Influence Of Internal Pressure To The Pipe Resistance Due To Impact Loading

IGN Wiratmaja Puja, Nguon Kollika
Engineering Design Centre, Dept. of Mechanical Engineering
Bandung Institute of Technology, Jalan Ganesha 10 Bandung, 40132
Email: iwpuja00@edc.ms.itb.ac.id, nguon@edc.ms.itb.ac.id

Abstract

This paper presents quasi-static analysis on the response of the pressurized in-service pipelines due to external impact loading. There is a high possibility that pipeline is damaged by accidental load caused by third parties such as excavator tooth for onshore pipelines or anchors trawl gears for subsea pipelines. The analysis compares two cases of damages; the empty and the pressurized bore mild steel seamless pipe in which the geometry and internal pressure of pipe, indenting depth, indenting impact force, shape of indenter, etc are the main variables. The results revealed that internal pressure has a significant influence on the pipe resistance to the deformation subjected to impact force; the response of the internal pressure is absolutely necessary to reduce the dent depth as well as the level of damage on the pipe structure like what we have expected. They have been confirmed to be acceptably correct by comparing to those obtained from the numerical analysis (Ls-Dyna), which lead to a common conclusion in which in-service pipelines of the same geometry require higher pressure to resist a higher indenting impact force.

Keywords: Pipes, impact force, indenting depth, pressure, local deformation.

Introduction

Pipelines are widely used throughout industry to transport a variety of potentially hazardous materials. The contents transported such as gases liquids under high pressure, may be released into environment by an impact event, which is sufficiently severe to cause material failure. All of the risks for pipelines, the impact loading is a significant factor that often comes across on pipelines damages. Due to the risk assessments of this case, it is highly possible that onshore pipelines are impacted accidentally by excavator tooth. Whereas, offshore pipelines may be indented by anchors, trawl gears or other hard objects dropped from the ships. Furthermore, the local dents caused by collision of adjacent risers may threaten the integrity of deep water floating production systems. These obviously show that the accidental loads caused by third party activities pose a serious treat to pipelines integrity. In order to avoid this kind of treat as well as to reduce the level of damages, the design and operation for safety should be applied in the operation of pipeline system. Therefore, it is utmost important that pipeline engineers are able to assess the significance of damage due to external interference.

According to some references, the theoretical and experimental studies on the risk assessments on pipeline due to impact loading have been conducted for many years. Further research should be carried out to investigate and provide more understanding on impact behaviors, which lead to more effective solutions that can ensure pipeline safety. Since 1973, many analyses have been conducted on pipes and rings to simulate the pipe indentation with combinations of dent depth or gouge. In most of these tests, the pipes and rings were first indented or perforated without any internal or external pressure. Then, they were pressurized to failure. Some regression and semi-empirical equations were derived from these tests for predicting the pipe resistance to impact loading.

Table I. Nomenclature

R	radius of pipe
r	average radius of the deformed arc of the ring
l	length of the indenter and of indent deformation zone
h	thickness of pipe shell
δ	displacement for ring in the deformation regions outside the indent zone ($\bar{\delta} = \delta / \boldsymbol{R}$)
δ_0	displacement for ring in the indent zone, maximum dent depth; ($\bar{\delta}_0 = \delta_0 / \boldsymbol{R}$).
ε	tensile stain
θ	angle corresponding to the deformed arc for ring in the deformation region outside the $\bar{\delta}_0$ indent zone
θ_0	angle corresponding to the deformed arc for ring in the indent zone
α	$= 4\theta$
ξ	length of outside deformation region
β	angle corresponding to point P
χ	rotation angle of generators
F	external impact force
F_c	ring crushing force
u, w	horizontal and vertical displacement
p_{in}	internal pressure
p_{out}	external pressure
p	$= p_{in} - p_{out}$
M_0	$= \sigma_y h^2/4$ fully plastic moment
σ_y	yield stress
N	$= \sigma_y . h$ fully plastic axial force
t	time
E_{crush}	dissipated energy in the rings
E_{gen}	dissipated energy in generator
E_p	resistant work done by pressure
ω	angle corresponding to arc P'B
ω_0	vertical displacement corresponding to the angle at initial point of the ring
x, y, z	coordinates
P	point selected on the curvature before indentation
P'	point of displacement of the portion of curvature.
B	point at middle of indentation, where a plastic hinge occurs
A&C	points on curvature of the ring where the plastic hinges occur.

A major difficulty in applying numerical methods to elastic-plastic structures is that the plastic strains are often much larger than the elastic ones. To bypass the potential difficulties associated with such a disparity, DeRuntz and Hodge [4] assumed rigid-perfectly plasticity and derived a formula for the load-deflection curve that occurs after 4 plastic hinges have been formed. Redwood refined the model of DeRuntz and Hodge [4] by adding linear strain-hardening. Reid and Reddy [5] refined these models by assuming rigid linearly hardening moment-curvature relation within a small region near horizontal, diametrically opposed points where the bending moment is the greatest and where the deformed curvature is high. They ignored unloading and assumed that the ring moved as rigid body in the portion of the ring between these plastic zones and the points of contact to the platens. However, it should be noticed that elastic effects may contribute significantly to the deflections of the ring depending on the load, the relative thickness, and the material properties.

The aim of the present paper is to analyze the pipe and ring deformation and the response of internally pressurized pipes subjected to lateral quasi-static denting loads [Fig. 1]. The pipes under consideration exhibit significant inelastic deformations and the lateral load is imposed by a wedge-shaped denting device assumed to be very small by comparing to the large pipe diameter. The discussions were made to compare the results, which revealed a good agreement between the theoretical and numerical analysis (Ls-Dyna).

2. Theoretical analysis
2.1. General information

Deformations of the pipeline structure influence not only the integrity and longevity of operations in this field but also the economic aspects. It should be reminded that pipelines have been mostly used to transmit and distribute potentially hazardous materials such as gases, oil and other liquids. Due to risk assessment, million of dollars may be spent on the environment as well as reparations of the failure pipes. It is therefore important that many analyses have been conducted to reduce and avoid these problems. The theoretical analysis in this paper is carried out to provide more understanding in the impact characteristics of the pipe structure.

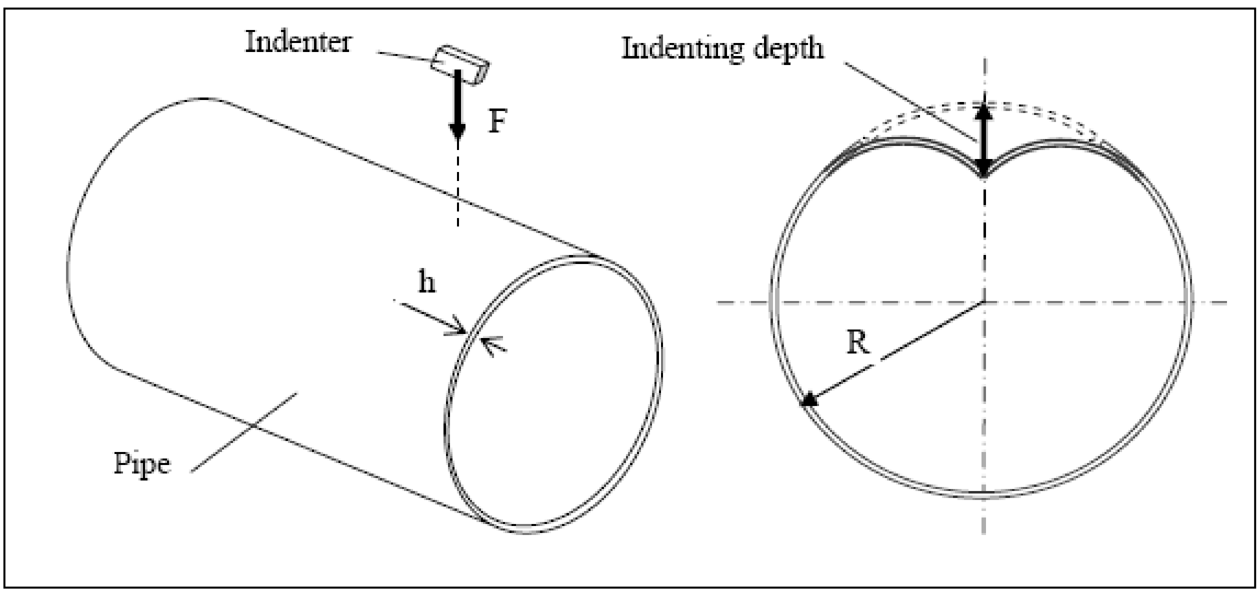

Figure 1. Deformation shape of pipe and ring subjected to impact

Due to impact loading, the failure of pipe is always initially defected by dents that allow a local deformation. The latter may propagate and cause continuously the global deformation, which leads to the complete failure of pipe structure. This seems to show that the analysis on dents is important. Figure 1 demonstrates the simulation of a dent of the pipe after getting impact.

2.1.1 Significances of dents

A dent in a pipeline is a permanent plastic deformation of the circular cross-section of the pipe. A dent is a gross distortion of the pipe cross-section [9]. Dent depth is defined as the maximum deduction in diameter of the pipe compared to the original diameter [Fig.2.b]. This definition of dent depth includes both the local indentation and any divergence from the nominal circular cross-section (example: out-of-roundness of ovality).

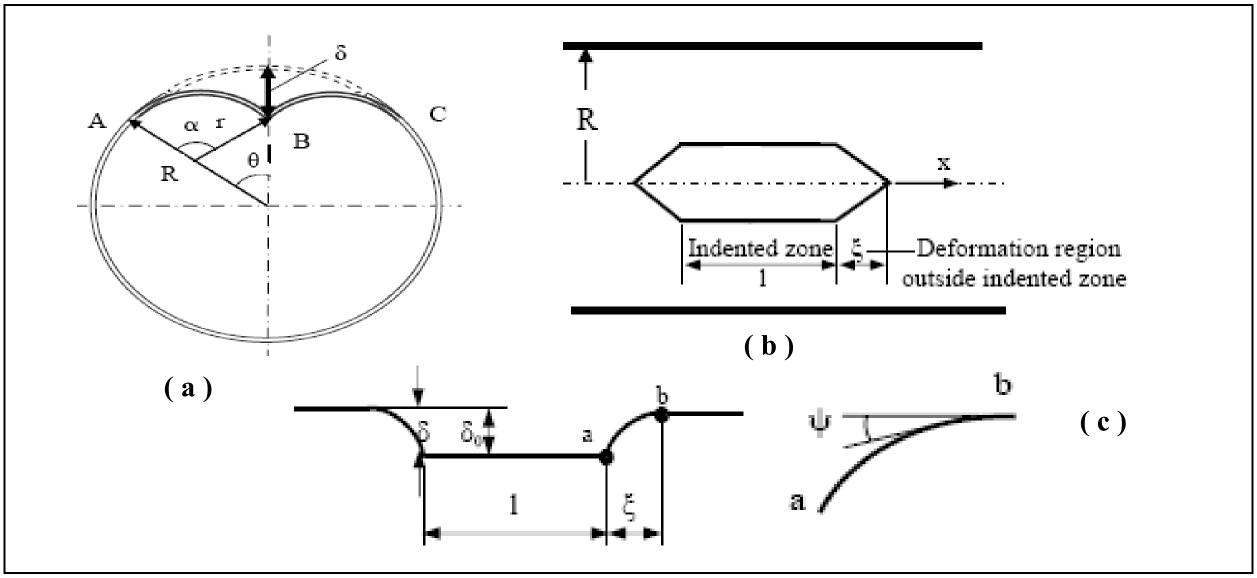

Figure 2. (a) Deformation shape of ring, (b) Plan view of deformation region (c) Slide view of deformation region

A dent causes local stress and strain concentration, and a local reduction in the pipe diameter. The dent depth is the most significant factor affecting the burst strength and the fatigue life of a plain dent. Dents caused by external interferences (unconstrained dents) are typically confined to the top haft of a pipeline. Dents may be associated with a coating damage, and hence may be sites for the initiation of corrosion or environmental cracking. Whether a pipe is gouged during indentation depends on many factors, including the trajectory of the indentation, the fictional resistance between the surfaces of the pipe and the indenter, the shape and sharpness of indenter, the geometry, the material properties and the internal pressure. The stiffer the pipe, the more resistant it is to denting. Damage introduced into pressurized pipes tends to comprise the shallower dents and deeper gouges than that introduced into non-pressurized ones because internal pressure stiffens the pipe.

2.2 Deformation profile

The present study discusses about the local deformation based on an assumption in which the global one is neglected for pipelines since most of denting energy is absorbed by the pipe shell as local deformation. The local indentation on pipeline is a complicated subject involving both geometrical and material nonlinearities. It is impossible to obtain a complete theoretical solution without any simplifications. In another word, some assumptions should be made to bypass the difficulties in this analysis; when the indenting deformation is large the elastic energy stored in the pipe may be neglected compared to with the dissipated plastic energy. In this meaning, the pipe can be simplified as that made of rigid-perfectly plastic material. Therefore, no deformation in the pipe will occur until the stress reaches the yield stress throughout the pipe shell thickness.

Deformation profile of a ring shown in Figure 2 can be used to analyze the impact characteristics of a pressurized and non-pressurized pipe. Further assumption is made for the inextensibility of the rings during the deformation. The local deformation [Fig.2(b)] consists of three zones: the indented zone in the middle which has the form and size like those of the indenter, and two triangular zones on either side. It should be observed that during the indenting process, the length of indented zone (**l**) does not change and remains equal to the length of indenter. The triangular zones will expand outwards, and the length ξ will be increased with the increasing of the denting depth $\boldsymbol{\delta_0}$. The arc length AB on the ring is defined by the angle $\boldsymbol{\theta}$ and also increased with the increasing of the dent depth.

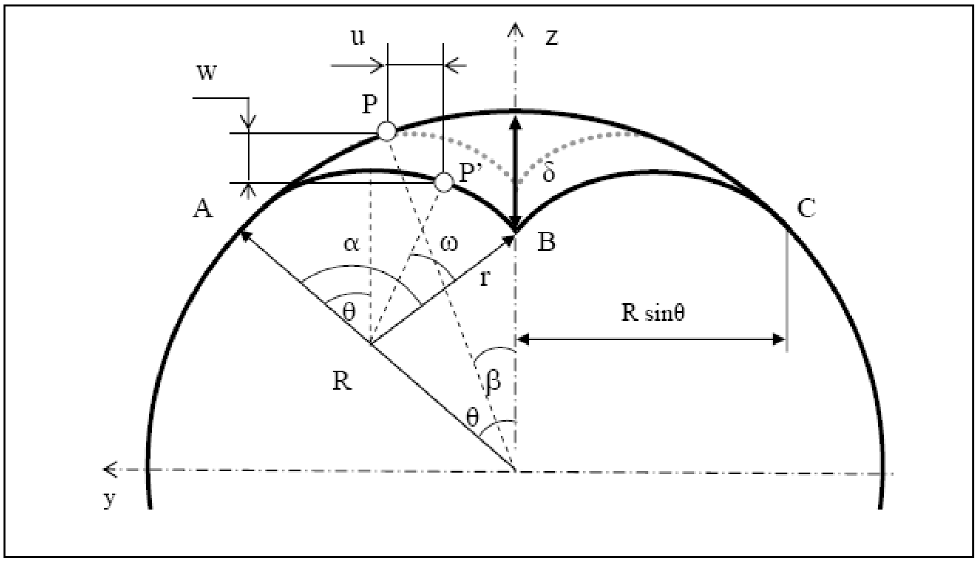

Figure 3. Deformation region on ring section

The curvature of circumferences of the ring, below points A and C, does not undergo any deformation due to the rigid plasticity of the ring. It is noticed that there three plastic hinges; one is the stationary hinge occurring at the force loading point B. Whereas the other two are moving hinges occurring at point A and C. The conditions of the inextensible ring circumference, vertical and horizontal geometry of Figure 3 enables to derive some relations.

Mathematically, the portions of arc from point A and C to another point at the top of ring remained the same length after indentation which gives:

arc AA' = arc AB, then

$$\mathbf{R\theta = r\alpha} \tag{1}$$

And the indent depth can be expressed as

$$\boldsymbol{\delta = R(1 - \cos\theta) - r[\cos(\alpha - \theta) - \cos\theta]} \tag{2}$$

Horizontal projection on axis X, another equation is obtained.

$$\mathbf{AB = BC = R\sin\theta} \quad \text{or} \quad \mathbf{R\sin\theta = r\,[\,\sin\theta + \sin(\alpha - \theta)\,]} \tag{3}$$

A rigid indenter is pushing the pipe downward from the mid-top until the dent depth reaches at least 12% of the pipe diameter. The average deformation arc (r) is changed in function of the indenting depth. With an assumption of R = 4r, a relationship between α and θ is obtained. From the equation (1), $\alpha = 4\,\theta$

And the indent depth can be written as:

$$\boldsymbol{\delta = R\,(1 - \cos\theta)\,(1 + \cos\theta + \cos^2\theta)} \tag{4}$$

By following the analysis of large dynamic deformation of a rigid-plastic beam [6], the velocity field of the leading generator can be assumed to vary linearly with x by which the relationship between the displacements occurring in and around the indenting zone [Fig.2(c)] is defined.

$$\delta = \delta_0\left(1 - \frac{x}{\xi}\right) \tag{5}$$

Application of derivative with respect to the time t of the indentation, gives:

$$\dot{\delta} = \dot{\delta}_0\left(1 - \frac{x}{\xi}\right) \tag{6}$$

Where, $\boldsymbol{\delta = R(1 - \cos\theta)(1 + \cos\theta + \cos^2\theta)}$ **and** $\boldsymbol{\delta_0 = R(1 - \cos\theta_0)(1 + \cos\theta_0 + \cos^2\theta_0)}$ (7)

A setting point P on the original ring moves to another point P' on the deformed arc [Fig. 3]. Then, the horizontal and vertical displacements are defined respectively as:

$$\mathbf{u = R\sin\beta\,[\,1 - \cos(\theta - \beta)\,]} \tag{8}$$

and $\mathbf{w = R\cos\beta\,[\,1 - \cos(\theta - \beta)\,]}$ (9)

During the process of indentation, the deformation region around the indented zone, is being increased with the increasing of the indenting depth. Two triangles representing the region of deformation at both end of indented zone shown in Fig. 2(b) and 2(c) and equation (5) give,

$$\sin\theta = \sin\theta_0\left(1 - \frac{x}{\xi}\right) \tag{10}$$

From equations (5), (7), and (10), it is defined that the angle of the deformed arc for ring outside and within the indented zone has very little different value, which can be assumed that $\theta = \theta_0$

And the derivative in respect to x gives $\dfrac{d\theta}{dx} = -\dfrac{\sin\theta_0}{\xi\cos\theta}$ ***or*** $d\theta = -\dfrac{1}{\xi}\,tg\theta\;dx$ (11)

Since the indenter is supposed to be applied downward onto the pipe, the vertical displacement and the displacement rate are subjected to this analysis, which can be expressed as follow:

$$\mathbf{w = R\cos\beta\,[\,1 - \cos(\theta - \beta)\,]} \tag{12}$$

$$\frac{dw}{dx} = -\frac{R\sin\theta_0\cos\beta\sin(\theta-\beta)}{\xi\cos\theta} \tag{13}$$

Or $$\dot{w} = R\dot{\theta}_0\cos\beta\sin(\theta-\beta) \tag{14}$$

Then, $$\frac{d\dot{w}}{dx} = -\frac{R\dot{\theta}_0\sin\theta_0\cos\beta\cos(\theta-\beta)}{\xi\cos\theta} \tag{15}$$

2.3 Crushing force of ring

While the object is moving downwards from the middle top of pipe, there is a circumferential bending by which the deformation is increasing continuously [3, 8]. Occurrence of three plastic hinges is considered. This shows the dominant absorption of energy in an inextensible ring. The rigid-plasticity of the ring allows the bending deformation to occur in the location where the bending moment reaches the yield stress throughout the pipe shell thickness. The fully plastic moment remained per unit length can be express as:

$$\mathbf{M_0 = \sigma_y h^2/4} \tag{16}$$

Where, σ_y is the yield stress, and h is the shell thickness of the pipe.

The dissipated energy rate for the stationary hinge is $2M_0(\dot{\alpha}-\dot{\theta}) = 6M_0\dot{\theta}$.

When the moving hinge passes through the top part of the ring with a certain moving velocity $v = R\dot{\theta}$, the curvature changes from 1/R to 1/r. The dissipated energy in the deformed arc AB and BC is expressed as $2M_0\alpha r(-\frac{\dot{r}}{r^2}) = 0$, since R = 4r is constant.

The balance of energy rate in the ring gives:

$$F_c\dot{\delta} = 6M_0\dot{\theta}\left[1+\left(\frac{R}{r}-1\right)\right] = 24M_0\dot{\theta} \quad or \quad \frac{F_cR}{24M_0} = \frac{1}{\sqrt{\bar{\delta}(2-\bar{\delta})}} \tag{17}$$

Where, F_c is the crushing force and $\bar{\delta} = \delta/R = (1-\cos\theta)(1+\cos\theta+\cos^2\theta)$

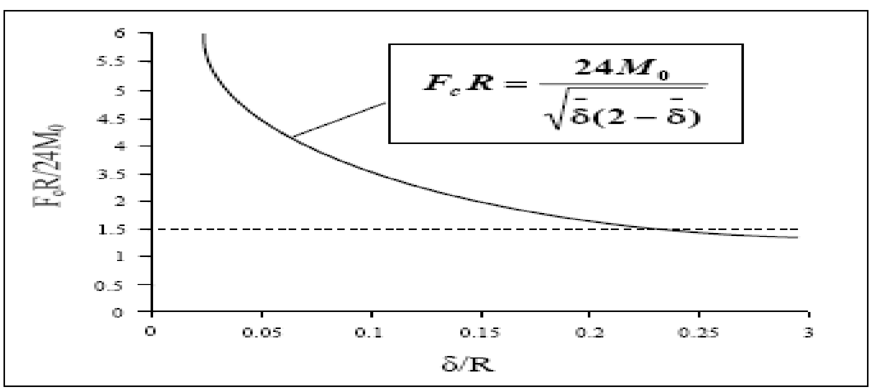

Figure 4. Ring crushing force F_c plotted with indent depth δ

The deformation mode in which each hinge is initiated gives initially an infinite crushing force and should not be considered as a valid mode [3]. As assumption, Wierzbicki and Suh used $F_c R/4M_0 = 2$ as crushing force for their ring mode consisting of four plastic hinges with flat top deformation. In this case, $F_c R/24M_0 = 1.5$ is chosen according to the relationship F_c - δ for the deformation model of the ring in figure 2(c).

2.4 Energy rate

2.4.a Ring crushing energy rate

The dissipated energy per unit length of deformed arc is M_0 (R- r)/Rr^2 . The total ring crushing energy rate in the whole deformation can be defined as the summation of crushing energy outside and within the indented zone.

$$\dot{E}_{crush} = 2\int_0^{\xi} F_c\,\dot{\delta}\,dx + \int_0^{l} F_c\,\dot{\delta}_0\,dx = 9\sigma_y h^2(\xi + l)\frac{\dot{\delta}_0}{R} \tag{18}$$

2.2.4.b Generator stretching energy rate

For the deformation region outside the indented zone, the bending energy rate along the pipeline axis can be neglected since the change of curvature of the generators is relatively smaller than that of the circumferential curvature of the rings [3]. However, within the indented zone, the ring deformation is the same and there is no stretching nor bending along generators (along pipeline axis). Therefore, the total dissipated energy on the generator is the stretching energy occurring in the deformation regions outside the indented zone. The stretching deformation occurs at the location where the axis force reaches and remains as the fully plastic axial force per unit area ($N_0 = \sigma_{yield}$ h).

The dissipated stretching energy rate per unit area of pipe shell is $N_0\dot{\varepsilon}$, where ε is the tensile strain for a moderately large deflection of the beam [3].

$$\varepsilon = \frac{\sqrt{(dw)^2 + (dx)^2 - dx}}{dx} = \sqrt{1 + \left(\frac{dw}{dx}\right)^2} - 1 \approx \frac{1}{2}\left(\frac{dw}{dx}\right)^2$$ and Tensile strain rate is $\dot{\varepsilon} = \left(\frac{dw}{dx}\right)\left(\frac{d\dot{w}}{dx}\right)$

The total stretching energy rate for generator can be expressed as:

$$\dot{E}_{gen} = 4N_0\int_0^{\xi}\int_0^{R\theta}\dot{\varepsilon}\,ds\,dx = 4\sigma_y\int_{\theta_0}^{0}\int_0^{\theta}\frac{dw}{dx}\frac{d\dot{w}}{dx}(Rd\theta)\left(-\frac{\xi\cos\theta}{\sin\theta_0}d\theta\right)$$

$$= N_0R^2\left\{\ln\left[1 + \sqrt{\bar{\delta}_0(2-\bar{\delta}_0)}\right] - \ln(1-\bar{\delta}_0) - (1-\bar{\delta}_0)\cos^{-1}(1-\bar{\delta}_0)\right\}\frac{\dot{\delta}_0}{\xi}$$

(19)

An approximation in Ref. [5] is applied to simplify the dissipated stretch energy rate of generator, which can be written as:

$$\dot{E}_{gen} \approx 1.4 N_0 R^2 \bar{\delta}_0 \sqrt{2\bar{\delta}_0}\,\frac{\dot{\delta}_0}{\xi} \tag{20}$$

The balance of energy rate with the incremental dent depth $d\delta_0$ gives: $F_0\,\dot{\delta}_0 = \dot{E}_{crush} + \dot{E}_{gen}$

$$F_0\,\dot{\delta}_0 = 9\sigma_y h^2\,\dot{\delta}_0\,\frac{(\xi + l)}{R} + 1.4 N_0 R^2 \bar{\delta}_0 \sqrt{2\bar{\delta}_0}\,\frac{\dot{\delta}_0}{\xi} \tag{21}$$

Or $$F_0 = 9\sigma_y h^2\,\frac{(\xi + l)}{R} + 1.4\sigma_y h R^2 \sqrt{2\bar{\delta}_0}\,\frac{\bar{\delta}_0}{\xi} \tag{22}$$

2.5 Resistant work done by pressure

The response of internal pressure has a significant influence on the pipeline resistance to the deformation subjected to impact loading and reduces the level of risk of this case. The pressure resisting the impact energy on pipe is the difference between the internal pressure and the external one in the case of subsea or deep water pipeline. When the indenter is applied downwards the pipe, the pressure $P = P_{in} - P_{out}$ will response in the opposite direction. This shows that the internal pressure has important influence in resisting the external impact load. The total resistant work done by the pressure can be expressed as:

$$\begin{aligned}\dot{E}_p &= 2l\int_0^{R\theta_0} p\cos(\theta_0 - \omega)\,\dot{w}_0\,dx + 4\int_0^{\xi}\int_0^{R\theta} p\cos\psi\cos(\theta - \omega)\,\dot{w}\;dsdx \\ &= pR\left(l + \frac{2}{3}\xi\right)\sqrt{\bar{\delta}_0\left(2 - \bar{\delta}_0\right)}\,\dot{\delta}_0\end{aligned} \tag{23}$$

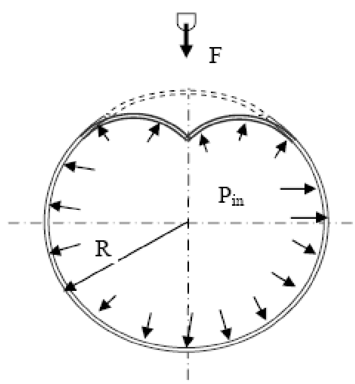

Figure 5. Deformation of a pressurized pipe

In the case of indentation on pressurized pipe, the balance of energy rate can be defined as:

$$F\,\dot{\delta}_0 = \dot{E}_{crush} + \dot{E}_{gen} + \dot{E}_p \tag{24}$$

Finally, a relationship between the indenting force F_c and indent depth δ is obtained:

$$F = 9\sigma_y h^2\,\frac{(\xi + l)}{R} + 1.4\sigma_y h R^2 \sqrt{2\bar{\delta}_0}\,\frac{\bar{\delta}_0}{\xi} + pR\left(l + \frac{2}{3}\xi\right)\sqrt{\bar{\delta}_0(2 - \bar{\delta}_0)} \tag{25}$$

3. Discussion

Deformation of the pressurized in-service pipeline probably caused by third party activities is theoretically analyzed in this paper. With the assumption of rigid-perfectly plasticity and inextensibility, the pipe impact behaviors are discussed to simulate the case of damages on pipelines during transmission

and operation. The deformation profile shown in Figure 2 is used for the analysis in both cases, pressurized and non-pressurized pipes. It is deduced that the indenter is applied accidentally downward from some height on the middle top of pipe. The indent of certain depth is one of the main variables from which some formulas have been derived. This analysis mentioned the balance of energy rate in which the sum of total dissipated incremental energy for ring crushing and generator deformation must equal the incremental work done by the indenting force [Equation (21)]. Finally, the externally accidental force of indenter affecting the pipe structure can be defined. However, when the internal pressure of the pipe is included, the right side of the balance of energy rate becomes bigger due to the resistant work done by pressure. Equation (23) implies that in pressurized pipe, the stronger force (F) is needed to cause the same indent depth to that in non-pressurized one. This reveals that the response of the internal pressure is absolutely necessary to reduce the dent depth as well as the level of damage on the pipe structure.

3. Conclusion

The theoretical quasi-static analysis presented in this paper is proposed to predict the response of pressurized in-service pipelines subjected to external impact loading. The pipe is assumed to be inextensible and made from rigid-perfectly plastic for which the deformation in the pipe will occur until reaches the yield stress throughout the pipe shell structure. The balance of energy is respected. The total energy of ring is the summation of dissipated energy of crushing force, dissipated energy of stretching for generator and the work done by resisting pressure of pipe. The local deformation on the pipe structure involves the analysis while the global deformation is neglected due to large local assumption of energy during the process of indentation. An important relationship between the dent depth and indenting force is defined. The analysis has revealed that the internal pressure has a significant influence in resisting and reducing the damage subjected to external interference on pipeline structure. The different of the required force between a lowly pressurized pipe and a highly pressurized pipe increases with increasing of the denting depth.

ACKNOWLEDGEMENT

The authors would like to express sincere thanks to AUN/SEED. Net, JICA for the strong support this research. The gratitude is also expressed to Dr.Ir. IGN Wiratmaja Puja for the valuable advices in the sake of this improvement. Some more lecturers in Department of Mechanical Engineering of Institut Teknologi Bandung have involved directly and indirectly in the completion of this paper. Furthermore, this success cannot be separated from the kind helps of some Indonesian friends in Engineering Design Centre of ITB.

REFERENCES

[1] Cosham A, Hopkins P. The effect of dents in pipelines. International Journal of Pressure Vessels and Pipings 81 (2004) 127-139, Newcastle, UK.

[2] Andrew P, MartinT, Murry A. Full-scale impact tests on pipelines. International Journal of Impact Engineering 32 (2006) 1267-1283, Scotland, UK.

[3] Wierzbicki T, Suh MS. Indentation of tubes under combined loading. International Journal of Mechanical science 1988; 30; 229-48

[4] DeRuntz JA, Hodge PG. Crushing of tubes between rigid plates. J Appl Mech 1963; 30; 391-5

[5] Reid SR, Roddy. Metal forming and impacts mechanics. New York: Pergamon Press 1985.

[6] Spyros A, Karamanos. Denting of internally pressurized tubes under lateral loads. International Journal of Mechanical Science 48 (2006) 1080-1094, Greece.

[7] Kaisheng CH, Wei Q. Further experimental study on the failure of fully clamped steel pipes. International Journal of Impact Engineering Vol. 21, 1998, Portsmouth, UK.

[8] Suh D, Deng J. Investigation of pressure in pipe subjected to axial-symmetric pulse loading. International Journal of Impact Engineering 25 (2001) 523-536, Singapore.

[9] Hopkins P, Jones DG. The significance of dents and defects in transmission pipelines. Conference on Pipework Engineering and Operating 1989, London, UK.