of the cylinder. It is also observed that there is no unique point at which the failure modes abruptly switch over to the next buckling failure mode but rather go through some transitional mixed buckling failure modes.

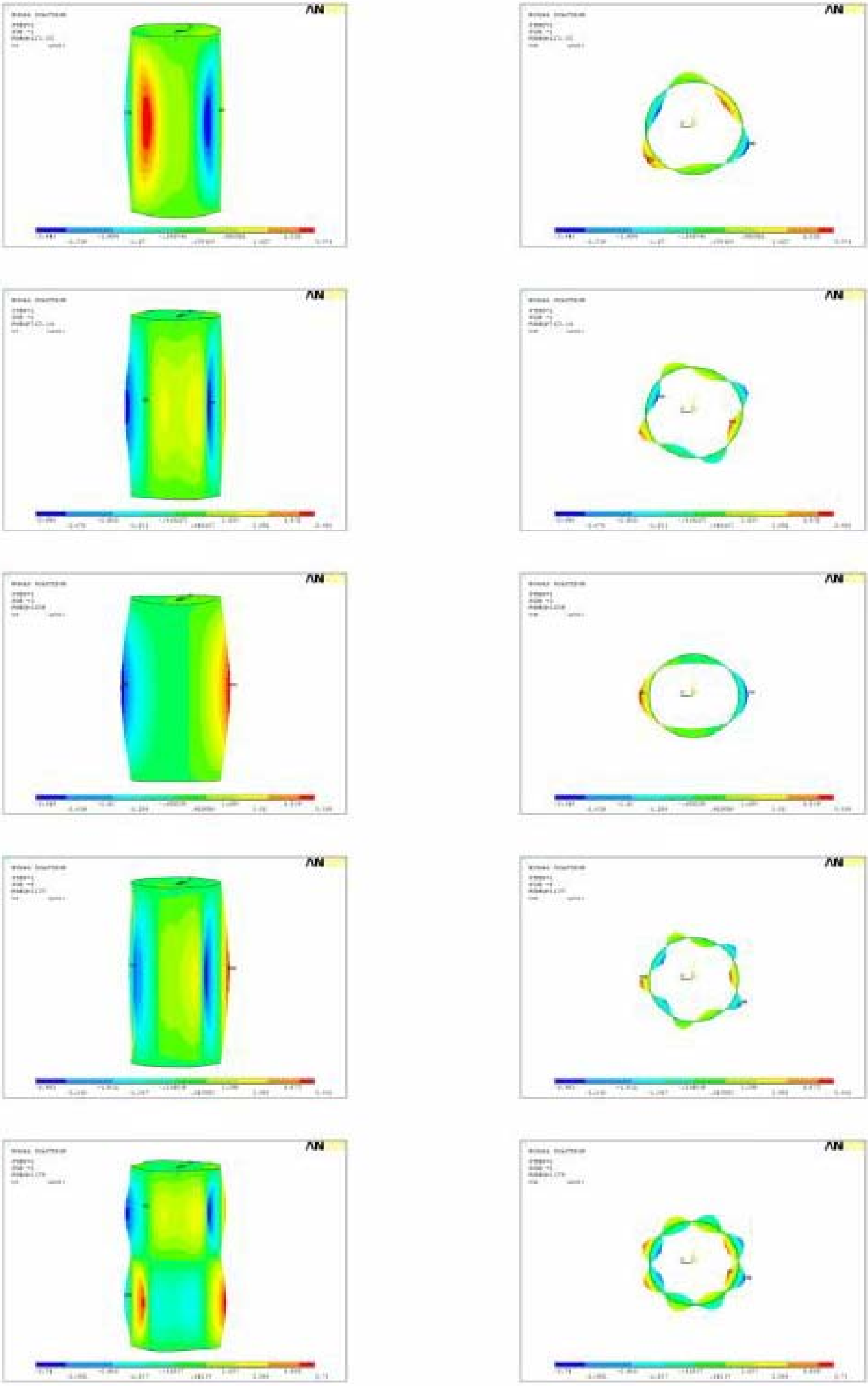


Fig.(7) Different mode shape of the cylinder under buckling condition



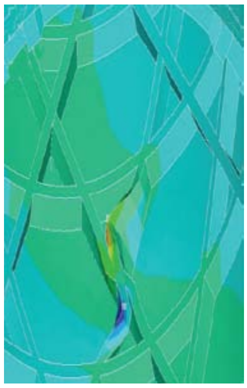


Fig.(8) Local skin Buckling Fig.(9) Global Buckling Fig.(10) Stiffener crippling

Conclusion

Finite Element model and the experimentation were successfully developed for the investigation of buckling problems of stiffened composite cylinders. The different failure modes of a stiffened composite cylinder were also studied .

Increase in skin thickness was shown to increase the buckling resistance of the stiffened structure continuously

These studies showed that the efficient utilization of material (load resistance per unit weight) highly depends on the buckling failure mode of the cylinder structure. For an isogrid stiffened cylinder, failure in global buckling mode resulted in the highest specific buckling load.

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