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On the Effect of Imperfection on Buckling load of Perforated Rectangular Steel Plates

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Abstract

Buckling behavior of plates and shells is one of the important characteristics in analysis of any structure. One the most significant parameter that must be considered in buckling phenomenon is imperfection. In this paper the effect of imperfection on buckling load of steel rectangular plates under uni-axial in-plane compressive loading is investigated by numerical and experimental methods. The plates were free on two opposite sides and simply supported at the load side whereas the opposite side is either clamped or simply supported. This means that the plate primarily exhibits a type of column's buckling.

Keywords: Buckling, perforated rectangular plates, finite element, imperfection.

Introduction

Using plates and shells in fabrication of many structures is unavoidable. These structures may experience axial compression loading in their longevity and buckle through these loads. Furthermore, these structures usually have discontinuities, such as cutouts, which may have effects on their stability. Therefore we must consider any parameter that affect on buckling load. Buckling analysis of thin-walled structures under axial compression has been investigated by many researchers[1-2]. Also thin plates under the concentrated and the distributed forces on the whole plate edge have been investigated by some other researches[3-4]. Of course, the stability of these plates is dependent on many parameters such as type of support, type of loading, ambient temperature, imperfection and etc. a direct matrix method for the bucking loads on structures was applied to the examination of the stability of flat square plates with central square perforations under various combinations of bi-axial loading by Yetterman and Brown[5]. Their results were given for a wide range of hole to plate size from 0 to 0.8. Christopher et al.[6] studied the elastic stability of plates containing perforations using the conjugate load-displacement method. The effect of the size of a central square hole in a square plate on the elastic buckling load was investigated. Chang-jun and Rong[7] have been provided and solved a system of new boundary integral equations to address the plane stress, critical loads and post-buckling problem for perforated thin plates, based on the general bifurcation theory and mathematical model for the stability analysis of perforated thin plates. Their results show that the boundary element method was efficiently applied to the post-buckling analyses of perforated thin plates.

Also in another study Shariati et al[8] investigate the buckling of tubular steel shells with elliptical cutout subjected to oblique loading. In their study the influence of shell length, shell diameter, shell angle and diameters of elliptical cutouts on the predicted buckling values has been explored. Shanmugam et al.[9] studied on post-buckling behaviour and the ultimate load capacity of perforated plates with different boundary conditions under the uniaxial and biaxial compression. They used the finite element method (FEM) and compared their numerical results with the experimental results. They established a design formula to determine the ultimate load of buckling of perforated plates. Suneel Kumar et al.[10] have been presented the details of tests carried out on the collapse load of stiffened plates with and without cutout, with reinforced cutout and initial imperfections. Based on strut approach and orthotropic plate approach, a generalized computer program for the semi-analytical solutions proposed by various investigators was developed. They proposed an approximate method based on strut approach to calculate the collapse load of stiffened plates with cutouts and initial imperfections. Eccher et al.[11] provided the application of the isoparametric spline finite strip method to the elastic buckling analysis of perforated folded plate structures. They introduced the general theory of the isoparametric spline finite strip method. A number of numerical examples of flat and folded perforated plate structures illustrated the applicability and accuracy of their proposed method. The linear buckling analyses of square and rectangular plates with circular and rectangular holes in various positions subjected to axial compression and bending moment were developed and has been studied by Maiorana et al.[12] The aim of their paper was to give some practical indications on the best position of the circular hole and the best position and orientation of rectangular holes in steel plates, when axial compression and bending moment acted together.

As depicted in above review, the cut out existing in the plate, the geometry of cut out, the type of loading, the supports of the plates are all among the factors which can have a great effect in the stability of the plates. In this paper, the numerical and the experimental investigation on the buckling behavior of the rectangular plates with circular and square cut outs under uni-axial in-plane compressive loading in elasto-plastic range with various loading bands are performed. The plates are free on two opposite sides and simply supported at the load side whereas the opposite side is simply supported. This means that the plate exhibits primarily a column type of buckling. Moreover, the relation between stability of the rectangular plates having square and circular cut out with the same cross section has been studied. Furthermore, the effects of the various loading lines have been studied numerically using ABAQUS. Also, some buckling experimental tests have been performed on the samples. Numerical and the experimental results are in good agreement.

Material and Methods

The meshed geometry and type of loading are shown in figure 1. In this figure, *l* is the loading band which varies in the range of ($0\leq l\leq a$). The position $l=0$ is relates to the concentrated load exerted on the middle of the width of the plate, *a*, and the position $l=a$ represents to the distributed load exerted on the entire the width of the plate.

In this investigation, the structural steel rectangular plates with 100 x 150 x 2.07 mm dimensions are used. These plates have square and circular cut outs. The side of the square cut out is considered to be *e=30 mm* and for having the same area of two types of cut out, the diameter of the circle is considered to be *D=33.84mm*. The lower edge of the plates has been placed in the clamped support and on the other edge, patch compression with various bands has been exerted through a simply support. The tests have been conducted for the width of loading of *l*= 15,30,50,75 and 100 mm. Only the loaded section of the boundary is constrained and the rest is free as it is shown in Figure 1. In these tests, the post buckling behavior of the plates has been fully studied, too.

The mechanical properties of the tested structural steel plates have been specified through the tensile test in accordance with the ASTM-E8 standard using an INSTRON8802 servohydraulic machine with stress-strain curve as displayed in figure 2-a. The obtained elasticity modulus from linear elastic region equal to *E=218Gpa* as it shown in figure 2-b. Moreover the Poisson's ratio value is considered to be $\nu=0.33$. The data of the plastic region of the stress-strain curve has been used for analysis of the plastic behavior in ABAQUS software.

The numerical analysis: In Abaqus after defining the geometry, boundary conditions and the applied loading, we must mesh the structure for analysis. This is achieved through using the S8R5 quadrilateral non-linear elements. This element is an 8-node element with 5-degree of freedom including 3 displacements in three directions of the coordinate axes and two rotations for each node. In this element, 8 nodes have been placed in such a way that 4 nodes have been placed on the square corners and 4 other nodes have been placed in the middle of the sides of the square (figure 3.). The S8R5 element is very suitable for the element arrangement of the thin plates and in this research all the analyses have been performed using this type of element[13].

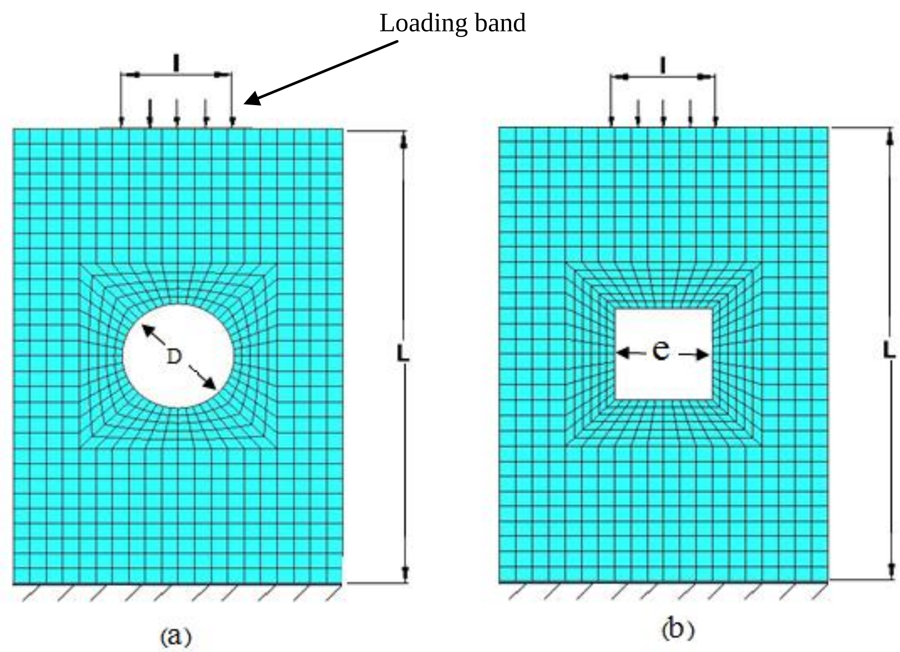
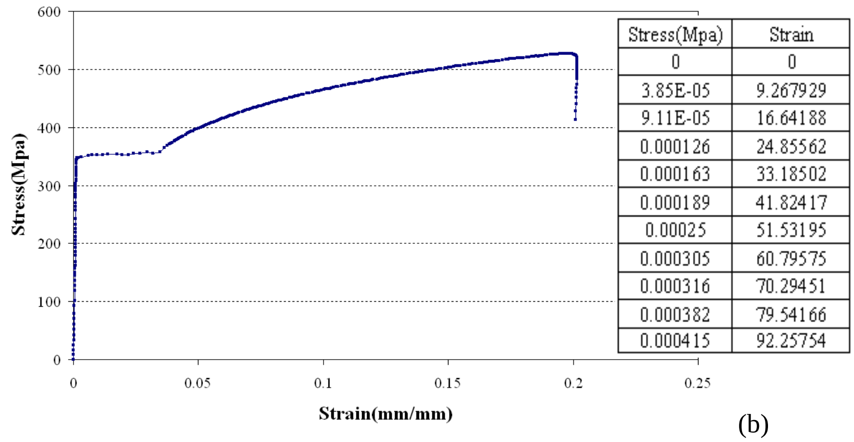


Figure-1
The meshed geometry of the plate and the type of the applied loading (a) The plate with circular cut out (b) The plate with square cut out

(a)



Stress(Mpa)	Strain
0	0
3.85E-05	9.267929
9.11E-05	16.64188
0.000126	24.85562
0.000163	33.18502
0.000189	41.82417
0.00025	51.53195
0.000305	60.79575
0.000316	70.29451
0.000382	79.54166
0.000415	92.25754

(b)

Figure-2
(a) The specimen being under the tensile test (b) The stress-strain curve resulted by the experimental test

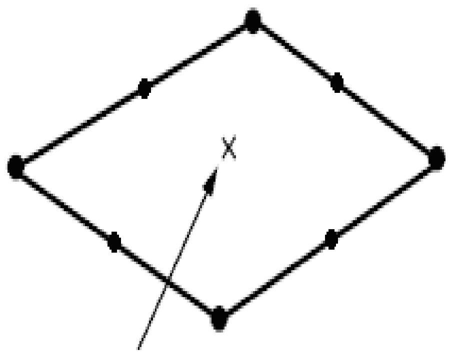


Figure-3
The S8R5 element

After meshing the specimen, a linear buckling analysis for getting eigenvalues is performed by using the “Buckle” solver in ABAQUS and the buckling mode shapes are obtained. Since in eigenvalue linear analysis, the plastic properties of the specimen are not taken into account, a linear finite element analysis (eigenvalues), particularly for the plates made of ductile materials, overestimates the real value for buckling load. Since buckling usually occurs in smaller mode shapes, a linear analysis should be performed first for all specimens, to find the mode shapes with smaller eigenvalues. For example, the three buckling mode shapes has been displayed in the figure 4. The displacements for these mode shapes are saved in a specific file and used in the next analysis (Static Riks), so that the effect of mode shapes are considered in buckling analysis. Otherwise, the software would choose the buckling mode in an arbitrary manner that resulting unrealistic results. This step is called Buckle in the software. For this step, the Subspace solver method of the software was used and three primary mode shapes were obtained. Also a numerical calculation was made over the loaded band with a constant displacement.

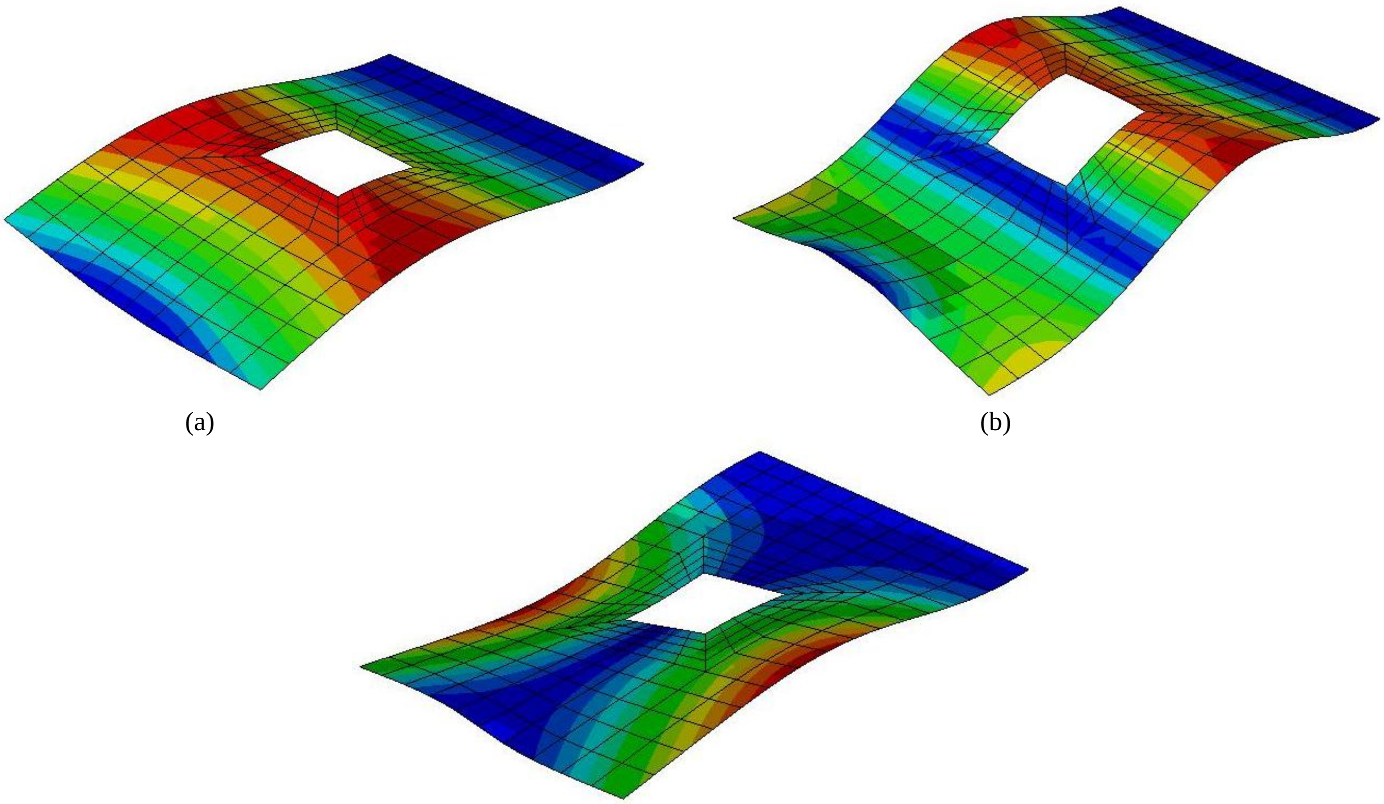
Since the imperfection of the plate considerably affects on ultimate buckling load, the imperfection is an important factor which must be considered in buckling of the plates.

Imperfection means the every deviance of ideal manner of each structure. To make this point more transparent and clear, the effect of the imperfection on the ultimate buckling load of a rectangular plate has been compared numerically with two different values of the imperfection. The sample is a plate with a square cut out which is placed with clamped support in lower side and is applied under a 15 mm loading band by a simply support in upper side. The critical buckling loads of 13.298kN and 11.856kN have been obtained for the values of the maximum imperfection amplitude, 0.1 and 0.48 mm, respectively. The difference between these two loads is 1.442kN and as depicted in figure 5, it is a considerable amount. Therefore, prior doing the non-linear analysis, the amount of the imperfection must be determined. As a result in this article, the maximum imperfection amplitude (the preliminary bend) of all plates has been considered to be equivalent to 0.48 mm. It is worth mentioning that, this quantity has been calculated through the quantity of the maximum imperfection amplitude found in the plates used in experimental tests.

Effect of loading band on buckling behavior of plates: In this part the effect of the loading band on buckling behavior of perforated plates has been numerically studied. In this research the loading band is varied between 15 to 100 mm. For sample, the behavior of load versus vertical displacement of plate with circular cutout (load-displacement curves) under various loading has been displayed in figure 6.

Also load-displacement curves for plates with circular and square cutouts and perfect (with no cut out) plate for loading band l=100 mm are compared in figure 7. Also the critical buckling loads of plates with and without cut out for different loading bands are given in tables (1).

As shown in the curves, after the load of the plate reaches to the critical value, the buckling phenomenon is occurred and the plate is bent by a low force. Moreover it is observed that as the loading band increases, the ultimate value of the buckling load is also increased.



(a) (b) (c)

Figure-4

The mode shapes of the plate with square cut out under a 15mm loading band and the fixed support. (a) Shape of the first mode (b) Shape of the second mode (c) Shape of the third mode

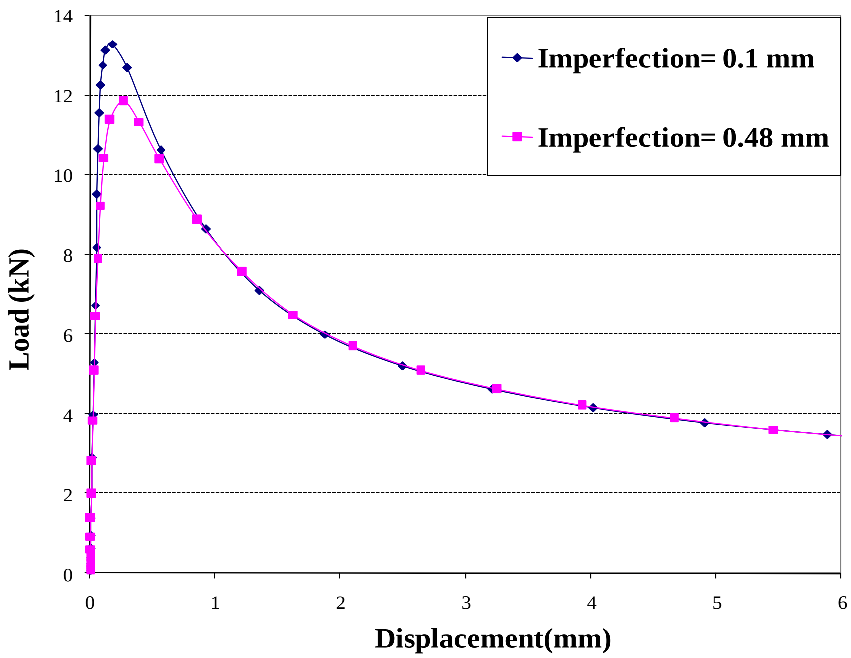


Figure-5

The effect of initial curvature (imperfection) on the ultimate buckling load

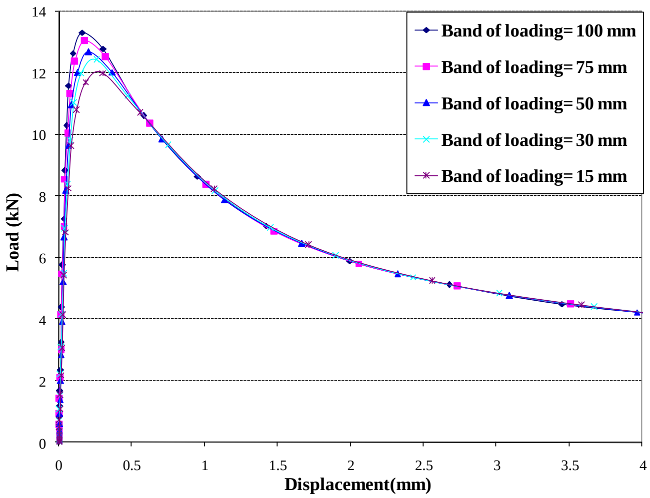


Figure-6
The behavior of load-displacement of the plate with circular cut out

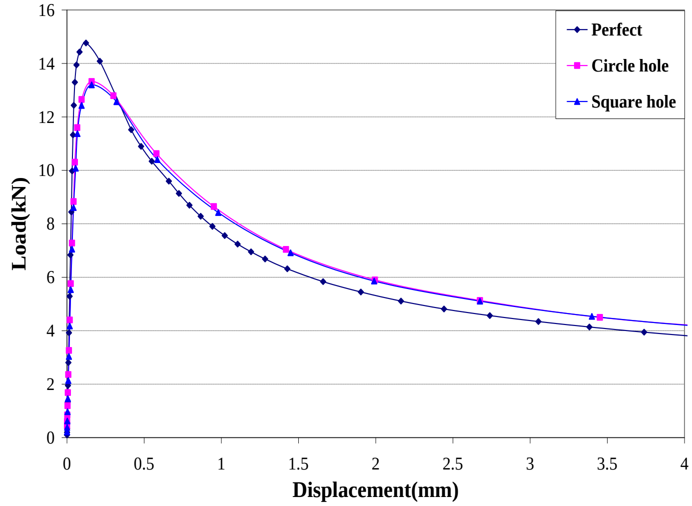


Figure-7
Comparison of load-displacement curves for plates with circular and square cutouts and perfect plate for loading band l=100 mm

Table-1
Results of buckling load for plates with no cutout (perfect) and plates with circular and square cutouts

Critical buckling load with square cutout (kN)	Critical buckling load with circular cutout(kN)	Critical buckling load (kN)	Loading band (mm)
11.851	12.004	12.986	15
12.256	12.424	13.353	30
12.486	12.676	13.762	50
12.821	13.048	14.126	75
13.076	13.372	14.664	100

As indicated in tables (1), for one specific loading band and under the same boundary conditions, the buckling load of the specimen with circular cut out is a little higher than the buckling load of the specimen with square cut out. For the less difficulties of manufacturing and creation of the circular cut out than the square cutout and due to the higher buckling load of the specimen with circular cut out than the specimen with square cut out, it is recommended that for perforated structures which don't have any limitations on type of the cut out's geometry, circular cut out is so better.

Experimental results: Experimental technique for measurement of maximum imperfection of plates: As mentioned above, if we want to make a correct comparison between the numerical and experimental results of the plates buckling, we must use the plates with the same imperfections. Therefore, at first a great number of rectangular plates with the desired dimensions were provided and then their imperfection was experimentally measured. For this purpose, the amount of the preliminary curve obtained through pressing the plate along its thickness and drawing its load-displacement diagram simultaneously. This operation which was performed by an INSTRON servohydraulic machine has been schematically displayed in the figure 8 and figure 9 shows a sample under imperfection's measuring test. δ in figure 8, indicates the amount of the imperfection. A sample of load-displacement diagram for measuring of imperfection has been displayed in figure 10. As it is evident in this figure, where the diagram assumes an asymptotic state, it indicates the amount of imperfection on the horizontal axis which is almost equal to 0.36 mm. Then for prevention of changing of the preliminary bend the boring operations were performed on the plates which had the same preliminary plates by SPARK device.

Experimental measurement of buckling load: The lower edge of the plates was placed in clamped support and their upper edge was placed in a simply support with various loading bands. Also the loading conditions are carried out under displacement control with speed of 0.01 mm/s. for sample, the results of experimental tests which were conducted on the rectangular plates with circular cut outs have been displayed in figure 11. It's obvious that the trend of experimental results is so similar to numerical and is in agree with some other reported FEM analysis[14-16].

Results and Discussion

As shown in figure 11, the trend of the experimental diagrams is wholly similar to the numerical diagrams trend and upon the increase of the band of the loading, the buckling load is also increased. For better understanding of the results, the numerical and experimental quantities of the buckling load are compared in proportion to the boundary conditions and the various loading bands for the specimens having square and circular cut out in tables 2 and 3.

Table-2
The numerical and experimental critical buckling loads for plate with square cut out (average error = 6.55%)

Numerical error (%)	Experimental buckling load (kN)	Numerical buckling load(kN)	Loading band(mm)
12.01	10.584	11.851	15
7.61	11.389	12.256	30
5.84	11.796	12.486	50
0.75	12.918	12.821	75

Table-3
The numerical and experimental critical buckling loads for plate with circular cut out (average error = 9.72%)

Numerical error (%)	Experimental buckling load (kN)	Numerical buckling load(kN)	Loading band(mm)
11.99	10.718	12.004	15
8.14	11.488	12.424	30
6.43	11.920	12.687	50
12.32	14.881	13.048	75

We can suppose that these plates buckle as columns. The critical elastic buckling load equation for columns is:

$$P_{cr} = \frac{\pi^2 E I}{L^2} \quad (1)$$

Where E is the modulus of elasticity, I is moment of inertia and L is the length of column for simply support boundary condition. And for simply-fixed ends boundary condition the critical elastic buckling load equation is:

$$P_{cr} = \frac{\pi^2 E I}{(0.7L)^2} \quad (2)$$

For example for this problem, we have:

$I = bt^3/12 = [100*(2.07)^3/12] = 73.914\ mm^4$, $L = a = 150\ mm$, $E = 218000\ N/mm^2$.

Therefore:

$$P_{cr} = \frac{\pi^2 (218000\,N/mm^2)(73.914\,mm^4)}{(0.7*150mm)^2} = 14410\,N = 14.41\,kN$$

This result is agree with result of figure 7 for load-displacement curve of perfect specimen (critical buckling load, P_{cr}=15 kN). Therefore, we observed that for calculation of nearly critical buckling load of plates with no cutout and full loading band with simply-clamped ends boundary conditions, we can use Euler's buckling formula with good approximation.

Conclusion

We can conclude from the results obtained in this research that: When the buckling phenomenon occurs, the capacity of the load toleration is considerably decreased. The effect of the imperfection in the buckling load is considerable and as comparison between buckling load of two similar plates with different imperfection isn't correct. The initial curvature (the maximum imperfection amplitude) of plate has no effect on post-buckling behavior of plate. The results show that, as loading band increases, the ultimate buckling load also increases. The buckling load of the specimen with circle cut out is a little more than specimens with the square cut out with the equal surface area. Therefore, it is suggested that, if possible, the specimens with circular cut out are used for structures, because the producing of circular cut outs are very easier than square cut outs. Numerical and experimental results are close to each other and their trend is completely the same. We can use Euler's critical buckling formula for calculation of nearly critical buckling load of perfect plates and full loading band with simply-clamped ends boundary conditions.

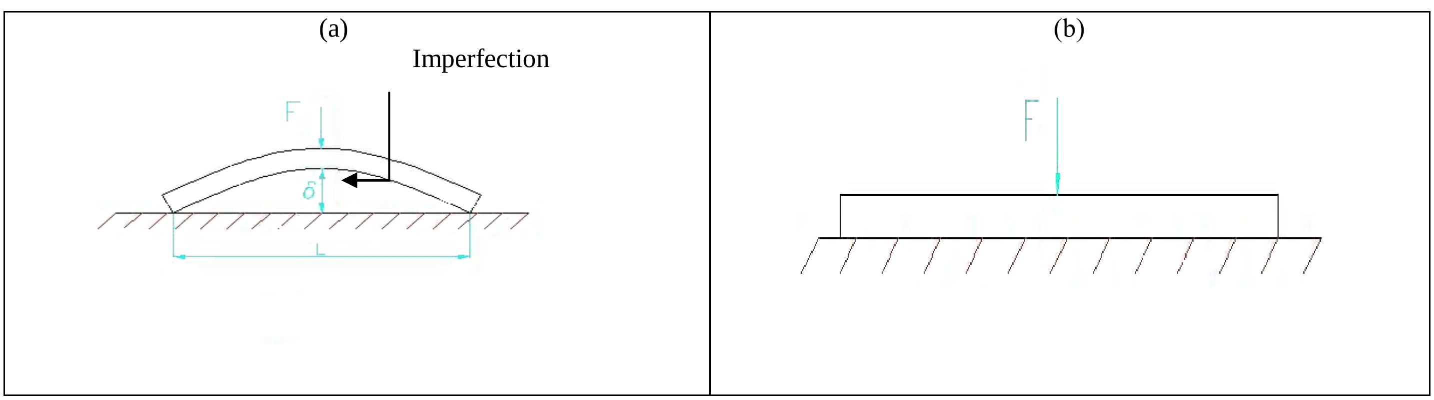


Figure-8
The calculation's procedure of the amount of the preliminary bends through experimental test: (a) Before pressing the plate. (b) After pressing the plate

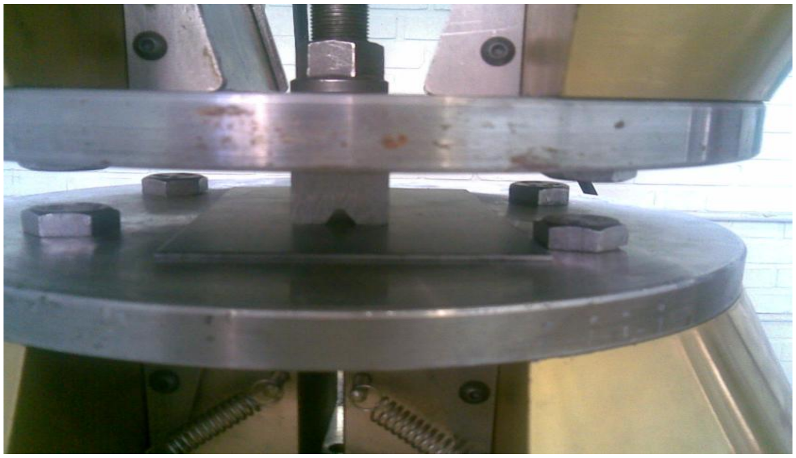


Figure-9
A sample under imperfection's measuring test

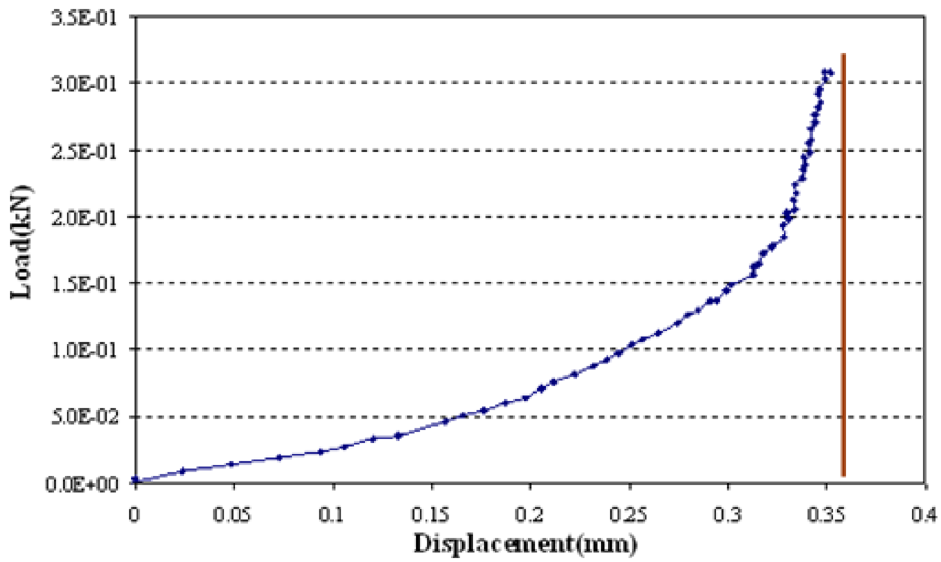


Figure-10
Load-displacement diagram for calculation of the imperfection (δ=0.36 mm)

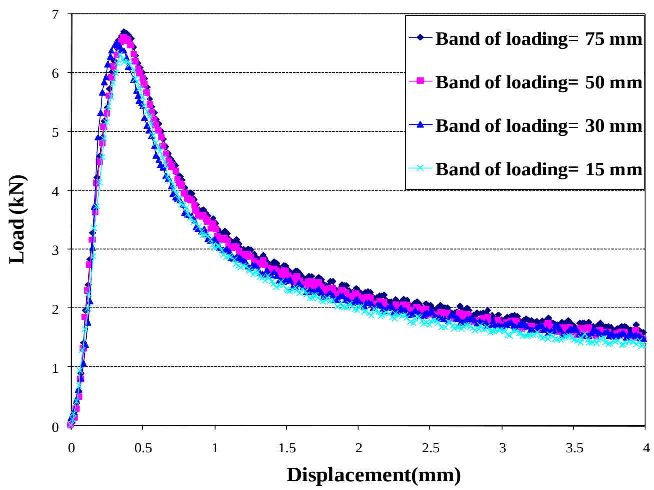


Figure-11
Load-displacement behavior of plates with circular cut out.

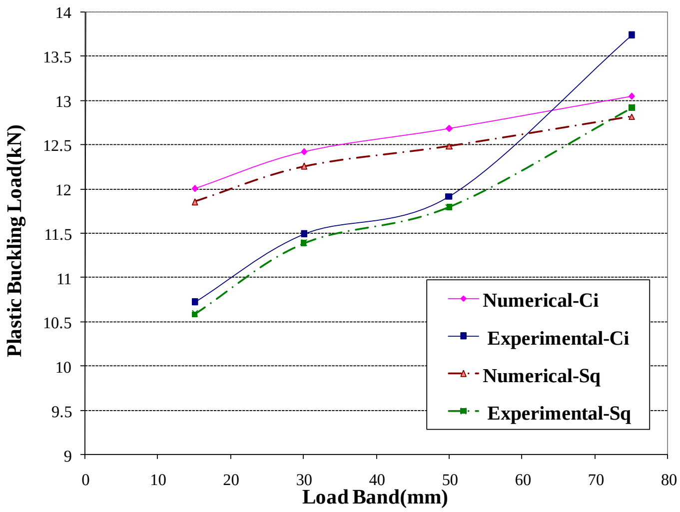


Figure-12
The numerical and experimental comparison of the behavior of the buckling load vs. the loading band for specimen with a circular and square cut out

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