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THE COLLAPSE BEHAVIOR OF THE RECTANGULAR HOLLOW PIPES UNDER COMPRESSION LOAD

Hartono Yudo
Department of Naval Architecture, Diponegoro University, Semarang, 50275, Indonesia.

Wilma Amiruddin
Department of Naval Architecture, Diponegoro University, Semarang, 50275, Indonesia.

Muhammad Harris Nubly
Division of Computer Aided Design and Engineering, PT. Arisma Data Setia, West Jakarta, 11630, Indonesia

ABSTRACT

The rectangular hollow beam used for many constructions. Therefore, the buckling load prediction is neces-sary to convince those constructions would not be overdesign and prevented the failure. Within uses the finite element method, the result is more accurate for the rectangular hollow model. This study has investi-gated the hollows considering the variable of its profile size, length and thickness. The boundary condition used the free-fixed configuration to convince the critical loads occurred from axial compression. The goal of this study is to obtain the collapse behavior of each hollow size also the buckling load factor constant. The mode-shape consisted by wrinkled and Curved shape, it is depending to the rectangular hollow length.

Key words: Rectangular hollow, Collapse behavior, Mode-shape, Wrinkled, Curved shape

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1. INTRODUCTION

The rectangular hollow is the basic component of structures. Several structures used the square and rectangular hollow as alternative component for the stiffener than the other profiles instead. However, the shape of hollow still has the damage problem especially the buckling phenomenon. The structures can be suddenly collapse due to buckle. The square and rectangular hollow mostly used as the foundation of structure, for instance the leg of the oil platform, roof's rib, ship construction, etc. The vertical axis orientations of the structures will give the compression loads to the hollow.

An investigation of elastic buckling of steel column under axial compression who was investigated by Avcar. Where was the fixed-free and pinned-pinned conditions used on the investigation. The length variates from 2.75 to 3.5 meter of circle and rectangle cross section beam. The critical load comparison of Euler equation and FEM calculation which the most lowest difference is the rectangle cross section[1]. The most structural failure due to buckling is depending by its geometry. It is necessary to take the parameter by the cross-section size and non-dimensional slenderness and combination of the pressure[2]. To capture the large range of geometric variable, the value would be an aspect ratio. Shen et al investigated the interactive buckling behavior and the aspect ratio of the length and cross-section variable has been used on these models[3]. The cross section classification is consisted to be four procedure[4]. According this classification, the elastic and plastic ultimate resistance could be distinguished. The 4th classification was shown that the elastic ultimate resistance capacity only for the higher slenderness cross section.

The geometry of the short body would fails as coincidence as its yield strength. The intermediate geometry would fails by the combination of yielding and buckling, however the several stresses were founded beyond the linear range of stress-strain curve. This geometry need the different calculation, there are the arc length method to solve the combination of yielding and buckling. The slender body always fails prior the yield point. Therefore, to predict the slender body still used the linear buckling analysis. The strength parameter of flexural material is depended by the yield strength. However, in the buckling phenomenon the yield strength cannot be a guarantee for the strength parameter, particularly on the slender body geometry. For the instance, the body instability occurred before the strength limit has reached[5].

2. MATERIAL AND LOAD CONDITION

This section described the formulation to obtain the critical load factor for several hollow sections. Also, the model geometries and the boundary conditions will clarify the end condition.

2.1. Buckling Load Prediction

In this case, the eigenvalue extraction was used for estimate the buckling load. The buckling load estimation is obtained as a multiplier of the pattern if perturbation loads, which are added to a set of base state loads. The equation of equilibrium for the configuration during buckling is expressed as,

$$\int_{V^B}^{a} P = \frac{\partial \overline{v}}{\partial X} dV^B = \int_{S^B}^{a} p.\overline{v}\, dS^B + \int_{S^B}^{a} b.\overline{v}\, dV^B \tag{1}$$

where $\overline{v}$ is an arbitrary virtual velocity field, p is the nominal traction on boundary S^B of the body in the base state, b represents the body force per unit volume in the base state and V^B is the volume that the body occupies in the base state. [6]

According Yudo *et al* (2017). the buckling loads are then calculated as part of the second loads step/subcase, by solving an eigenvalue problem[7]

$$\{K + \lambda K_G\}x = 0 \tag{2}$$

where K and K_G are the stiffness matrices, λ is the multiplying factor and x is the eigenvectors (mode shape).

2.2. Buckling Critical Load Formulation

There are the classical methods to predict the buckling loads. Euler's critical load could be quite accurate on the truss. The variation of the structural design demanded a different kind of the structural component. The buckling load prediction of the pipes and the hollows are needed the finite element method. To record the buckling loads behavior for the non-truss body, the loads would be changed to non-dimensional loads. In this case the applied loads from the finite element were compared to the Euler's loads. The buckling load curve consisted of the non-dimensional load and the non-dimensional length of body. The reason to make this dimensionless curve is for the convenience to scale the body geometry.

The free-fixed condition was used in this investigation. This condition expressed by Euler's load equation as[8],

$$P_{cr} = \frac{\pi^2 EI}{L^2} \tag{3}$$

where P_{cr} is the critical load, E is the modulus of elasticity, I is the second moment of area, L is the length.

2.3. Eigenvector Mode Shape

The modes of the linear buckling analysis were provided the shape prediction in the critical load point. In this investigation, to obtained the modes were used the subspace-based eigenvalue extraction method. Where this method used due to smaller amount of eigenvalue. Thus, the Finite Element job process is more faster [9]. The equation as shown in Eq. 2.

According Ádány, the subspace is consisted into four type [10]. Where the axial compressed buckling is categorized with G subspace. Because, compression load would be global buckling. Different with a bending load, which the buckle occurred on partial location.

The eigenvalue not only the single number, however multiple number would be have a different shape along the increasing of its value. According Hearn, the eigenvalue could be combine with the critical load[11] as follows,

$$\boldsymbol{\lambda^2 = \frac{P}{EI}} \tag{4}$$

Thus, obtained with Euler's equation,

$$\boldsymbol{P_n = \frac{\lambda^2 \pi^2 EI}{l^2}} \tag{5}$$

where λ is the eigenvalue (mode number). Actualy the critical value is increasing along the increase of the mode number. It could be described as shown,

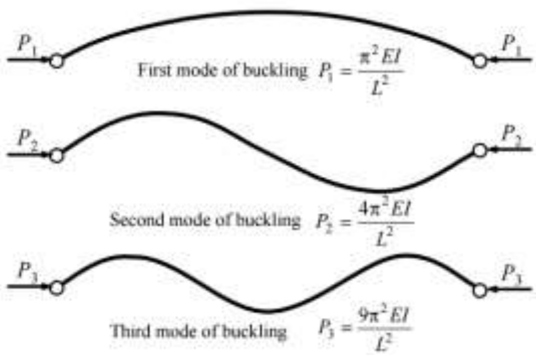


Figure 1: Mode Shape Visualization (Courtesy by Vurg Amit)

2.4. Model Geometry and Material Properties

The models geometry were considering the height and the wide of the hollow, also the thickness and the length. These value were dimensionless, thus the geometries can be scaled. The comparison of the wide per height geometries were symbolized as *a/b* value, the *a/t* is the value of wide per thickness and *l/a* is the value of length per wide. The variable can be shown on the table 1,

Table 1: Variable of rectangular hollow geometries

a/b	a/t
1	10
	20
	30
	40
	50
2	10
	20
	30
	40
	50

where, the *l/a* are in the range of 10 to 70.

Corresponding Euler's formulation of buckling critical load. The material parameter used only the elastic modulus. In this case, steel material has 210 Gpa for it elastic modulus. The eigenvalue equation also shows this parameter. Thus, the elastic modulus is sufficient for calculation to obtain the buckling load factor.

If the displacement vector of eigenvalue equation consisted with reaction force. According Gavin (2006) for a given stiffness matrix, the equation $\{p\}$ = [K] $\{d\}$ will produce a corresponding set of force vectors (in equilibrium). [12]

2.5. Boundary Condition

To define the boundary condition, loading and meshing are the important thing in finite element analysis. Any mistake in the planning of those things would be obtained the different result. These conditions were depending on the situation where the hollows were loaded. In this case the hollows were given free-fixed conditions, where the load points constrained axially by rigid body constrain, however shear stress will be occur. The end condition also gave the rigid constraint. This condition used to obtain the result as the compression load. The end condition constraint shown as,

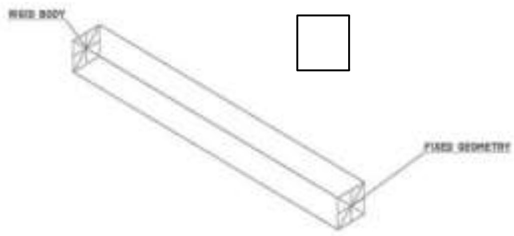
$$\text{Translational} \quad \begin{bmatrix} x \\ y \\ z \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix} \tag{6}$$

$$\text{Rotational} \quad \begin{bmatrix} rx \\ ry \\ rz \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \\ 0 \end{bmatrix} \tag{7}$$

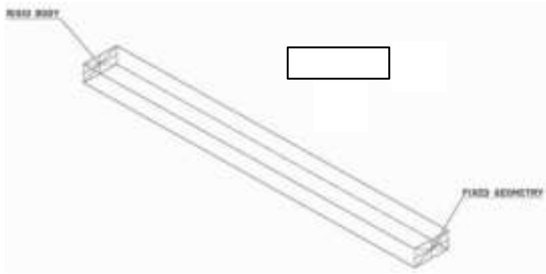
Which the displacement constraint of the load vector shown as,

$$\text{Translational} \quad \begin{bmatrix} x \\ y \\ z \end{bmatrix} = \begin{bmatrix} \infty \\ \infty \\ 1 \end{bmatrix} \tag{8}$$

$$\text{Rotational} \quad \begin{bmatrix} rx \\ ry \\ rz \end{bmatrix} = \begin{bmatrix} \infty \\ \infty \\ \infty \end{bmatrix} \tag{9}$$



(a)End condition of *a/b=1*



(b)End condition of *a/b=2*

Figure 2: Hollow's end condition

3. CALCULATION RESULTS AND DISCUSSION

The aim of buckling analysis is to find the maximum pressure. This pressure is a threshold of collapsed or safe for the structure. Due to the bukle suddenly occurred and sometimes before the yield of material. Therefore, this discussion would be described the behaviour and relationship the buckling conditions by the hollow length and thickness consideration.

3.1. Buckling Critical Load

In the Finite Element analysis, the buckling critical load represented by the eigenvalues. This value could be generated from several modes. But, the first mode is preferred. The behaviour could be described by the relationship curve of pressure-thickness. For the square hollow depicted as follows,

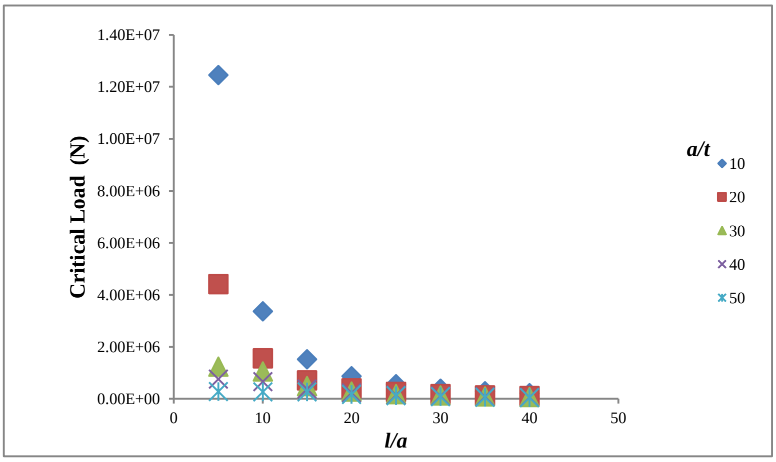


Figure 3: Pressure-Length Relation Curve of *a/b=1*

In this sampe, the 100x100mm square hollow was used to represent *a/b=1*. This curve shown above described that along the increasing of the *l/a*, the critical loads were constant. Thus, the rest of length would be same critical load.

Also, the behaviour was similar for the pressure-thickness curve. The curve would be shown as follows,

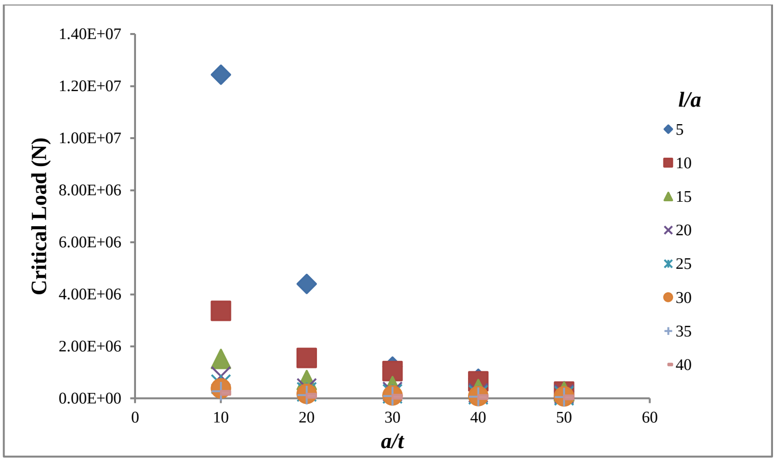


Figure 4: Pressure-Thickness Relation Curve of *a/b=1*

The behaviour of the rectangular hollow similarly described as the square hollow. However, the critical load is smaller than the square hollow. This behaviour due to the rectangular hollow a bit more slender than the square one. The lateral location would be more unstable againts the pressure. It is described that the increasing of the slenderness, the critical load would smaller and lasted to be constant value. The curve of the rectangular hollow as shown by the figure 5 and 6,

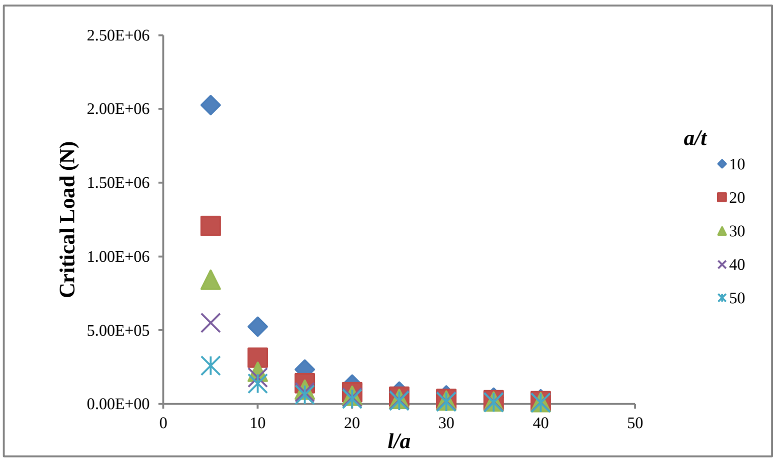


Figure 5: Pressure-Length Relation Curve of *a/b=2*

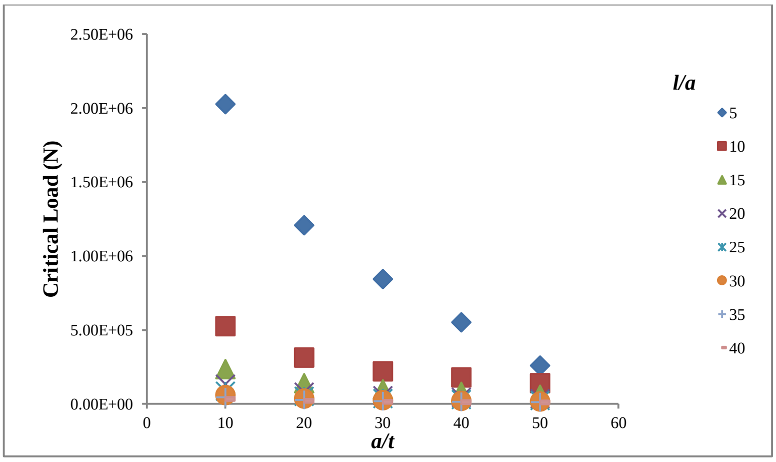


Figure 6: Pressure-Thickness Relation Curve of *a/b=2*

3.2. Load Factor

After performed the linear buckling analysis, the critical load would be obtained from multiplying the eigenvalue. To convinced that the correct load, the eigenvalue must be valued as 1. This eigenvalue were appeared in the post-process interface and calculated with initial load to be buckling critical load. Thus, these critical loads compared with Euler's critical load to make the critical loads to be a non-dimensional value. Then, the loads data were processed to the pressure-length and pressure-thickness relationship, these curves shown in figure 7 and 8,

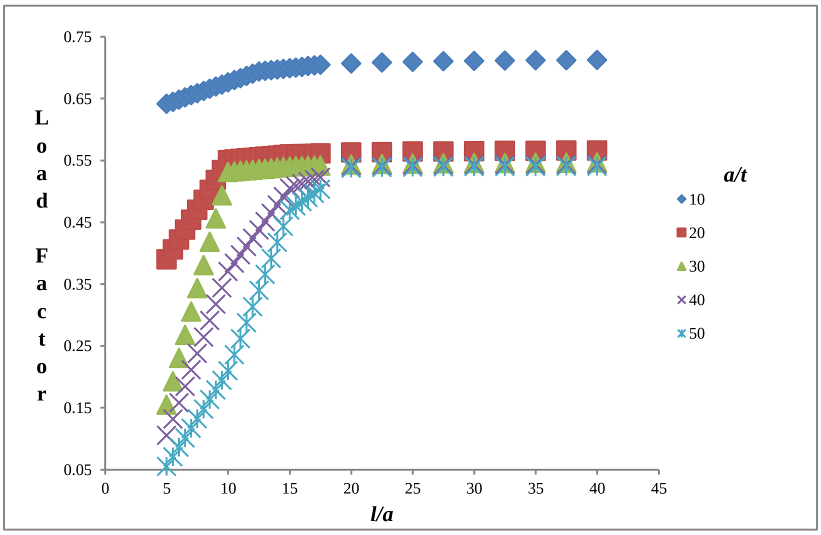


Figure 7: Non-Dimensional Pressure-Length Relation Curve of *a/b=1*

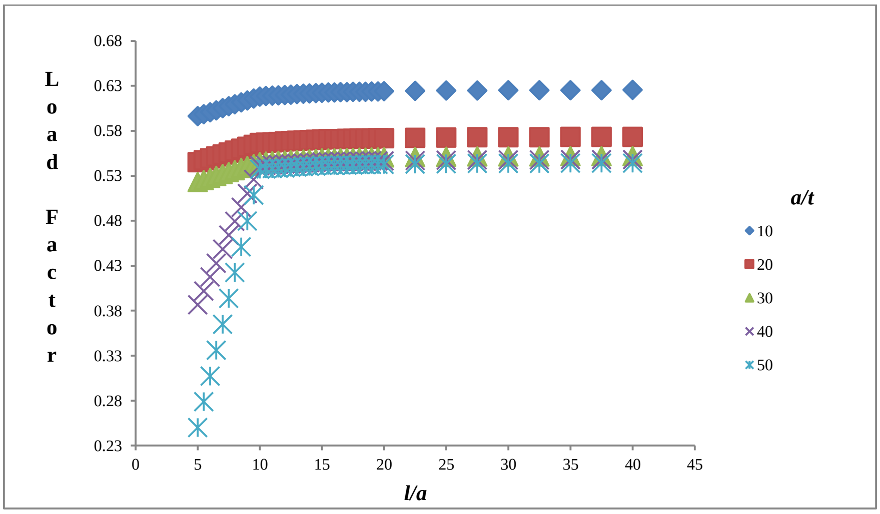


Figure 8: Non-Dimensional Pressure-Length Relation Curve of *a/b=2*

The load factor which is provided by the curve above could be noted as *n* value (in Y axis) per *l/a* (dimensionless length in X axis). Where the *n* value consisted by several *a/t* (dimensionless thickness) of the hollow. This behavior shown that increasingly the length of hollow, the critical loads would be same value. If *n* value combined to Euler's equation, *n* value is inversely proportional. Thus, the actual critical load would be decreased along the increasing of the hollow length. This formulation could be expressed as,

$$P_{cr} = n \frac{\pi^2 EI}{(KL)^2} \quad (4)$$

Where, the *n* value as load factor (in Y axis) per thickness (in X axis) curve. The *K* coefficient is *K=1* if free-fixed of the hollow end condition, *K=0.5* for another end condition. The *n* value represented by the curves as shown,

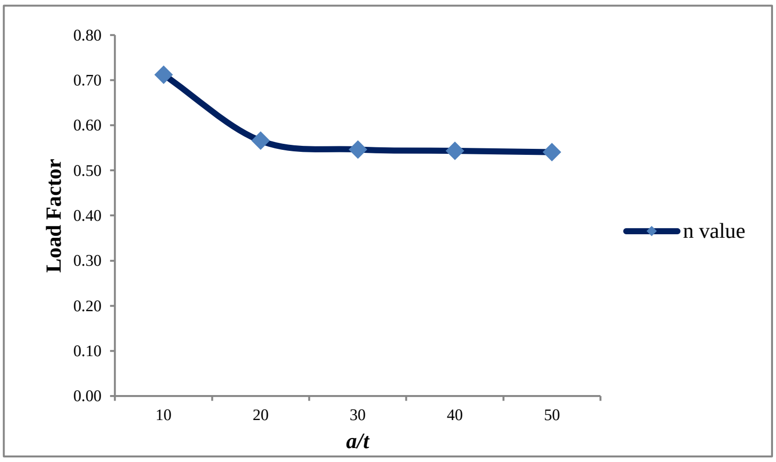


Figure 8: Non-Dimensional Pressure-Thickness Relation Curve of *a/b=1*

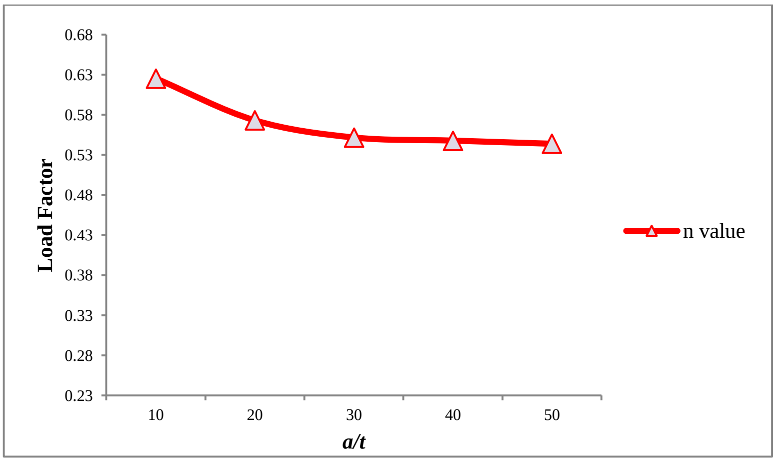


Figure 10: Non-Dimensional Pressure-Thickness Relation Curve of *a/b=2*

The length curves above described that the behaviour of square and rectangular hollow almost similar. Along the increasing of *a/t*, in other words decreasing the thickness. Remember, the *n* value is inversely propotional to the actual critical load. The load factor fell constantly in the square hollow.

3.3. Buckling Mode Shape

Remember that the mode shape is not a real value. The value is a prediction of the hollow shape after passing the critical load. The shape behaviour could be represented by non-dimensional value. The curve shown as follows,

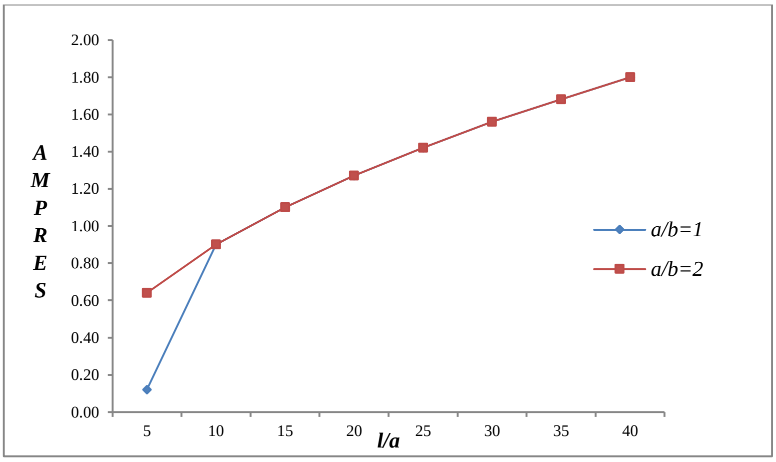
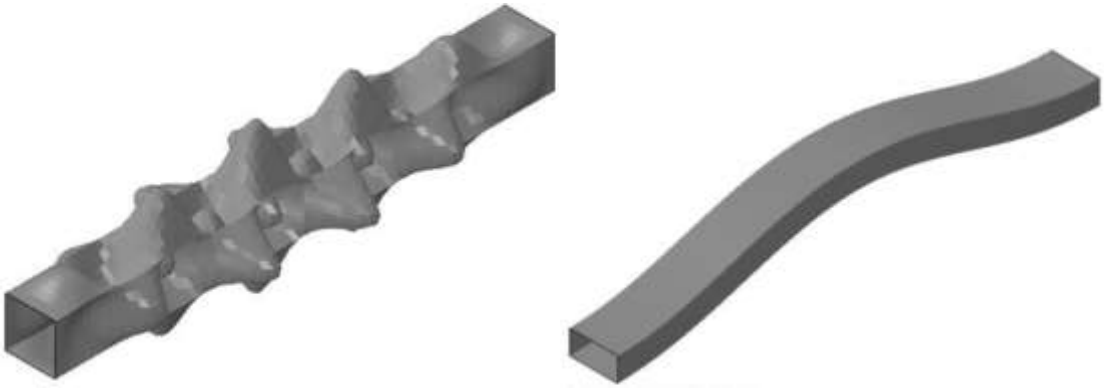


Figure 11: Non-Dimensional curve of Mode Shape

The non-dimensional value represent the buckling shape behaviour. Which, the value is called AMPRES (Amplitude Resultant). According the analysis result, AMPRES < 1 and l/a < 10 shown that the buckle shape was wrinkle. However the AMPRES >1 and l/a > 10 shown that buckle shape was curved.The both shape as shown as Fig 12.



(a)Wrinkled shape

(b) Curved shape

Figure 12: Two different shape behavior of the hollow

4. CONCLUSION

The calculation of square and rectangular hollow section, to find a buckling critical load has been finished by compared the Euler's formulation and Finite Element analysis. These critical load result has been formed in the non-dimensional curve. This value is appropriate for the model geometry variable which formed in an aspect ratio. The non-dimensional curves would be convenienced to calculate an elastic buckling and its has a large number for the geometric variable. The geometric parameters is consisted by $a/b=1$ to 2 for the width and height ratio, $a/t=10$ to 50 for the thickness ratio, $l/a=5$ to 40 for the length ratio. The buckling behaviour of these hollow pipe could be concluded as follows,

1. The non-dimensional critical load which called n value has been captured by the linearity constant of the square and rectangular hollow pipes. For both hollow pipes, $n=0.54$. For more detail, the non-dimensional curve provided the n value for the short body which is above of $n=0.54$.
2. This investigation based on Euler's equation. Euler's critical load provided the material only for its elastic modulus. The new equation is valid for the material

which has *E=210* GPa. Its can be concluded that along the increase of the hollow pipes geometry, the buckling critical load also increasing. Meanwhile, the yield parameter could not included for this formulation. Therefore, the hollow pipes load should a pure compression. For the instances, the thermal load could not be a parameter.

3. The non-dimensional pressure-thickness curves are shown that along increasing of slenderness, the hollow pipes would shown an elastic behaviour. Which, the critical load above *a/t=30* would be similar.
4. Also the non-dimensional pressure-length curves are shown that along increasing the length of hollow pipes, the critical load would be decreasing and at *l/a=20* the value is constant for *a/b=1,* also *l/a=10* for *a/b=2.* This behaviour occurred due to the geometry of *a/b=2* is more slender than *a/b=1*, which the *a/b=1* has a higher value of the buckling critical load.
5. The end condition has a different buckling critical value each other. The sample of this investigation used the free-fixed condition. For the pinned-pinned and fixed-fixed end conditions has diverged in the half amount of the free-fixed condition. Therefore, *K* value is added for the new equation parameter. Which, *K=1* for the free-fixed and *K=0.5* for the fixed-fixed and pinned-pinned condition.

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