

NUMERICAL TESTS ON THE BUCKLING OF THE PLATES MADE OF COMPOSITE MATERIALS

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Abstract: A study on buckling analysis of the ship deck plate specimens, made of composite materials is presented. Certain deck plate specimens, of various thicknesses, so in the case of perfect manufacturing and in the case of existing imperfections are analysed. The analysis is based so on the analytical determination of the critical buckling load of perfect molding composite, and on the finite element modeling, made by licensed soft COSMOS/M. Analysis was made using various element types and meshes.

Keywords: composite materials, buckling, ship structures, FEM analysis

1. INTRODUCTION

Composite structures are increasingly being considered and used for lightweight, advanced applications, in areas with high corrosion, and in areas requiring the integration of the structure with other ship systems. Uses include composites for naval vessels, i.e. patrol boats, craft vessels, and corvettes; composite substructures; composite masts; composite propulsion systems, i.e. propellers, propulsors and shafts; composite secondary structures and machinery-fittings; and composite submarine structures, i.e. pressure hulls, control surfaces, and masts.

When composite materials are chosen for naval structures (navy ships), it is generally because they can offer properties, which are particularly attractive for that specific application. For example, composites are used in mine countermeasure vessels (MCMVs) due to their nonmagnetic properties. A major advantage of composite sandwich structures, compared to traditional stiffened steel structures, is weight reduction (usually 30% to 70%).

The design of any composite structure depends strongly on the fabrication methods. It is not always feasible to fabricate the optimum design configuration. It is very important for a designer to know the state of the art of the fabrication.

In case of composite structures, many materials show nonlinear stress-strain relationships and lamina strength may dominate the ultimate strength especially after lateral deflection grows by buckling when the panel has a laminate structure. Such behaviour is quite different from metal structures.

Using of composite materials for ship structures, especially for outside skin involves characteristic tests so numerical and experimental.

A study on buckling analysis of the column specimen, made of composite materials is presented. The plate specimen (column), so in the case of perfect manufacturing and in the case of existing imperfections was analysed. The specimens has an effective length of 200mm. For the clamped/clamped specimens the specimens is 300mm in length (50mm at each end were inserted into the grips). The thickness of the specimen was

dependent on the number of plies used. In this work 3 ply laminates was analyzed. These had nominal thicknesses of 1.6 mm. The lay-up was a 780g/m² 0/90 deg woven roving E-glass reinforcement (2x2 twill).

The critical buckling load of a plate will be analytically determined in two ways:

1) using derived equations, and 2) using the finite element soft COSMOS/M.

2. ANALYTICAL CRITICAL BUCKLING LOAD OF PERFECT PLATES

Buckling of FRP laminated plates is a complicated topic, and buckling solutions for only a few laminate cases have been published. The solution that will be presented is for a symmetric, especially orthotropic laminated plate clamped-free on edges.

The resulted efforts, on unit length may be written in the form

$$\begin{pmatrix} N_x \\ N_y \\ N_{xy} \\ M_x \\ M_y \\ M_{xy} \end{pmatrix} = \begin{pmatrix} A_{11} & A_{12} & A_{13} & B_{11} & B_{12} & B_{13} \\ & A_{22} & A_{23} & B_{21} & B_{22} & B_{23} \\ & & A_{33} & B_{31} & B_{32} & B_{33} \\ \text{symmetric} & & & D_{11} & D_{12} & D_{13} \\ & & & & D_{22} & D_{23} \\ & & & & & D_{33} \end{pmatrix} \begin{pmatrix} \epsilon_{x0} \\ \epsilon_{y0} \\ \gamma_{xy0} \\ \kappa_{x0} \\ \kappa_{y0} \\ \kappa_{xy0} \end{pmatrix} \quad (2)$$

where N are membrane forces, ϵ_0 are strains in the mean plane and κ_0 are curvatures.

A specially orthotropic laminate has no shear-extension coupling ($A_{13}=A_{23}=0$), no bend-twist coupling ($D_{13}=D_{23}=0$), and no bending-extension coupling ($B_{ij}=0$). The critical buckling load per unit length for a symmetric, especially orthotropic laminated plate clamped-free is

$$N_{x,cr} = \frac{k\pi^2 D_{11}}{a^2} b \quad (3)$$

where

$$D_{ij} = \frac{1}{12} Q_{ij} t^3. \quad (4)$$

$$Q_{11} = \frac{E_x}{1 - \mu_{xy}\mu_{yx}}; Q_{22} = \frac{E_y}{1 - \mu_{xy}\mu_{yx}}; Q_{12} = \frac{\mu_{xy}E_y}{1 - \mu_{xy}\mu_{yx}}; Q_{33} = G_{xy}; \quad (5)$$

k is a constant determined by the boundary condition and aspect ratio of the plate ($k=4$).

The critical buckling load per unit length is resulted of 337.29 N/m.

In the case of material used in our test

$E_x=18671\text{MPa}$, $E_y=18671\text{MPa}$, $G_{xy}=4.14\text{GPa}$, $\mu=0.26$, $\rho=1800\text{ kg/m}^3$.

$D_{11} = 5.95\text{ Pa}$; $D_{22} = 5.95\text{ Pa}$; $D_{12} = 1.55\text{ Pa}$; $D_{33} = 1.41\text{ Pa}$

A more accurate solution for the buckling load of general laminated plates (laminates having nonzero terms for all components of the bending stiffness matrix) has been done, but the solution procedure is complicated.

Equation (3) is considered suitable for this work and is compared to COSMOS/M buckling load results.

3. CRITICAL BUCKLING LOAD OF PERFECT PLATES

Critical buckling loads of “column plates” were found using the commercially available finite element software, COSMOS/M. Using COSMOS/M, an eigenvalue buckling analysis was done to determine the critical buckling load. Buckling can be defined as the sudden deformation which occurs when the stored membrane energy is converted into bending energy with no change in the externally applied load. Buckling occurs when the total stiffness matrix becomes singular

$$(\mathbf{k} - \lambda \mathbf{k}_g) \mathbf{d} = \mathbf{0} \quad (4)$$

where $\mathbf{k}_g$ is the geometric stiffness matrix (also referred to as stress stiffness matrix or differential stiffness matrix), and it is independent of the material properties of the structure in contrast to the conventional structural stiffness matrix. The multiplier λ represents the factor to the applied loads to cause buckling. The computation of buckling load factors is an eigenvalue problem, and certain methods which will be discussed under modal analysis are applicable. In the linear buckling analysis, the buckling loads and the associated mode shapes of eigenvalue buckling problems are calculated. In a typical buckling analysis, the quantities to be computed include the critical loads at which the structure becomes unstable, and the corresponding buckling mode shapes. For eigenvalue buckling, the first few modes are of practical importance.

Modal analysis which determines the natural frequencies and mode shapes is an important phase in the design of many structural components. Similar to buckling, modal analysis involves the computation of eigenvalues. The modules of software provide many types of eigenvalues extraction techniques, such as: Subspace iteration, Lanczos, Jacobi, Inverse power iteration (one pair only), Guyan Reduction.

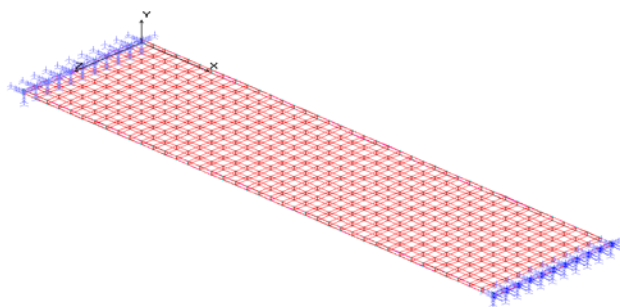


Fig. 1

The dynamic behaviour analysis was made for a perfect molded deck plate specimen, having a thickness of 1.6 mm.

The specimen has the characteristics:

(0/90 deg woven roving E-glass (780g/m³))

$E_x=18671\text{MPa}$, $E_y=18671\text{MPa}$, $G_{xy}=4.14\text{GPa}$, $m=0.26$, $r=1800\text{ kg/m}^3$.

The specimen was considered as clamped at the both smaller sides (Fig. 1).

For buckling analysis of the specimen, the following element types were used: SOLID, SOLIDL, SHELL4T, SHELL4L.

Table 1 - Buckling force [N] for perfect specimen

Element type	Analytical result	400 elements	10000 elements
SOLID	337.29	337.90	324.65
SOLIDL	337.29	335.77	321.51
SHELL 4T	337.29	325.51	323.92
SHELL 4L	337.29	323.44	322.23

In the table 1, the first buckling force values are presented for different model element type used. As it is seen, the value of force is decreasing when the number of elements is increasing for the same model type used.

4. CRITICAL BUCKLING LOAD OF IMPERFECT PLATES

The buckling analysis of the laminate composite plates was made for a imperfect molded plate specimen having a thickness of 1.6 mm, and 3 ply laminates. The specimen has an imperfection (cylindrical one) of 40 mm length and 2 mm high (Fig.2). The mechanical characteristics of the material (0/90 deg woven roving E-glass (780g/m³)) are: $E_x=18671\text{MPa}$, $E_y=18671\text{MPa}$, $G_{xy}=4.14\text{GPa}$, $\nu=0.26$, $\rho=1800\text{ kg/m}^3$.

The specimen was considered as clamped at $x=0$ and $x=a$, and free at $y=0$ and $y=b$.

In the table 2, the buckling force values for the different models are presented. As it is seen, the value of buckling force is decreasing when the number of elements is increasing for the same model type used.

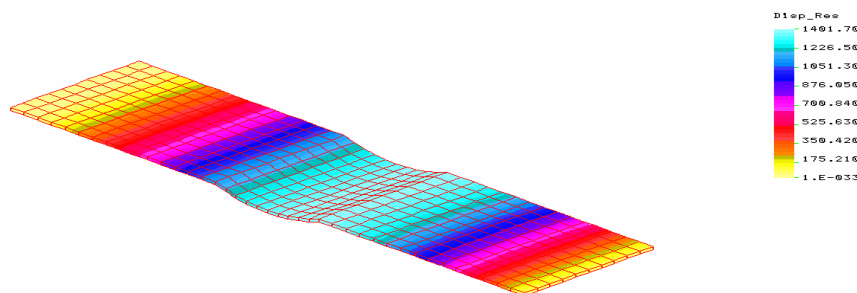


Fig. 2 – FEM model

Table 2 - Buckling force, in [N], for imperfect specimen

Element type	400 elements	10000 elements
SOLID	333.65	312.28
SOLIDL	322.26	307.48
SHELL 4T	326.35	324.68
SHELL 4L	324.94	323.54

5. REMARKS

For the perfect molded specimen, the analysis was made so using solution of the equations of composite buckling plates and columns and by using the finite element program COSMOS/M. The value of the critical force for the specimen having an imperfection is lesser than the one for the perfect specimen. Depending on the used model, the difference between the buckling force for imperfect and perfect specimen is placed between 0.5% and 5%. It has been shown, through the use of finite element analysis, that for the laminates analyzed in this work, a material orientation having (0/90) gives the best results for critical buckling loads for FRP laminates clamped-free on the edges. The local imperfection, similar to a hinge, facilitates the bending and so reduces the buckling load.

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