



Effect of Pad Thickness and Pad Diameter Variations on Plastic Limit Moment in Cylindrical Pressure Vessels Due to Nozzle Torsion Load

Petrus Lioe¹, Indraswari Kusumaningtyas¹, and Rachmat Sriwijaya^{1,*}

¹ Department of Mechanical and Industrial Engineering, Faculty of Engineering, Universitas Gadjah Mada, Grafika Street No.2, Yogyakarta 55281, Yogyakarta

Corresponding author's email: sriwijaya@gadjahmada.edu

Abstract. During its operating life, pressure vessels can experience various types of internal and external loads, which cause stress concentration on the nozzles. One of the load types is torsional load, which occurs due to a combination of internal pressure and thermal load due to fluid temperature inside the pipe. The moment occurs because the load shall not exceed the plastic limit moment of the nozzle so that failure does not occur. For this reason, it is critical to study the limits of moments that arise in nozzles. This paper aims to determine the plastic limit moment in cylindrical pressure vessels due to torsion load with various pad thicknesses and diameters. The simulation was conducted using a non-linear finite element analysis method using ANSYS Workbench R2 Static Structural, following the modeling approaches from previous studies. The load-deformation curve based on the Twice Elastic Slope (TES) theory obtains the plastic limit moment for each variation simulation. The results indicate that the plastic limit moment (MpL) due to torsion load will increase by increasing the pad thickness until there is a certain optimum pad thickness, which the plastic limit moment (MpL) will not increase by increasing the pad thickness. In addition, the plastic limit moment (MpL) and deformation limit under torsion are lower than those observed for out-of-plane and in-plane moments. The result of this study is beneficial in increasing knowledge about the effect of pad thickness and diameter on the plastic limit moment of pressure vessels. It can be developed further to optimize the design of cylindrical pressure vessels based on ASME Section VIII Division 2.

Keywords: Cylindrical pressure vessel, Plastic limit moment, Displacement limit, Torsional loading, ANSYS simulation.

1 INTRODUCTION

Pressure vessels are widely used in various sectors, such as oil and gas processing, nuclear and chemical industries, petrochemicals, power plants, etc. A pressure vessel is a closed container that holds a fluid, either gas or liquid, higher or lower than the ambient pressure. During its operating life, pressure vessels can experience various internal and external loads. Internal loads result from the pressure and temperature of the working fluid, while external loads can include dead loads (due to the vessel's weight), wind loads, earthquake loads, and loads on nozzles connected to the piping system.

All these loads can act simultaneously, creating a combination of stresses. Therefore, this combination of loads must be considered in the design of the pressure vessel to ensure a safe design that can operate throughout its service life. If the pressure vessel experiences an overload that causes actual stresses to exceed the maximum allowable limit, catastrophic structural damage may occur. Such damage can compromise worker safety and cause environmental pollution. To prevent these risks, the pressure vessel must be designed to withstand all applicable internal and external loads.

The standard used in designing pressure vessels is called a code. The standard code used to design pressure vessels is ASME Section VIII, Divisions 1 and 2. Each pressure vessel component is designed using specific formulas or rules in this code. Pressure vessel components consist of pressurized and non-pressurized parts. Pressurized components include the shell, head, and nozzle, while non-pressurized components include the lug, skirt, and saddle. The shell is a part of the pressure vessel shaped like a hollow cylinder or sphere. The nozzle functions to connect the pressure vessel to the piping system. With the combination of internal pressure on the shell, the joint area between the shell and the nozzle becomes critical, experiencing the most significant combined stress.

ASME Section VIII, Division 1, is based on the design-by-rules method, while Division 2 uses the design-by-analysis approach. Design by rules provides general guidelines for loading within certain limits, whereas design by analysis is used for designing non-standard pressure vessels, such as those with multiple nozzles, non-cylindrical nozzle shapes, or asymmetrical nozzle locations. The design-by-rules method is the most widely used in practice due to its practicality and conservative approach, which involves following general guidelines without detailed analysis. However, this often results in a more expensive pressure vessel design. On the other hand, design by analysis requires more detailed calculations but can yield a more economical design. The fundamental difference between design by rules and design by analysis lies in the failure theories used. Design by rules follows the standard stress theory, while design by analysis employs the Von Mises theory.

ASME Section VIII, Division 2 using a design by analysis utilized limit load analysis, specifically the experimental determination of plastic limit loads found in Appendix 6. Mackenzie [3], in his paper, stated that the determination of plastic limit loads is known as the Twice-Elastic-Slope (TES) method or the Double Elastic Slope Method. Prueter et al. [6] compared ASME Section VIII, Division 2 before 2007 with the latest edition and concluded that TES aligns with the elastic-plastic method used in the current ASME Section VIII, Division 2. Furthermore, Prueter et al. [6] highlighted that the advantage of using TES is the consistency in the design margin, where the maximum design load is set at two-thirds of the plastic limit load. This underscores the importance of determining the plastic limit load in the pressure vessel design process.

During the operation of a pressure vessel, various loads occur on the nozzle. These loads can be axial, moment, and torsional. Axial loads can arise from the pipe's thermal expansion due to the fluid's temperature. Moment loads may occur due to the weight of the pipe, flange, and valve connected to it. Torsional loads can result during operational conditions due to the combination of internal pressure and thermal load due to the fluid temperature inside the pipe.

As a result of these load combinations, a combination of stresses affects various pressure vessel components. Among these components, the junction area between the

nozzle and the shell is considered a weak point because the hole in the shell creates a high stress concentration in this area. To reduce stress in critical areas such as the junction of the nozzle and shell, the esteemed ASME Section VIII provides a reliable solution for the installation of pads. Pads, metal plates attached to the junction area, are a proven method to alleviate stress, providing a sense of reassurance and confidence in the structural integrity of the pressure vessel.

Mackenzie [3], in his research, used a finite element method approach using a cantilever bar given a bending moment load. Mackenzie showed in his study that out-of-plane bending moments provide lower plastic limit loads than in-plane bending moments. Sang et al. [7] conducted experimental research and finite element methods with a nozzle given out of plane bending moment load. The study results showed that the plastic limit load was calculated using the finite element method on the pressure vessel using the experimental method. Sang et al. [7] also showed that pad, shell, and nozzle size variations affect their plastic limit loads.

Sang et al. [8] conducted experimental research and finite element methods on a pressure vessel with a nozzle given an in-plane bending moment load. The study's results showed that the plastic limit load was calculated using the finite element method on the pressure vessel using the experimental method. Ginting [2] conducted a study on determining the plastic limit load on pressure vessels with various sizes and thicknesses of shells, nozzles, and pads, applying out-of-plane bending moment loads on the nozzle by the research by Sang et al. [7]. Ginting [2] concluded that cylindrical pressure vessels' plastic limit moment increases with pad thickness and diameter. The plastic limit moment (MpL) reaches its maximum at an optimal pad thickness and diameter, beyond which further increases in pad thickness or diameter do not affect the plastic limit moment. Additionally, increasing shell thickness (T) and nozzle diameter (d_i) in cylindrical pressure vessels leads to a more excellent plastic limit moment.

Nuridin [4] conducted a study to determine the plastic limit load on pressure vessels with various sizes and thicknesses of shells, nozzles, and pads. The study, which applied in-plane bending moment loads on the nozzle, following the research by Sang et al. [7], found that the plastic limit moment (MpL) also reaches its maximum at an optimal pad thickness and diameter. Beyond this point, increasing these parameters no longer affects the plastic limit moment. Notably, the study revealed that the plastic limit moment of pressure vessels with an unpadded nozzle is greater under in-plane moment loading compared to out-of-plane moment loading, providing a clear contrast between the two loading scenarios.

Based on those studies, it is concluded that the type of loads will affect the magnitude of the actual stress that occurs. Until now, there has been no research that links the type of load with the magnitude of the plastic limit moment due to torsion load. Therefore, research is needed to prove this. In this study, a finite element analysis (FEA) method approach will be used to examine the effect of torsion load on the specific geometry of the pad on a pressure vessel. The analysis is carried out on various types of pad geometry to observe the effect of each geometry on its plastic limit moment due to torsion load.

The limitations of the problems to be analyzed, along with the assumptions to be used, are as follows. Determining the plastic limit load uses the double elastic slope method according to ASME Section VIII Div 2. The shape of the pressure vessel model is by the experimental construction of Sang et al. [8]. This study does not use an

analytical method with a mathematical formula to determine the magnitude of the plastic limit moment in a cylindrical pressure vessel. The effect of earthquake loads and internal loads due to fluids are not considered in this study; only the loading due to torsional at the nozzle of the cylindrical pressure vessel is considered.

In determining the plastic limit moment, this study does not consider the effect of other parts of the cylindrical pressure vessel, such as saddles, heads, lugs, and skirts.

The aim of this study is to determine the plastic limit moment in cylindrical pressure vessels due to torsion load with various pad geometry and compare the plastic limit moment due to various loads such as torsion, in-plane moment, and out-of-plane moment.

2 RESEARCH METHOD

In this study, simulations were carried out to determine the plastic limit moment due to external torsion load on the nozzle with the joint construction between the shell and nozzle reinforced with a pad. This study uses a finite element analysis method, using Autodesk Inventor software to create a 3D pressure vessel and ANSYS software to perform the simulations. According to Prueter [6], one way to determine the plastic limit load is to use a twice-elastic slope, as illustrated in Figure 1. The double elastic slope is determined by drawing a regression line from the origin following the elastic line in the load-strain curve. A straight line of double elastic slope is drawn from the origin intersecting the load-strain curve in the plastic region with the following conditions: if the angle of the regression line to the load axis is θ , then the angle of the straight line of the double slope is ϕ , with the condition $\tan \phi = 2 \tan \theta$ satisfied.

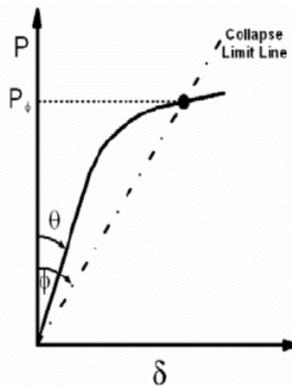


Fig. 1. Double Elastic Slope (Prueter, 2015)

The dimensions of the pressure vessel are shown in Figure 2 and Table 1 in the model.

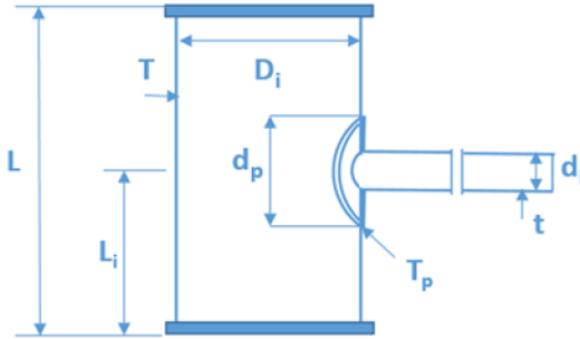


Fig. 2. Pressure Vessel Dimension [2]

Table 1. Pressure Vessel Dimension

Parameter	Dimension
Vessel Length (L)	1000 mm
Vessel Diameter (Di)	500 mm
Vessel Thickness (T)	8 mm
Distance from center line nozzle to the bottom of the vessel (Li)	500 mm
Nozzle Length (l)	1000 mm
Nozzle Diameter (di)	86 mm
Nozzle Thickness (t)	3 mm
Pad Diameter (dp)	160 mm
Pad Thickness (Tp)	10 mm

The shell and pad are made of Q235-A steel, which is a low-carbon steel compatible with A36-77 steel. The nozzle is made of Steel 20, which is a low-carbon steel compatible with A106-80 Grade A steel. Table 2 shows the chemical composition and mechanical properties of the pressure vessel material.

Table 2. Chemical Composition and Mechanical Properties of Material (Sang, 2006)

Part	Material	% Chem. Composition					Tensile Test				
		C	Si	Mn	P	S	S _u (MPa)	S _y (MPa)	δ ₅ (%)	E (GPa)	μ
Shell	Q235-A	0.19	0.22	0.51	0.029	0.01	448	352	23.1	202	0.3
Pad											
Nozzle	Steel 20	0.19	0.27	0.46	0.019	0.017	479	319	33.9	212	0.3

The welding method between the shell and nozzle uses butt weld, while pad welding on the shell is done by fillet welding method. Welding geometry is made to obtain simulation results that are close to the experimental pressure vessel because the welding position is in the critical area of the observed pressure vessel. Details of the welding geometry are shown in Figure 3.

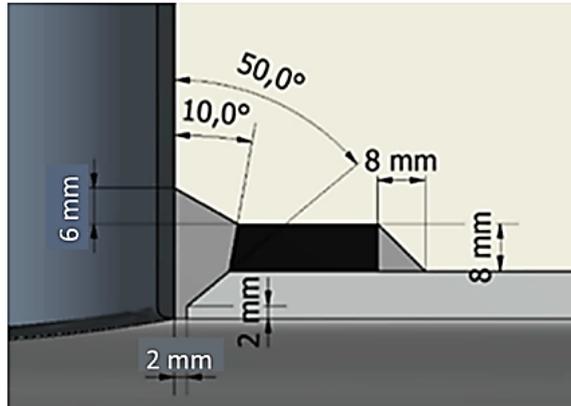


Fig. 3. Detail Welding Configuration Between Shell and Nozzle

Finite element analysis simulation was performed using ANSYS software. The 3D model that was created in Autodesk Inventor was then imported into ANSYS software. After modeling and entering material parameters, the next step is to perform meshing. For specific areas to be observed, for example around the shell and nozzle junction, a smaller mesh is used than other areas. This method to obtain calculations with higher accuracy. The results of the model meshing in ANSYS can be seen in Figure 4.

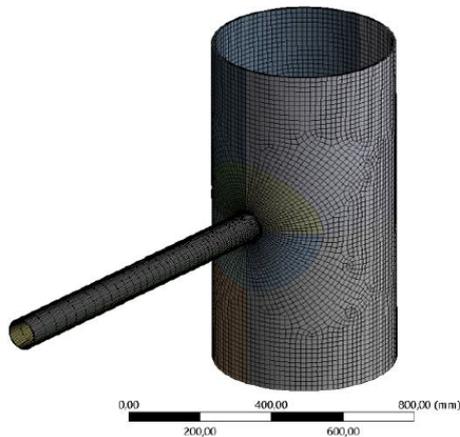


Fig. 4. Meshing of Pressure Vessel Model

The boundary conditions used in the simulation were made the same as those in the Sang et al. [7] experiment, which is fix for the lower side of pressure vessel and free for the upper side of pressure vessel. The connection between the nozzle and the shell is connected as shown in Figure 5. External loading is done at the nozzle tip by means of a load that is gradually increased (incremental). The loads are torsional loads. ANSYS software will calculate the deformation due to external gradual loading.

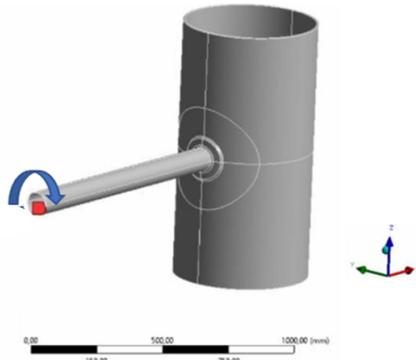


Fig. 5. Pressure Vessel Loading

3 RESULT

The finite element simulation results will provide stress (σ), deformation (δ), and the plastic moment (load) (Mp) at the location of the points shown in Figure 6 for the distribution of the plastic area in the pressure vessel due to load.

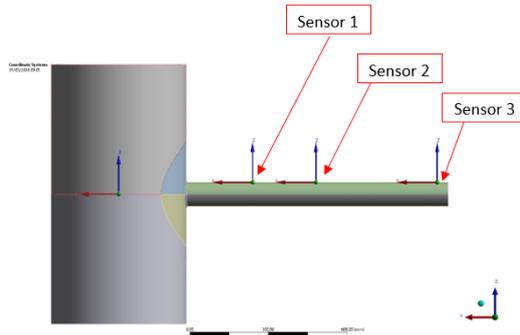


Fig. 6. Measurement point location at nozzle

For model validation, a pressure vessel simulation is developed using the same model as Sang et al. [8] with shell thickness (T) of 500 mm, pad thickness (t_p) of 0 mm, pad diameter (d_p) of 0 mm, nozzle thickness (t) of 3 mm, and nozzle inner diameter (d_i) of 86 mm. Validation can be used by comparing the plastic limit moment. To determine the moment (load) of the plastic limit, the double elastic slope method is used, as described above. The simulation results get a moment and deformation curves, and then a regression line is drawn from the original point following the elastic line at the moment–deformation curve and the double elastic slope line from the original point intersects the moment–deformation curve in the plastic region, then Figure 7 is obtained as shown below. Figure 7 shows only Sensor 3 as a reference.

The difference in plastic limit moment from the experiment of Sang et al. [8] and the simulation of the research results can be seen in Table 3. The average deviation

between the simulation results and the Sang et al. [8] experiment was 6%, so it can be said that the model created is considered valid with an average deviation and individual deviation still below 10%. They are furthermore, considering the acceptable design limit of ASME Section VIII, such as two-thirds of the plastic load limit, such deviation below 10% is considered sufficient.

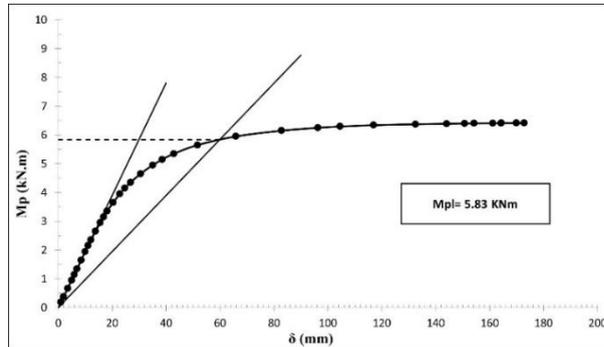


Fig. 7. Plastic Moment vs Deformation Curve for Validation Model

Table 3. Plastic Limit Moment Sang et al. [3] vs Simulation

Reference Data	Plastic Limit Moment / MpL (kN.m)		
	Sensor 1	Sensor 2	Sensor 3
Experiment (Sang et al., 2010) [8]	5.96	5.77	5,5
Simulation Data	5.4	5.6	5.83
Deviation	0.56	0.17	0.33
(Experiment – Simulation Data)	(9%)	(3%)	(6%)
Average Deviation		6%	

After validation of the model, various pad thicknesses (t_p) and diameters (d_p) are added to the nozzle to follow the dimensions from Ginting [2] and Nurdin [4]. The simulation begins with step-by-step increasing torsion load as per Figure 5 until the stress at the nozzle passes its yield point and enters the inelastic area. It requires seven kNm torsion to make the nozzle enter the inelastic area. Then, the yield stress is obtained in the nozzle area and around the nozzle and shell branches, as shown in Figure 8, which represents the stress contour due to loading of 7 kN.m torsion at specific pressure vessel and nozzle geometry. Figure 8 shows the maximum stress is 344.64 MPa and indicates that the plastic deformation starts from the nozzle and moves to the pad and shell around the junction between the nozzle and the shell. The strain due to torsion loading is shown in Figure 9, and the maximum elastic strain that occurs at the same location is 0.0016257 mm/mm.

T : 8 mm, di : 86 mm, dp : 160 mm

D: A1-3
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 8

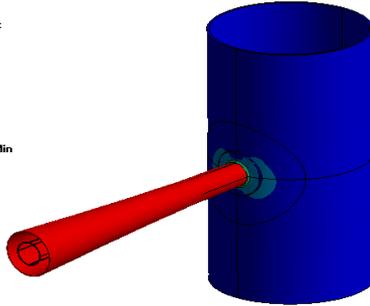
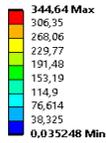


Fig. 8. Maximum Stress (Von mises)

T : 8 mm, di : 86 mm, dp : 160 mm

Equivalent Elastic Strain
Type: Equivalent Elastic Strain
Unit: mm/mm
Time: 8

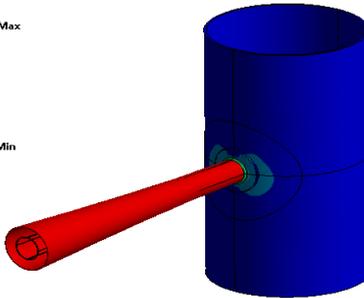
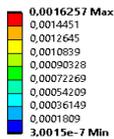


Fig. 9. Equivalent Elastic Strain

The result of the plastic limit moment due to torsion load with pad thickness variation is shown in Figure 10. It is shown that the plastic limit moment increases slightly with the thicker pad until certain points at 4 mm, which, in addition to pad thickness, does not significantly increase the plastic limit moment. Comparison with the plastic limit moment due to out-of-plane moment [2] and the in-plane moment [4] due to pad thickness variation is shown in Figure 11. The plastic limit moment due to torsion is lower than the plastic limit moment due to out-of-plane and in-plane moments. The result of the plastic limit moment due to torsion load with pad diameter variation is shown in Figure 12. It is shown that the plastic limit moment is unchanged with the pad diameter variation.

Figure 13 compares the deformation at the Plastic limit moment due to torsion, the out-of-plane moment [2], and the in-plane moment [5]. It observes that deformation due to torsion is far below the deformation due to out-of-plane and in-plane moments. This is because the nature of deformation due to torsion is more in the form of twisting than the nature of deformation due to moment loads, which is in the form of axial deformation.

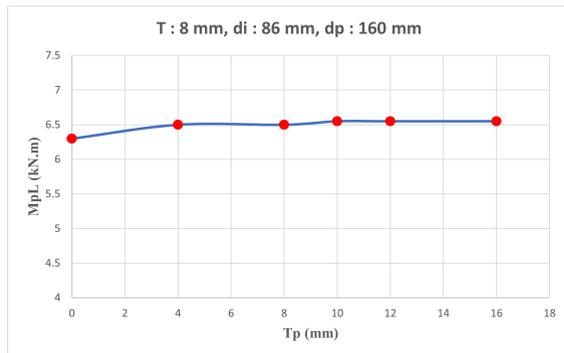


Fig. 10. Plastic Limit Moment for Various Pad Thickness

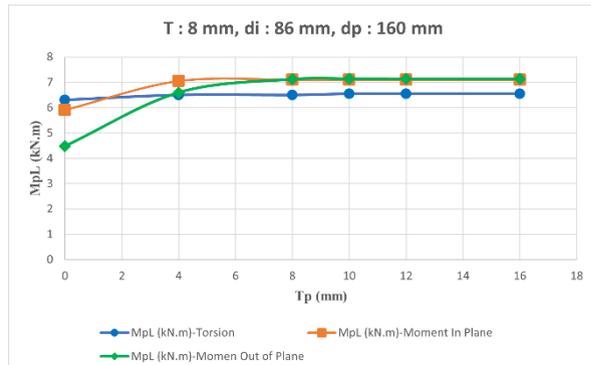


Fig. 11. Comparison of Plastic Limit Moment for Various Pad Thickness

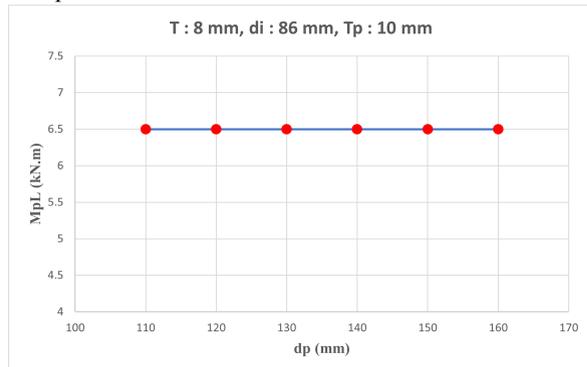


Fig. 12. Plastic Limit Moment for Various Pad Diameters

4 CONCLUSION

The conclusions obtained from this research are as follows:

1. The cylindrical pressure vessel's plastic limit moment (MpL) will increase by increasing the pad thickness until a certain optimum thickness, whereas the plastic limit moment (MpL) will not increase by increasing the pad thickness.
2. The plastic limit moment (MpL) due to torsion for a padded nozzle is lower than the plastic limit moment (MpL) due to the moment out of the plane and moment in the plane. This means the pressure vessel-nozzle configuration is weaker against torsion load compared to a moment in and out of the plane.
3. The plastic limit moment of the pressure vessel-nozzle configuration is not sensitive to pad diameter (dp).
4. The deformation of the nozzle at the plastic limit moment (MpL) value due to torsion load is shallow compared to the deformation of the nozzle at the plastic limit moment (MpL) value due to the moment out of plane and the moment in plane.

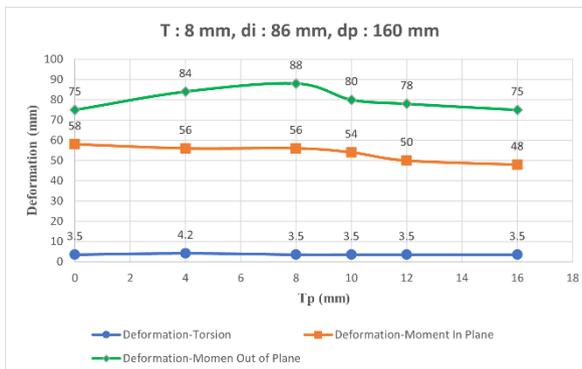


Fig. 13. Comparison of Deformation at Plastic Limit Moment for Various Pad Thickness

Acknowledgements. The authors thank the Laboratory of Dynamics Universitas Gadjah Mada for the facility for doing this research.

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