THE EFFECTS OF LOCAL BUCKLING ON THE CRASH ENERGY ABSORPTION OF THIN-WALLED EXPANSION TUBES

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ABSTRACT: This paper deals with the local buckling characteristics and the energy absorption of thin-walled expansion tubes during tube flaring processes. The local buckling load and the absorbed energy during the flaring process were calculated for various types of tubes and punch shapes by finite element analysis. The energy absorption capacity of the expansion tube is influenced by tubes and punch shapes. The parametric study shows that the absorbed energy of the expansion tube increases as the diameter and the wall thickness of tubes increase. Larger punch angle and expansion ratio also improve the energy absorption. However, local buckling takes place relatively easily at larger punch angle and expansion ratio. Local buckling loads are also influenced by both the tube radius and thickness. Accurate prediction of the local buckling load is important to improve the energy absorption of the expansion tube since the absorbed energy of an expansion tube decreases significantly when local buckling occurs. Local buckling loads were predicted by modification of the Plantema equation and compared with numerical results. A modified Plantema equation shows a good agreement with the numerical result.

KEYWORDS: Expansion tube, Crash energy absorption, Local buckling, Modified Plantema equation.

1 INTRODUCTION

Energy absorption mechanism of an expansion tube is mainly expansion of the tube diameter by pushing a conical punch into the tube while the bottom of the tube is fixed. It is noted that the specific energy absorption for an expansion tube is not efficient compared with the crushing energy absorber [1]. However, an expansion tube can reduce the crushing acceleration since the reaction force during expanding process increases gently compared to the other types of energy absorber. A smaller peak load can reduce the damage in the main equipment caused by the crushing acceleration. The other advantage of an expansion tube is that the total length of an expansion tube can be used for energy absorption. For that reason, an expansion tube can absorb the crash energy as much as a crushing energy absorber since the total length of a crushing energy absorber cannot be used for energy absorption. In spite of those advantages, an expansion tube cannot be used for the equipment which needs weight reduction because of the heavy weight of a punch for expanding a tube. Therefore, an expansion tube is being used for heavy equipments such as a train. Further study for space efficiency of an expansion tube should be carried out in order to overcome the disadvantage of an expansion tube.

The tube expanding mechanism can be referred by a tube flaring process. Flaring is understood as a forming process involving the expansion of a cylindrical tube which is expanded by a conical punch pushed into the tube. Many researchers carried out studies for tube flaring processes. Hill [2] presented a mathematical model of stress flow during tube expansion. It can be a general method for many studies on tube flaring processes. Manabe *et al*. [3-4] conducted a series of experiments related to a tube flaring process using the rigid plastic finite element theory. They investigated the effects of the size and mechanical properties of tubes, lubricants and punch angle on the flaring process. Elasto-plastic finite element analyses for tube flaring processes were presented and compared with experimental results by Huang *et al*. [5]. Lu [6] investigated the expansion ratio and punch velocity in flaring process by finite element analysis. Failure due to plastic buckling or necking is not considered in these studies although those are the most important instability in tube flaring

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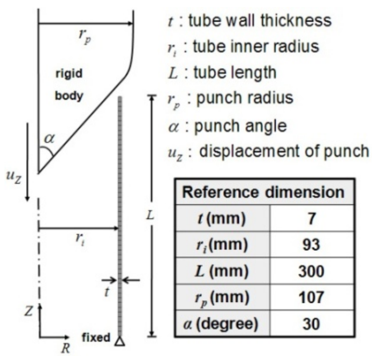
process. Daxner *et al*. [7] investigated the effect of local buckling and necking on the tube end in flaring. They found out the forming loads for preventing buckling and necking by finite element analysis and compared with experimental results. Almeida *et al*. [8] considered the effect of lubricant on instability phenomena in flaring process by experiments. Analytical expressions were derived for determining stress and strain fields as well as the force required for driving the tube expansion and compared with finite element solutions by Fischer *et al*. [9].
This paper investigates mechanism of energy absorption during a tube flaring process. The effect of a tube and the punch shape on energy absorption is calculated using finite element analysis. Local buckling characteristics during a tube expanding process are evaluated since local buckling has a critical effect on energy absorption of an expansion tube. For enhancement of energy absorption, local buckling load is predicted by a modified Plantema equation with respect to the tube shape.

2 ENERGY ABSORPTION OF EXPANSION TUBES

2.1 FINITE ELEMENT MODEL AND BOUNDARY CONDITIONS

Finite element analysis is carried out for considering the effect of a tube and the punch shape on energy absorption of an expansion tube. Energy absorption and the buckling load of an expansion tube are derived by finite element analysis with respect to a tube and the punch shape. An implicit elasto-plastic finite element code, ABAQUS/Standard, is used for parametric study.
An expansion tube is being used as the energy absorber in a safety device for light collision in a train. The dimensions of a tube and the punch shown in Figure 1 are a typical example of expansion tubes in a train. Figure 1 shows a finite element model and the dimensions of a tube and a punch. The axi-symmetric condition is used for efficiency of analysis. Nodes on the tube bottom are all fixed. For parametric study, each parameter is changed from the reference dimension in Figure 1.



Reference dimension	
t (mm)	7
r_i (mm)	93
L (mm)	300
r_p (mm)	107
α (degree)	30

Figure 1: *Finite element model of tube and punch*

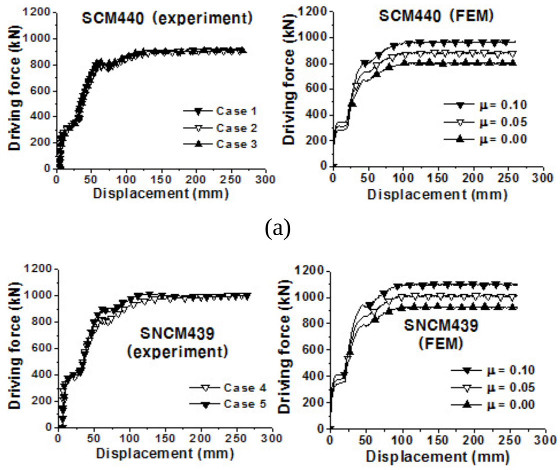


Figure 2: *Driving force of punch from experiment and numerical analysis: (a) SCM440; (b) SNCM439*

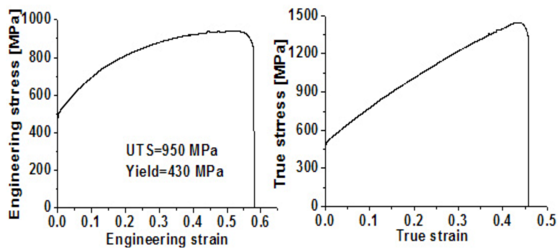


Figure 3: *Engineering and true stress–strain curves of TWIP steel.*

2.1.1 Friction

The friction condition is important in an expansion tube since the inner surface of a tube experiences sliding during the expanding process. Choi *et al*. [10] carried out experiments using two materials, SCM440 and SNCM439. Finite element analyses are performed with respect to various friction coefficients in order to find out the friction conditions of experiments. Figure 2 shows the comparison between experimental and analysis results. As shown in Figure 2, analysis results by using the friction coefficient of 0.05 give a good coincidence with experimental results. In this paper, the friction coefficient is selected as 0.05 for all analyses.

2.1.2 Material Selection

In order to enhance crashworthiness of an energy absorber, a high strain hardening material should be used. In case of an expansion tube, tearing in a tube end can occur during the tube expanding process. In order to avoid tearing phenomena, materials for an expansion tube should have high elongation characteristics as well as high strain hardening. In this paper, TWIP steel has been used for a tube material. TWIP steel has not only high strain hardening properties but also high elongation. Tensile tests are carried out in order to obtain the material properties of TWIP steel. Figure 3 shows engineering and true stress–strain curves of TWIP steel.

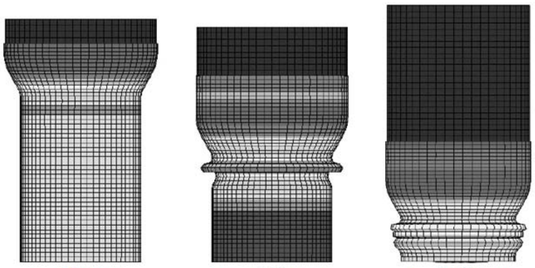
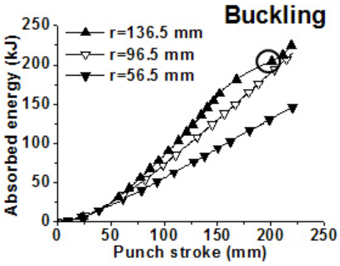
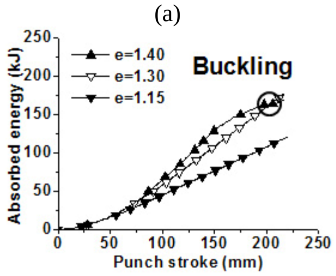


Figure 4: *Deformed shape when buckling occurs (t=9 mm, r_i =113 mm, α=35°, e=1.4)*



(a)



(b)

Figure 5: *Effects of buckling: (a) effects of tube radius when t=5 mm, α=45°, e=1.3; (b) effects of expansion ratio when t=7.5 mm, r_i =93 mm, α=45°*

2.2 PARAMETRIC STUDIES FOR ENERGY ABSORPTION

A parametric study is carried out with finite element analysis in order to evaluate the effect of a tube and the punch shape on energy absorption. Each parameter is changed from the reference dimension in Figure 1. Parameters for the parametric study are the tube wall thickness, the tube radius, the punch angle and the expansion ratio. The expansion ratio means the ratio of the punch radius to the tube inner radius. In this paper, a symbol, *e*, represents the expansion ratio.

In accordance with the result of the parametric study, energy absorption capacity of an expansion tube increases as the tube wall thickness, the tube radius, the punch angle and the expansion ratio increase [11]. However, buckling can occur when all parameters increase excessively. Figure 4 shows the deformed shape when buckling occurs. Dimensions of all parameters in Figure 4 are larger than those of the reference shape. Figure 5 shows the effect of local buckling on energy absorption. As shown in Figure 5, absorbed energy decreases significantly when buckling occurs. Therefore, studies to prevent buckling are needed to enhance the energy absorbing capacity of an expansion tube.

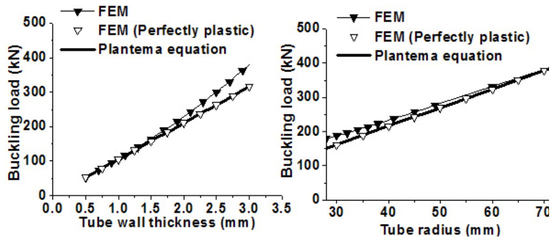


Figure 6: *Local buckling load of expansion tubes*

3 LOCAL BUCKLING OF EXPANSION TUBES

In the previous results, buckling is a critical factor in energy absorption of an expansion tube. In order to predict the buckling load, buckling mode should be determined. There are four types of buckling behaviors, elastic column buckling, inelastic column buckling, elastic local buckling and inelastic local buckling [12]. Local buckling is used to define failures that involve a major change in the geometry of the cross section. Buckling mode of an expansion tube is inelastic local buckling since the buckling mode is an wrinkle shape as shown in Figure 4 and the stress flow exceeds the elastic limit.

3.1 PLANTEMA EQUATION

The equation most commonly used for predicting inelastic local buckling of a tube is the Plantema equation [12]. This relationship is given by:

$$\begin{aligned} &\sigma_{cr}/\sigma_y = 1.0 && \text{for } \alpha \geq 8 \\ &\sigma_{cr}/\sigma_y = 0.75 + 0.031\alpha && \text{for } 2.5 \leq \alpha \leq 8 \\ &\sigma_{cr}/\sigma_y = 0.33\alpha && \text{for } \alpha \leq 2.5 \end{aligned} \tag{1}$$

in which α is the nondimensional local buckling parameter which can be expressed for a circular tube as follows:

$$\alpha = \left(\frac{E}{\sigma_y}\right)\left(\frac{1}{D/t}\right) \tag{2}$$

Figure 6 shows comparison between the buckling load predicted by the Plantema equation and finite element analysis. The reference tube dimension for comparison has a thickness of 2 mm and a radius of 40 mm. The tube wall thickness and radius are changed from the reference dimension in order to verify the Plantema equation for various shapes of tubes. The tube shape for analysis is the same as the one in the previous chapter. The punch shape is, however, a flat type since the punch angle is not needed for buckling analysis. As shown in Figure 6, the Plantema equation cannot predict the accurate local buckling load for various tube shapes. Hollow symbols in Figure 6 mean the analysis result with perfectly plastic assumption and those results agree well with the result from the Plantema equation since the Plantema equation did not include the strain hardening effect.

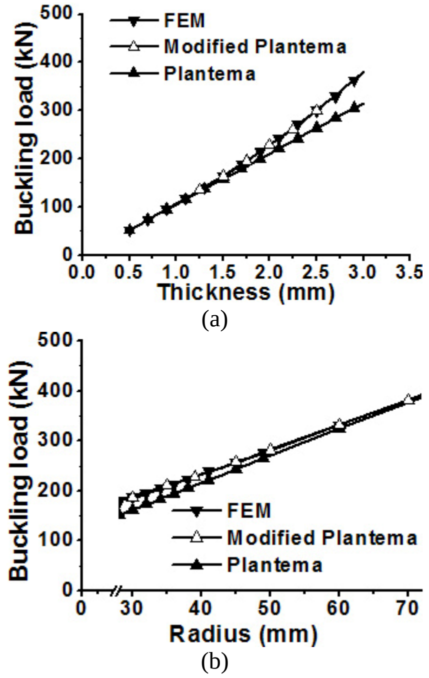


***Figure 7:** Local buckling load of expansion tubes using modified Plantema equation w.r.t.: (a) tube wall thickness; (b) tube radius*

3.2 MODIFIED PLANTEMA EQUATION CONSIDERING STRAIN HARDENING EFFECT

In order to predict the accurate local buckling load, the strain hardening effect should be added to the Plantema equation. Figure 7 shows the result predicted by substituting averaged compressive stress on fixed nodes for the yield stress in the Plantema equation. The result shows a good agreement with numerical results since the compressive stress includes the strain hardening effect. For that reason, effects of averaged stress on fixed nodes have to be investigated in order to predict the accurate local buckling load. Figure 8 shows the relation between the averaged stress and the equivalent strain. For the simplicity, the equivalent strain is defined by the ratio of the punch stroke to the tube length. Figure 8 shows that the averaged compressive stress on fixed nodes follows the stress–strain curve of the material. By considering these results, the local buckling load can be predicted as follows:

$$P_{cr} = 2\pi rt\left(A + B\varepsilon_{cr}^{n}\right)$$
$$\text{where, } \sigma = A + B\varepsilon^{n} \text{ (Ludwik model)} \tag{3}$$
$$\varepsilon_{cr}: \text{buckling strain}$$

The buckling strain is the equivalent strain when buckling occurs. Figure 9 shows the analysis result of buckling strain for various shapes of tubes. The buckling strain for each shape is propotional to the square value of the ratio of the tube wall thickness to the tube radius. Therefore, equation (3) can be modified as follows:

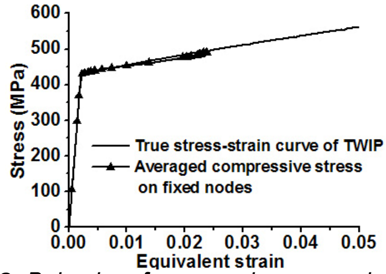


***Figure 8:** Behavior of averaged compressive stress on fixed nodes*

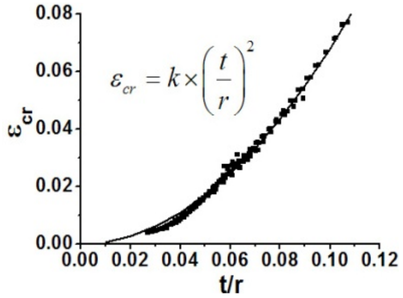


***Figure 9:** Buckling strain w.r.t. the ratio of tube wall thickness to radius*

$$P_{cr} = 2\pi rt\left[A + B\left\{k\left(\frac{t}{r}\right)^2\right\}_{cr}^{n}\right] \tag{4}$$

where k is a fitting parameter determined by the material properties. In case of TWIP steel, k has a value of 6.8. By using equation (4), local buckling loads for various tube shapes can be predicted with the fitting parameter k of each material. The fitting parameter of each material has to be obtained from finite element analysis. Table 1 shows local buckling loads for various tube shapes. Numerical results show a good agreement with the results by the modified Plantema equation rather than the Plantema equation.

***Table 1:** Buckling load predicted by each method*

Tube shape		Buckling load (kN)		
r (mm)	t (mm)	ABAQUS /Standard	Modified Plantema	Plantema
30	1.5	130.8	130.2	121.6
	2.0	187.8	188.6	162.1
	2.5	255.1	256.5	202.6
	3.0	333.4	337.6	243.2
40	1.5	164.6	162.8	158.1
	2.0	230.0	229.8	210.8
	2.5	302.7	305.4	263.4
	3.0	385.8	390.5	316.1
50	1.5	206.1	204.8	202.6
	2.0	282.3	282.7	270.2
	2.5	367.2	368.6	337.7
	3.0	458.8	462.9	405.3
Averaged error			0.7 %	11.4 %

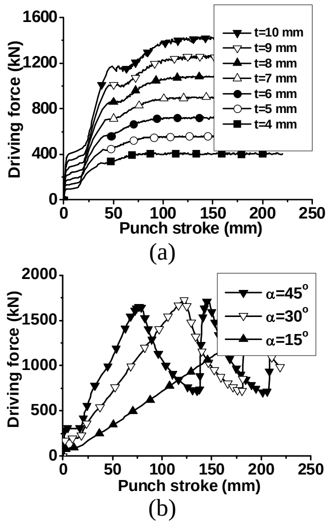


Figure 10: *Reaction force of punch w.r.t.: (a) tube wall thickness; (b) punch angle*

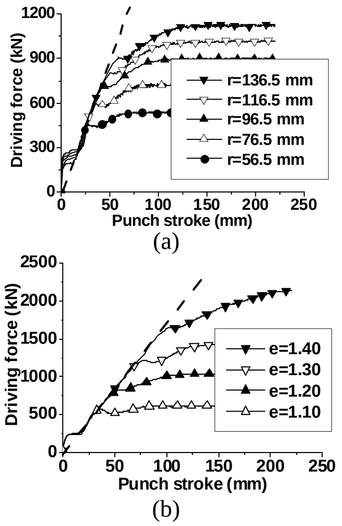


Figure 11: *Reaction force of punch w.r.t.: (a) tube radius; (b) expansion ratio*

4 DESIGN OF EXPANSION TUBE

In order to prevent local buckling, the maximum reaction force of the punch should not exceed the local buckling load. The maximum reaction force for each tube shape has to be verified to compare with local buckling load. Figure 10 and 11 show the reaction force of the punch with respect to various shapes of tubes and punches. As shown in Figure 10, the initial gradient of the reaction force to the punch stroke, P/u_Z, is related to the tube wall thickness and the punch angle. The tube radius and the expansion ratio have no effect on the gradient of the reaction force as shown in Figure 11. In accordance with the analysis results, the gradient of the reaction force to the punch stroke is proportional to the product of the tube wall thickness and the punch angle. In case of TWIP steel, the reaction force of the punch increases with the relation of

$$P = 4.73\alpha t u_Z \quad (5)$$

When the radius of a tube end is fully expanded to the punch radius, the increment of the reaction force decreases. By geometry, the radius of the tube end is fully expanded when the punch stroke reaches to the value of

$$u_Z = (\rho + t/2)\alpha + \frac{(e-1)r_i}{\sin\alpha} \quad (6)$$

where, ρ : radius of punch fillet

Therefore, the maximum reaction force of a punch can be represented as follows:

$$P_{\max} = 4.73\alpha t\left[(\rho + t/2)\alpha + \frac{(e-1)r_i}{\sin\alpha}\right] \quad (7)$$

Figure 12 shows the reaction force predicted by equation (5) and (7). In some case, as shown in Figure 12(b), the reaction force keep increasing after the radius of the tube end is fully expanded

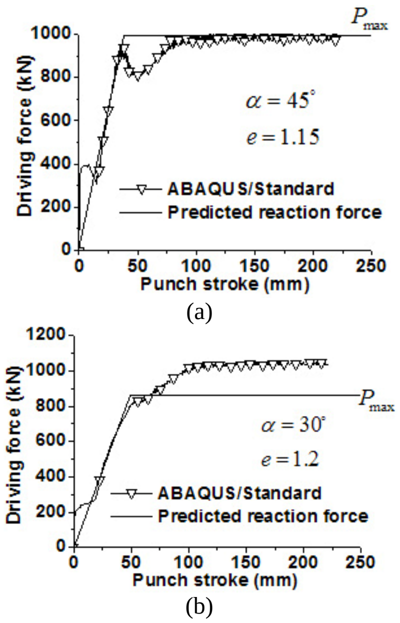


Figure 12: *Predicted reaction force of punch: (a) α=45°, e=1.15; (b) α=30°, e=1.2*

because of the effect of friction and the simplification of geometry in equation (6). In this paper, those effects are neglected since buckling hardly occurs in that region.

Local buckling predicted by the modified Plantema equation is the value when the punch has a flat shape. In case of the expansion tube, however, the punch shape has an inclined angle for tube expansion. According to analysis results, local buckling load decreases by 10 % of the maximum when the punch has an inclined angle. Following safety factor has to be adopted to predict local buckling load of the expansion tube considering the effect of the punch angle.

$$\begin{aligned} P_{\max} &< 0.9P_{cr} \quad \text{: stable} \\ P_{\max} &\geq 0.9P_{cr} \quad \text{: buckling} \end{aligned} \quad (8)$$

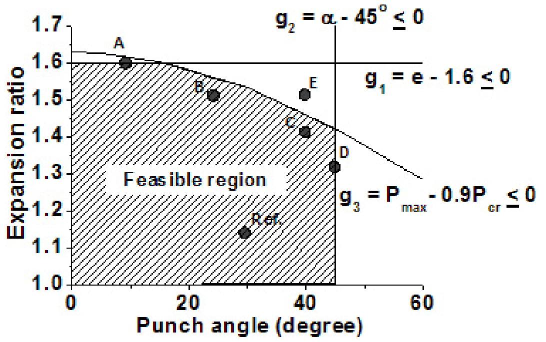


***Figure 13:** Stable region for design*

Figure 13 shows the example of design of the expansion tube. A new design is carried out in order to improve energy absorption of the expansion tube occupying the same space with the tube in Figure 1. The tube wall thickness and the radius are chosen as the maximum value in that space, since absorbed energy increases when both parameters increase. The first constraint, g_1, is determined in order to prevent tearing of the tube end [9]. The second constraint, g_2, is determined to prevent outward curling. When the punch angle increases, outward curling can occur [13] and it gives a significant decrease of energy absorption. In addition, the third constraint, g_3, should be determined by the modified Plantema equation to prevent local buckling. Table 2 shows the absorbed energy for each design point. In point C, absorbed energy increases by 101.1 %.

***Table 2:** Absorbed energy of newly designed model*

Design point	Shape		Absorbed energy (kJ)
	α	e	
Ref.	30°	1.15	200.0
A	10°	1.6	181.4
B	25°	1.5	343.0
C	40°	1.4	404.4
D	45°	1.3	360.1
E	40°	1.5	295.4 (buckling)

5 CONCLUSIONS

This paper suggests a guideline to prevent local buckling in expansion tube design. Local buckling load has to be predicted in order to improve energy absorption of the expansion tubes. The Plantema equation which used to be a most common equation for predicting inelastic local buckling load could not give the accurate local buckling load. The Plantema equation is currently modified by considering the strain hardening effect and the modified Plantema equation gives accurate local buckling loads corresponding to numerical results. The modified Plantema equation proposed in this paper is able to suggest a constraint for a tube design to avoid a local buckling.

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