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PREDICTION OF BUCKLING AND POSTBUCKLING BEHAVIOUR OF COMPOSITE SHIP PANELS

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ABSTRACT

The behavior of ship plating normally depends on a variety of influential factors, namely geometric/material properties, loading characteristics, initial imperfections, boundary conditions and deterioration arising from corrosion and fatigue cracking. The analysis is presented for a uniaxially in-plane loaded, clamped, composite laminated quadratic plate. The imperfection is considered as the initial deformation due to the manufacturing operations: cosinusoidal shape in both of the longitudinal and transverse direction. Usually, the initial deformation mode appears in the form such as the fundamental mode of the buckling or vibration (figure 1) Variation of the maximum transverse displacement regarding the inplane load (displacement controlled after nonlinear buckling analysis) for three cases are presented.

Keywords: Composite panels, Buckling, Postbuckling

1. Introduction

The increasing interest in minimum weight designs for ship structures has generated important studies in the analysis of the elastic stability and postbuckling behavior of structures subjected to in-plane compressive loads. Due to their high strength and stiffness-to-weight ratios, laminated composite materials are increasingly being used in the shipbuilding industry.

The overall failure of a ship hull girder is normally governed by buckling and plastic collapse of the deck, bottom or sometimes the side shell stiffened panels. Therefore, the relatively accurate calculation of buckling and plastic collapse strength of stiffened plating of the deck, bottom and side shells is a basic requirement for the safety assessment of ship structures. In stiffened panels, local buckling and collapse of plating between stiffeners is a primary failure mode, and thus it would also be important to evaluate the buckling and collapse strength interactions of plating between stiffeners under combined loading.

The behavior of ship plating normally depends on a variety of influential factors, namely geometric/material properties, loading characteristics, initial imperfections (i.e., initial deflections and residual stresses), boundary conditions and deterioration existing local damage related manufacture imperfection or fatigue cracks.

For thin homogeneous (metallic) plates, classical plate theory predicts deformations and inplane stresses that are comparable to those of three-dimensional elasticity. In thin plates, transverse stresses are generally small compared to in-plane stresses, and thus, both classical theory and first-order shear deformation theory give satisfactory results. However, since both theories are two-dimensional, they are not accurate enough to predict transverse stresses directly.

Generally, composite plates support greater compressive load beyond the buckling load. Therefore, buckling and postbuckling behaviors of composite plates are very important factors of structural designs. There are more studies of buckling behavior than those of postbuckling behavior of composite plates. VandenBrink et al. (1987) investigated postbuckling behavior of graphite/epoxy composite plates of [qn–qn]s layup with a hole. Englestad et al. (1992) analyzed postbuckling response and failure of graphite/epoxy panels with and without holes. In this paper, a progressive failure (buckling) analysis was introduced into the nonlinear finite element analysis. Postbuckling behavior and compressive strength of composite plates was studied through analyses.

Parametric studies were performed in order to investigate the effects of initial deformations sizes on buckling load and postbuckling compressive strength.

Buckling and postbuckling analysis are essential to predict the capacity of composite plates carrying considerable additional load before the ultimate load is reached, and manufacturing-induced geometric imperfections often reduce the load-carrying capacity of composite structures. The nature of initial geometric imperfection induced during manufacturing is accounted for in the analysis. The stiffeners presence induces the cosine shape of the initial deformation. Examples of postbuckling analyses of symmetric cross-ply, angle-ply are presented, and the accuracy and performance of the method are examined. The numerical methodology presented can be used as an accurate and efficient tool for postbuckling analysis of imperfect composite plates.

2. Analytical study of global buckling of clamped orthotropic rectangular plate

For global buckling may be considered the theoretical study of the orthotropic, rectangular plate, without imperfections, clamped on the sides, and compressed along the *x*-axis. The external loading is considered as uniform one. The main orthotropic directions are considered as parallel with the sides. The critical value of the buckling load, p_{cr}, will be determined.

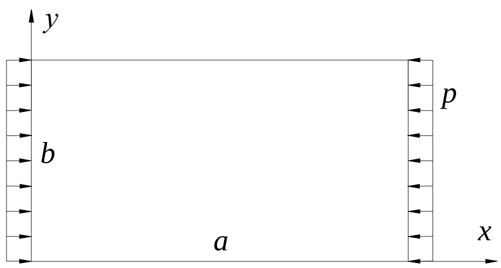

Fig. 1. *The perfect plate model*

The sides ratio is considered as $c=a/b$, and the loading per unit length is p. According to the figure 1, the coordinate system axis are considered parallel with the sides.

Before the buckling, the plate is considered loaded as in plane stress, with $N_x=-p$, $N_y=0$, $N_{xy}=0$.

The solution may be determined by choosing the static method. The equation of the deformed plate is

$$D_1\frac{\partial^4 w}{\partial x^4}+2D_3\frac{\partial^4 w}{\partial x^2\partial y^2}+D_2\frac{\partial^4 w}{\partial y^4}+p\frac{\partial^2 w}{\partial x^2}=0$$

(1)

where

$$D_1=\frac{E_1h^3}{12\left(1-\mu_{21}\mu_{21}\right)},\quad D_2=\frac{E_2h^3}{12\left(1-\mu_{12}\mu_{21}\right)},$$

$$D_k=\frac{G_{12}h^3}{12},\ D_3=D_1\mu_{21}+2D_k.$$

E_1, E_2, E_3, G_{12}, μ_{12}, μ_{21} are material characteristics.

The solution of the equation (1) is to satisfy the boundary conditions

- For x=0 and x=a: $w=0,\ \frac{\partial w}{\partial x}=0$; (2)

- For y=0 and y=b: $w=0,\ \frac{\partial w}{\partial y}=0$. (3)

The boundary conditions may be satisfied by the any term of the Fourier series

$$w(x,y)=\sum_m\sum_n\left[C_{mn}+A_{mn}\cos\frac{2m\pi(x-a/2)}{a}\cos\frac{2n\pi(y-b/2)}{b}\right]$$

(4)

where C_{mn} and A_{mn} are constant coefficients and m, n are integers. By substituting (4) in (1) it is obtained

$$p=\frac{4\pi^2\sqrt{D_1D_2}}{b^2}\left[\sqrt{\frac{D_1}{D_2}}\left(\frac{m}{c}\right)^2+\frac{2D_3}{\sqrt{D_1D_2}}n^2+\sqrt{\frac{D_2}{D_1}}\left(\frac{c}{m}\right)^2n^4\right].$$

(5)

The constants A_{mn} are undetermined. Equation (5) delivers all values of p, associated to the pairs of indices m=1, 2, 3,..., n=1, 2, 3... as possible deformed forms of the plates, such as the equation (4). From the all values, the smallest one will be chosen, this being the critical value. Obviously, the smallest value of p is obtained for n=1, that is corresponding to the shape with one semi-wave along the side b.

In the following, the value of m for the smallest value of p (named critical value) will be determined.

For n=1, the expression of p is

$$p=\frac{4\pi^2\sqrt{D_1D_2}}{b^2}\left[\sqrt{\frac{D_1}{D_2}}\left(\frac{m}{c}\right)^2+\frac{2D_3}{\sqrt{D_1D_2}}+\sqrt{\frac{D_2}{D_1}}\left(\frac{c}{m}\right)^2\right] \quad (6)$$

3. Numerical results obtained for buckling and postbuckling behaviour of clamped orthotropic imperfect plate

Numerical studies were performed on the buckling and postbuckling behaviour of the imperfect plate. In this case, the initial deformation is in the form given by the first buckling modal shape for the same plate but considered as a perfect one.

Table 1. *Plate lay-up*

α	t [mm]
$+45^0$	0.195
-45^0	0.195
4×0^0	2.36
$+45^0$	0.195
-45^0	0.195
6×0^0	3.54
α	t [mm]
$+45^0$	0.195
-45^0	0.195
4×0^0	2.36
$+45^0$	0.195
-45^0	0.195

Maximum initial deformation magnitude is considered as a rate from the thickness, *t*, that is

$$w_o = \gamma t.$$

In the analysis, for the rate γ the following values were chosen: 0.002, 0.32 and 0.96.

The FEM model, composed of shell elements, for geometric representation of a plate sized 320×320, with the thickness of 10 mm, having an initial deformation, was made.

The material used for the plate has the characteristics:

- E-glass/Polyester

E_1 = 46 GPa, E_2 = 13 GPa, E_3=13 GPa

G_{12} = 5 GPa, G_{13} = 5 GPa, G_{23} = 4.6 GPa

μ_{12} =0.3, μ_{23} =0.42, μ_{13} =0.3

- UD layers: t_1 = 0.59 mm
- Biaxial layers: t_2 = 0.39 mm, modeled as two UD layers having thickness $t_2/2$, at ±45°
- Lay-up is presented in Table 1.

Numerical studies, non-linear analysis (large displacements), were performed with FEM package COSMOS/M. For FEM analysis 3-node structural layered shell element (SHELL3L) were used.

Boundary conditions introduced on the sides were:

- On x=0 edge, completely fixed;
- On x=a edge, all DOF fixed, except x-translation;
- On y=0, y=b edges, all DOF fixed, except x and y translations and z rotation.

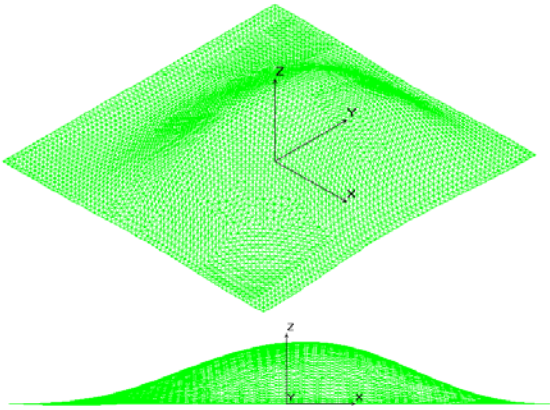

Fig. 2. *The plate geometry*

For the perfect plate, the critical value is p_{cr}=172.45 MPa.

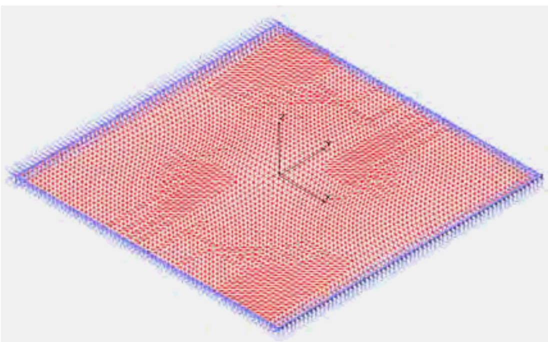

Fig. 3. *Plate mesh*

The theoretical buckling load obtained for a perfect plate is 207.21 MPa, determined only for future experiments calibrations.

Table 2*. Buckling load*

Case		Buckling load [MPa]
Theory		207.21
Perfect		172.45
Imperfect	Case 1: w_0=0.02 mm	181.66
	Case 2: w_0=3.2 mm	203.59
	Case 3: w_0=9.6 mm	288.56

Buckling load values obtained for the three cases of imperfect plates are presented in table 2.

As it is seen, the buckling load value obtained according to the equation (6) is far from that obtained for perfect plate, but is more closed to the value obtained for Case 3.

In figures 4, 5 and 6 the postbuckling behaviour of the imperfect plate for the three cases are presented. The results obtained after non-linear analysis show that the collapse does not occurs at buckling load value. The vertical dashed line corresponds to the buckling load for the perfect plate (172.45 MPa).

As it is seen, the buckling load is increasing with the increasing of the initial deformation.

4. Conclusions

This paper presents buckling and postbuckling results for composite imperfect plates, subjected to longitudinal compressive loading. The side edges of the plates are clamped. The imperfection is considered as initial deformation in the form given by the first buckling modal shape for the same plate but considered as a perfect one.

Based on the numerical results, there can be derived out the following conclusions of this study:

Values of the buckling load, p_{cr}, are increasing with the increasing of the initial deformation. That is, the stiffness of the plate is increasing with the increasing of the initial deformation.

Study made for the nonlinear buckling analysis show that the collapse does not occurs at buckling load value.

The displacement controlled nonlinear buckling analysis shows that at the same postbuckling load the maximum displacement is decreasing with the increasing of the initial maximum displacement.

The FEM models can deliver the whole range of the eigen modal buckling for the panel structure.

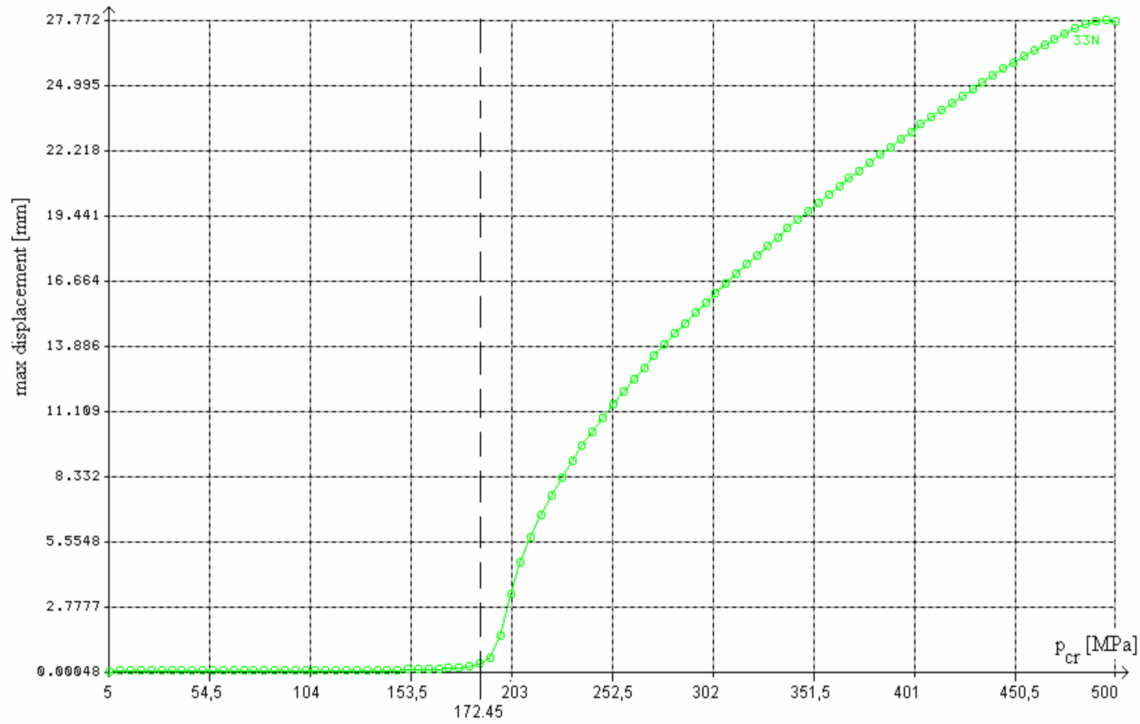

Fig. 4. *Case 1: Displacement controlled nonlinear buckling analysis*

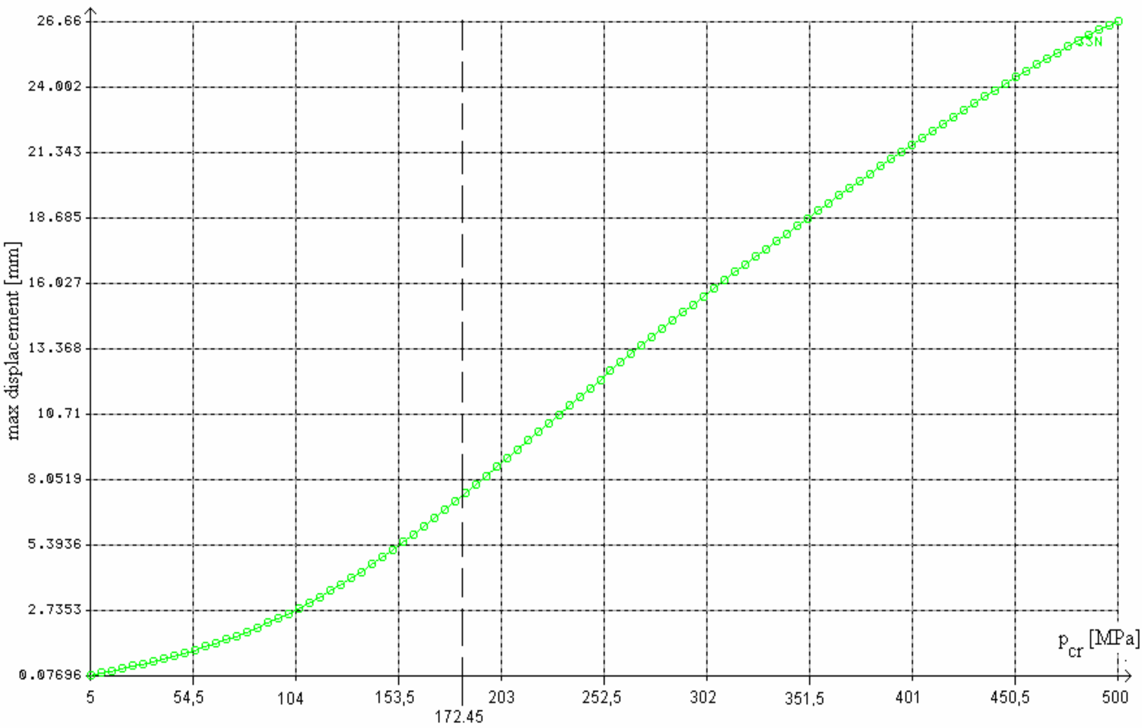

Fig. 5. *Case 2: Displacement controlled nonlinear buckling analysis*

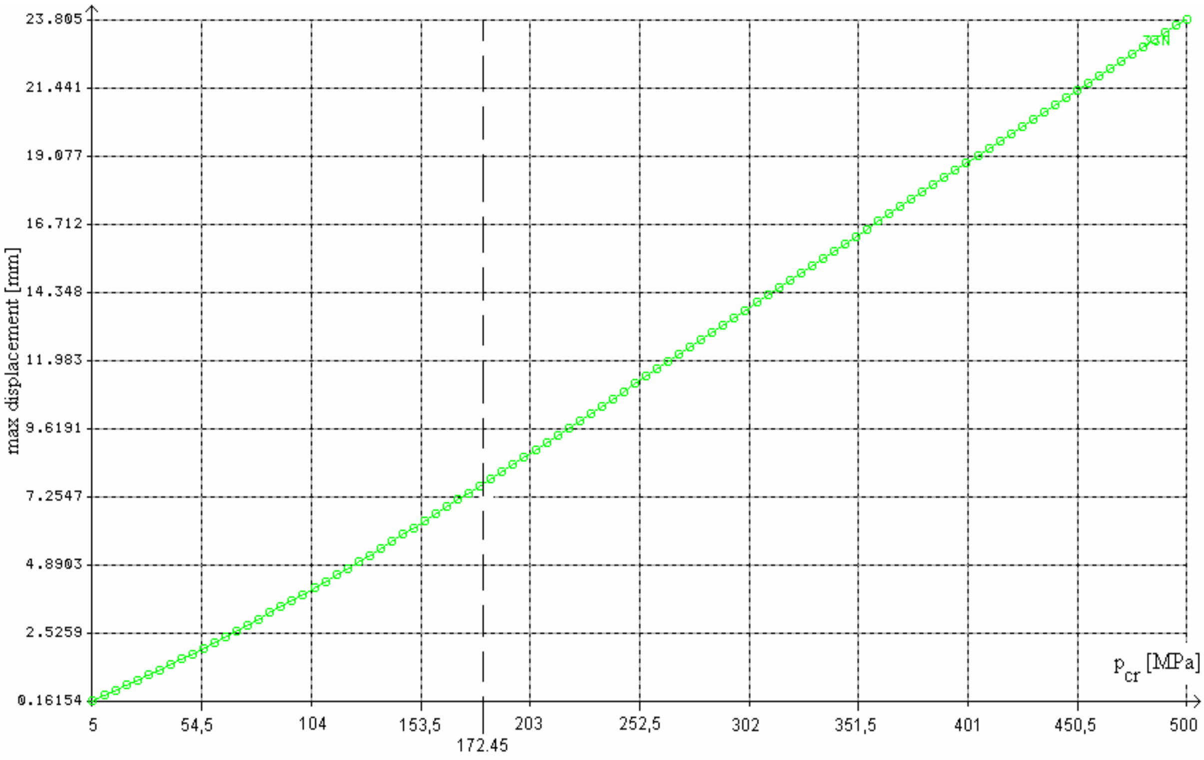

Fig. 6. *Case 3: Displacement controlled nonlinear buckling analysis*

Acknowledgements

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