Journal of Engineering and Applied Sciences 14 (19): 6959-6963, 2019
ISSN: 1816-949X

Experimental and Numerical Estimation of Plates Buckling Load

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Abstract: The present study focuses on the calculation of plates buckling load experimentally under uniaxial pure compression load. To achieve this goal, first of all, a buckling test apparatus has been designed and manufactured. The boundary conditions of the specimens are (Clamped-Clamped-Free-Free (CCFF)). Pure compression load was applied on 1 side of plate and the other side is constrained at all degrees of freedom. Two types of rectangular plates have been tested, namely 7075 aluminum alloy and low carbon steel with usual aspect ratios (a/b = 1.5, 2 and 2.5) where (a) is the plate length and (b) is the plate width. The experimental results were compared with the results of ANSYS Software for verification purpose. The results showed that the buckling load for low carbon steel plates is larger than aluminum plates, also, the buckling load decreases as the aspect ratio increases. The results showed a good agreement between experimental and numerical results of ANSYS.

Key words: Experimental buckling, rectangular plate, clamped-clamped-free-free plate, buckling load, numerical results, ANSYS

INTRODUCTION

Many engineering structures such as columns, beams or plates their failure develops by buckling. When a flat plate is subjected to low in-plane compressive loads, it remains flat and is in equilibrium condition. As the magnitude of the in-plain compressive load increases, however, the equilibrium configuration of the plate is ultimately changed to a non-flat configuration and the plate becomes unstable. The magnitude of the compressive load at which the plate becomes unstable is called critical buckling load (Arun, 2009).

Joshi *et al.* (2013) have discussed the buckling of rectangular plate with circular cut-out under biaxial compression using 2D FEM by ANSYS Software. The effect of holes, aspect ratio (a/b) width to thickness ratio (b/t) were studied. The results of clamped-clamped boundary conditions showed that the buckling load is more at top located hole, decreases with increase in aspect ratio, decrease with increase width to thickness ratio.

Jayashankarbabu and Karisiddappa (2014) used FEM ANSYS to obtain the buckling load for simply supported isotropic square plate containing circular, square and rectangular cutouts. The applied loads considered are uniaxial and biaxial compressions. It is found that cutouts have significant effect on the buckling load factor and the effect is larger when cutout ratios >0.3 and for thickness ratio >0.15.

Ibearugbulem *et al.* (2014) carried out the buckling analysis of rectangular plate with one free edge and other edges simply supported (SSFS). The study was completed through a theoretical formula based on polynomial series and the application of Ritz method. They have compared the values of buckling loads with the values of Timoshenko with the aspect ratios ranging from 0.1-2.0. It was discovered that for aspect ratios of 0.5 and 1.0, the percentage differences between the critical buckling load of their study and that of Timoshenko were -2.9576-5.4079%, respectively.

Ealavarasan and Selvarasu (2015) studied the influence of the length to thickness ratio, the aspect ratio, the fiber orientation and the cut-out shapes on the buckling load for the woven glass epoxy laminated composite plate in clamped-free-clamped-free configuration. The buckling load has been calculated experimentally and numerically by using finite element analysis using ANSYS Software. The results showed that the buckling load decreases as the length to thickness ratio increases as the aspect ratio (a/b) increases, the critical buckling load decreases the plate with $(0)_8$ layup had the highest buckling load and the plate with $(45)_8$ layup had the least the plate with circular cut-out yielded to the greatest critical buckling load while the rectangular cut-out failed for the lowest buckling load.

Dima (2015) have established a quick methodology to determine the critical buckling stresses of a flat isotropic plate with simply supported or clamped edges

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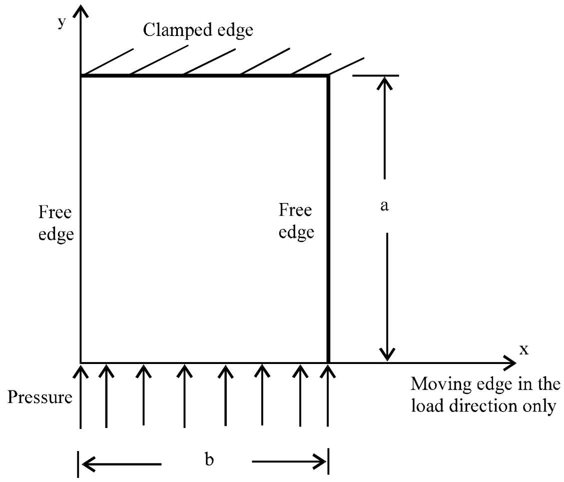

Fig. 1: Clamped-clamped-free-free plate under uniaxial compression

under single compression or single shear or combination between them. The comparison of the critical stresses obtained from the derived methodology and ASSIST 6.6.2.0 AIRBUS France Software shows there is a good agreement between them.

Viswanathan *et al*. (2015) studied the buckling of thin plates of linearly, exponentially and sinusoidal varying thickness and subjected to a constant in plane load. The plate is resting on an elastic foundation and is subjected to two types of boundary conditions on a pair of opposite edges with the other edges simply are supported. An elegant spline function approximation for the unknown transverse deflection is used and the buckling loads and mode shapes of buckling are computed.

Al-Waily (2015) evaluated the critical thermal effect caused the buckling of unidirectional and woven composite plate with different aspect ratio of plate combined from different types of unidirectional and woven reinforcement fiber and different resin material type. The thermal buckling load was calculated analytically and numerically by ANSYS Software for a simply supported orthotropic composite plate. The results showed that the critical temperature of unidirectional fiber more than the critical temperature of woven fiber and the buckling temperature increasing with increase the volume fraction of reinforcement fiber.

In the present research, the buckling load has been calculated experimentally, for rectangular aluminum and steel plates by simple an efficient homemade apparatus. The experimental results have compared with finite element method results using ANSYS Software.

Specifications of rectangular plate model: Figure 1 shows a rectangular plate with length (a) and width (b). The boundary conditions are: at y = a, the plate is constrained as all degrees of freedom are zero, at y = 0 the plate can move in y-direction only, i.e., $Uy \neq 0$, at x = 0 and x = b the plate is free at all degrees of freedom.

MATERIALS AND METHODS

Manufacturing of plate buckling test rig: The first step of experimental research was designing a buckling test apparatus which can be used to provide the required boundary conditions and loading case mentioned in the previous item. The designed rig is suitable to test a specimens with various aspect ratios (a/b). Moreover, a high capacity hydraulic jack has been used to provide a wide range compression load as shown in Fig. 2.

Experimental test procedures: The critical buckling load determination of plates can be briefly summarized by the following steps:

- Locate the center of the specimen plate and fix the plate by the two grips in the test rig
- Attach the deflection gauge dial at the center of the plate to record the lateral deflection of the plate
- Observe the plate side of buckling by applying a pressure and noticing, if the deflection gauge is reading or not
- Apply pressure gradually by the hydraulic jack and record the pressure value from the pressure gauge and the resulting lateral deflection by the deflection gauge
- The pressure gauge reading is representing the applied pressure on the hydraulic jack piston and must be convert to a hydraulic force from the relation:

$$F_1 = \frac{\pi}{4} d^2 \times P \quad (1)$$

Where:

F_1 = The compressive hydraulic force (N)
d = Hydraulic jack internal piston diameter (45 mm)
P = Pressure (N/mm^2)

Divid the resulting compressive force in Eq. 1 by the width of the plate:

$$F = F_1/b \quad (2)$$

Where:

F = The critical buckling load (N/mm)
b = The plate width (mm), loading side

At a certain load the deflection is still increasing while the load still constant although trying to increase it this load represents the buckling load. Draw the (load-deflection) curve and obtain the buckling load.

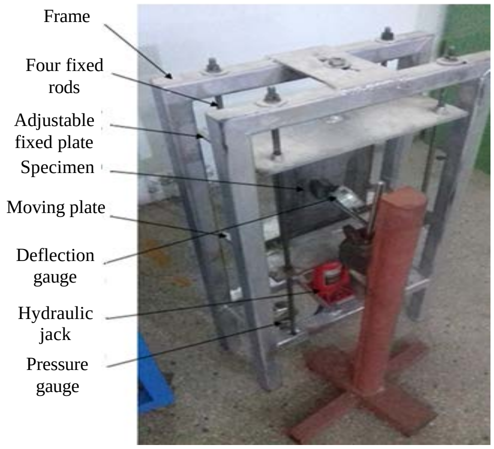

Fig. 2: Plate buckling test rig

Table 1: Specifications of the selected plates

a (mm)	b (mm)	Thickness t (mm)	Aspect ratio (a/b)
300	200	2	1.5
300	150	2	2.0
300	120	2	2.5

Numerical analysis: The finite element method software (ANSYS 9.0) has been used, so as to compare the experimental and numerical results. In this analysis (shell 93 element) is used. The element has 6° of freedom, translation and rotation in x-z (Table 1).

RESULTS AND DISCUSSION

The critical buckling load has been calculated for two types of materials with dimensions shown in Table 1. The mechanical properties are:

- The 7075 aluminum alloy E = 73 GPa, v = 0.33
- Low carbon steel E = 200 GPa, v = 0.3

Figure 3 and 4 show the experimental buckling test with various aspect ratios for 7075 aluminum alloy and low carbon steel plates, respectively. Furthermore, Fig. 5 and 6 show the critical buckling load being calculated by ANSYS Software for 7075 aluminum alloy and low carbon steel plates, respectively. Eventually, Fig. 7 shows the comparison between the experimental and numerical results for all tested plates.

The line graphs in Fig. 3 and 4, illustrates the buckling load with respect to lateral deflection for two types of materials, namely 7075 aluminum alloy and low carbon steel plates with different aspect ratios.

Overall, it can be seen that the buckling load for low carbon steel is larger than 7075 aluminum alloy plates. In

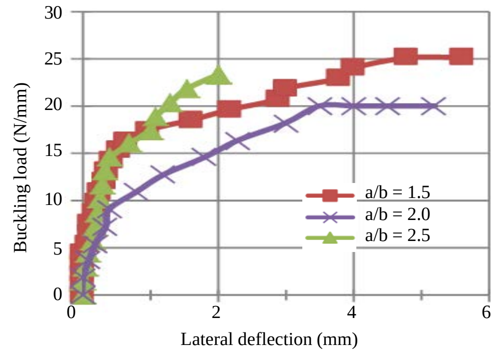

Fig. 3: Experimental buckling of 7075 aluminum alloy plates

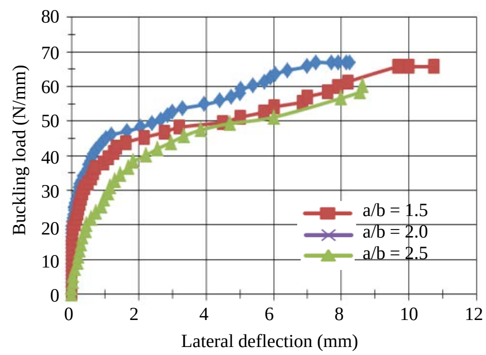

Fig. 4: Experimental buckling of low carbon steel plates

addition, the buckling load is decrease as the aspect ratio increase, the maximum buckling load has been resulted at a/b = 1.5.

From Fig. 3, it is obvious that for a/b = 1.5, the deflection was increased with gradual increase in the applied load before reaching the value (23.15 N/mm) this load still constant while trying to increase it but the deflection was still increasing in fact the plate has lucked the resistance to applied load at this value, so, this load represents the critical buckling load. As the aspect ratio was increased, the buckling load was decreased to be (22.94 N/mm) at a/b = 2. After that it was declined to be (22.7 N/mm) at a/b = 2.5. In contrast, Fig. 4, the recorded values are higher for low carbon steel plates where at a/b = 1.5, the critical buckling load was (66.8 N/mm) before declined to (54.8 N/mm) at a/b = 2.5.

Moreover, Fig. 5 and 6 show the contour plot of buckling analysis which have been done by ANSYS Software for verification purpose. Finally, Fig. 7 states the comparison of results which have been conducted in this analysis. A little consideration will show that a good agreement between experimental and numerical results.

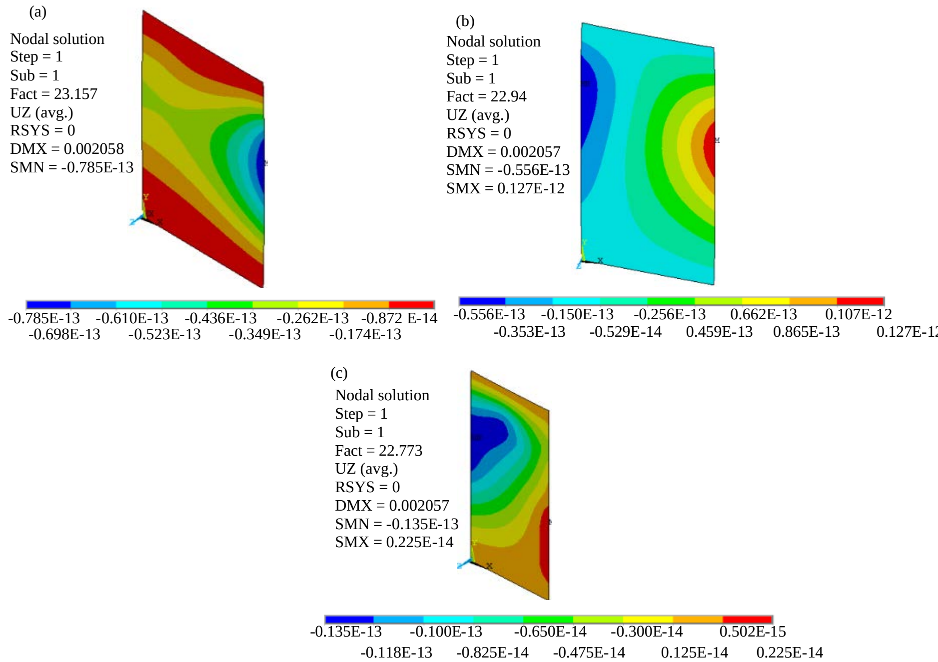

Fig. 5: Buckling load for 7075 aluminum alloy plates: a) a/b = 1.5; b) a/b = 2 and c) a/b = 2.5

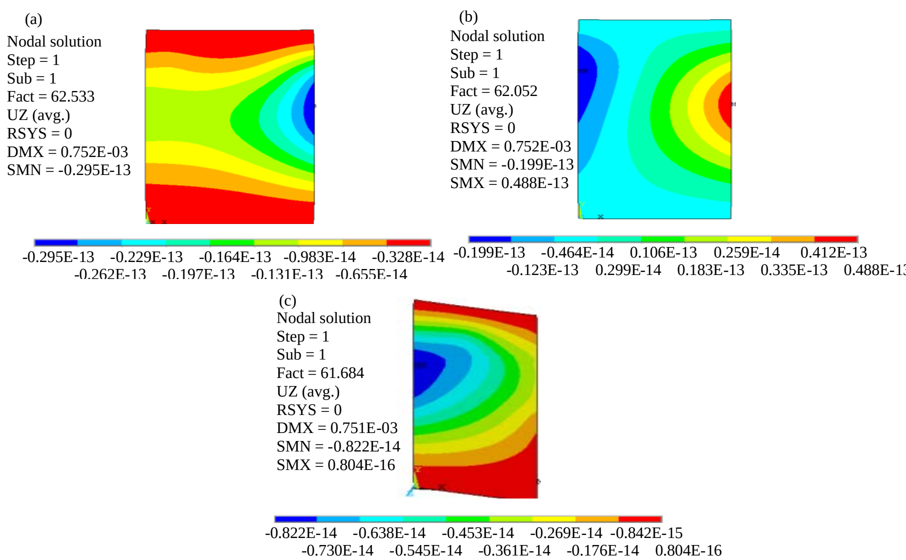

Fig. 6: Buckling load for low carbon steel plates: a) a/b = 1.5; b) a/b = 2 and c) a/b = 2.5

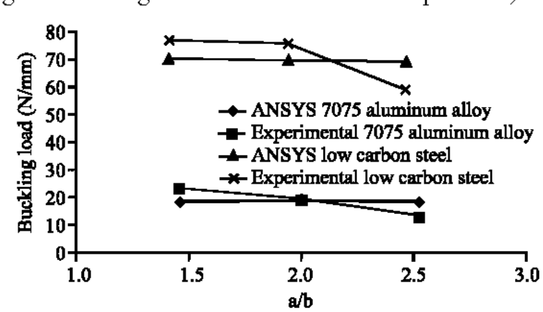

Fig. 7: Experimental and numerical buckling load

CONCLUSION

The buckling analysis of clamped-clamped-free-free rectangular plates has been studied under uniaxial load. This study has concerned deeply on the experimental research, so, a common materials have been adopted, specifically, 7075 aluminum alloy and low carbon steel in this analysis. A homemade apparatus, simple and efficient has been manufactured to achieve this task. To trust our experimental results, finite element Software ANSYS was used for numerical calculation of plates buckling load. The conducted results clearly showed that the buckling load is affected by the aspect ratio and material type. A good convergence between experimental and numerical results is achieved. Future studies may concern on composite plates with or without holes with various constrained conditions.

ACKNOWLEDGEMENT

Thanks are given to the Laboratory at the Engineering College of University of Diyala for getting this great opportunity doing this research.

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