Investigation of Pre-deformation's effect on the Energy-Absorption of Thin-walled Tube

Wang Siwen[1], *Yang Jikuang*[1, 2]

（1．State Key Laboratory of Advanced Design and Manufacture for Vehicle Body, Hunan University, Changsha 410082,china；2．Department of Machine and Vehicle Systems, Chalmers University of Technology, Sweden）

Abstract: We performed a theoretical analysis of the application of thin-walled tubes in the field of engineering and constructed the FE model of such structure in LS-DYNA, validated this model by the result of crash test. Analysis of the acceleration and energy absorption characteristics of thin-walled tube based on the FE model regarding pre-deformation was performed. We found that pre-deformation is an important factor that increases the energy absorption because it made easy to get overlapping. Moreover the parameters of the trigger influence the energy absorption in various ways. The results confirmed also that the thin-walled tube can be used to absorb energy efficiently in the engineering field.

Key words：Thin-walled tube, FE, pre-deformation, energy absorption characteristic

1. Introduction

To minimize the deformation of some compartments under crash we often need for energy absorbers, devices designed to absorb the impact energy in a controlled manner. Thin-walled tubes, particularly with square or circular cross-section, are a common type of energy absorbers since they are relatively cheap, versatile and efficient. Such absorbers are used in automotive structures and trains, and as arrestors at the base of lift shafts. The crash behaviors of the thin-walled structures during axial collapse and their axial resistance have aroused a lot of interest. Researchers have thoroughly investigated the crash behavior of different types of the thin-walled structures by experimentation or numerical simulation and developed a series of mathematical equations to predict the resistances and developed various models to analyze the axial bucking property based on plastic hinge theory.

Wierzbicki et al. [1] and Abramowicz and Wierzbicki [2] developed mathematical equations that accurately predict the axial resistances of the rectangular, square, and hexagonal section tubes. Abramowicz and Jones [3] also studied the crushing behavior of the circular thin-walled tubes when subjected to an axial impact during the crash.

Mamalis et al. [4] simulated the crash behavior of steel thin-walled tubes subjected to axial loading, which has octagonal cross section. In this article, based on the prediction method of crush behavior of multicorner thin-walled tubes [2], a general equation is derived to estimate the axial resistance of these tubes. They derived axial resistance then can correctly predict the relationship between the axial load and the axial deformation of the octagonal tubes.

One of important research area is a simplified modeling of the axial resistance of thin-walled tubes. This is usually performed on computers by FE model that is composed of beam and spring elements. This modeling technique has been extensively applied in early design stage of product evaluation and crashworthiness analyses. Compared to detailed model, it requires less modeling work and consumes less computer resources.

Liu and Day [5], Kim et al.[6], and Drazeticet al. [7] have created simplified models of the thin-walled tubes as well as an entire vehicle assembly and summarized modeling methodologies for such tube members. Their model was then used for crash analyses and the results were compared to those from the detailed model and from the published literature [8].

Previously performed investigations considered not only model developed, but also studied the characteristics of thin walled tube. Alexander [9] observed in the experiment an accordion-like axi-symmetrical deformation pattern of a cylinder under plastic buckling, and proposed the concept of the static plastic hinge. Based on observations of the destruction pattern of the cylinder under axial compression, he proposed the model describing energy absorption, which is still adopted by most of the scholars in the field because computation is simple and results are consistent with experimental outcomes. Jones [10] divided the plastic buckling of a cylindrical shell under axial loading into dynamic-progressive and dynamic-plastic one. When a structure is exposed to impact at low velocity, members from one end to the other gradually develop overlapping shrinkage, which process is called dynamic- progressive buckling. Dynamic- plastic buckling is an overall buckling behavior that is caused by the repeated reflection of the stress wave when the structure is under an impact at high velocity.

Langseth, et al. [11] conducted impact tests on a rectangular AA6060 aluminum-alloy structure. To investigate if such material is suitable to absorb energy, they studied the effect of impact mass and speed by LS-DYNA. They found that he destruction pattern and the energy absorption properties in simulations were consistent with experimental results.

Karagiozova and Jones [12] adopted the non-linear finite element code of ABAQUS to analyze an aluminum-alloy rectangular thin-walled tube, determining the effect of the impact speed and the impact masses on the axial buckling behavior. In their simulation, the property of aluminum was found to be insensitive to the strain rate. This finding was consistent with the experimental results. They also found that the discrepancy between the dynamic buckling behaviors of cylindrical and rectangular tubes was mainly associated with the transmission of the stress wave in the structures of different geometric shape.

Tai et al [13] used the non-linear finite element software LS-DYNA to investigate the dynamic compression behavior and energy absorption property of high strength thin-walled tubes under various impact masses and impact velocities, taking into account the strain rate and geometric thickness of the material.

Based on the literature review one can say that the velocity and mass of impactor, the material and geometry of thin-walled tube, the strain rate is influencing the energy absorption. Moreover it seems that the pre-deformation is also important for the energy absorption [14]. However, a few studies have been reported if the pre-deformation affect the characteristics of the thin walled tube, especially under crash test condition. In the current study we decided to use properties of thin-walled tubes manufactured of high strength stainless steel. This material was selected because exposed to great bending, compression or extension deformation; it will be strong and can resist loads [15]. This type of steel is resistive for corrosion and has high cost-function efficiency.

In the study we will use the non-linear finite element software LS-DYNA to investigate the pre-deformation, which is an important factor to affect the energy absorption. The model of stainless steel thin-walled tube will be built and validated by crash test performed with trolley. The model will be used in the analysis of parameters that affect the energy absorption of stainless steel thin-walled tube.

2 Methods and materials

2.1 Energy absorption theory of the thin-walled tube

Take into account energy absorption; we can divide the thin-walled tube's deformation under axial load into three types: gradual progress of overlapping-shrink, Euler and mixed. The main difference of the three deformation types is the way to form plastic joint. During deformation the tube forms a lot of basic crush buckles so called salient. The first deformation model, which is most common one, can be result of both dynamic and static loading. When a tube is under dynamic load as in Figure 1, the displacement response is as presented in Figure 2[16]. We can see from this figure that the load-compression curve is oscillating. In fact every peak value in the curve corresponds to a crumble of the tube. Usually the crumble is formed successively from the end of the tube; it is so called gradual progress of overlapping-shrink deformation. The more material is compressed the better the material is used. According to [16] it was showed that the average force during the compression is constant. We can consider that it is reasonable when the preparative compression force is on the level of point 3.

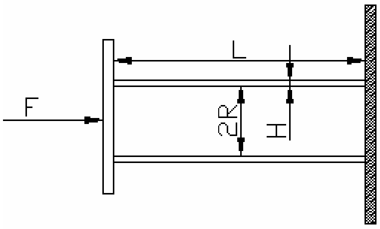
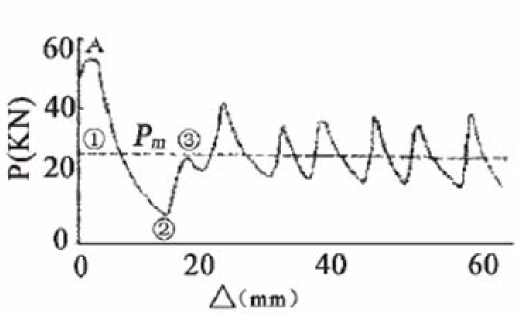


Figure 1. Thin walled tube under axial load



to [16].

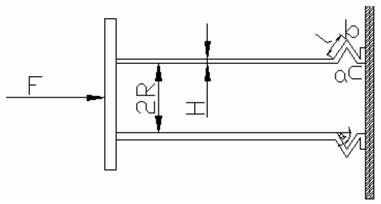


Figure 3. Deformation model under axial crash[16]

The energy-absorption during axial plastic loading when an integrated overlapping is created by joint a and c as in the Figure 3 is:

$$F_m 2l = E_1 + E_2 \tag{1}$$

where F_m is the average force, E_1 is the absorption energy of axial deformation, E_2 is the energy absorbed by circumferential deformation, and l is the length. E_1 can be expressed as:

$$E_1 = 4\pi M_0(\pi R + l) \tag{2}$$

where R is the radius of the tube, M_0 is bending moment of plastic deformation, and this is the maximum moment under Von Miscs' yield conditions. M_0 can be calculated as follows:

$$M_0 = \left(\frac{2\sigma_0}{\sqrt{3}}\right)\left(\frac{H^2}{4}\right) \tag{3}$$

where σ_0 is the yield limit of the material. H is the thickness of the tube.

At the same time, during formation of one integrated overlapping from Q to Q+dQ the energy absorbed by circumferential deformation is E_2.

$$E_2 = 2\pi\sigma_0 H l^2 \left(1 + \frac{1}{3R}\right) \tag{4}$$

All energy-absorption during formation of one integrated overlapping is E.

$$\sum E = E_1 + E_2 \tag{5}$$

So the power of the average force during this time is:

$$F_m 2l = E_1 + E_2 \tag{6}$$

From the above description we can see that the thickness, diameter, and parameters of the joints affect the energy absorption.

2.2 Construction and validation of the FE model

In order to investigate the energy absorption of the thin-walled tubes and to have a data set to validate the mathematical model, crash tests with a trolley were performed at the State Key Laboratory of Advanced Design and Manufacture for Vehicle Body. The test material was the stainless steel thin-walled tube, which is common in the market.

Due to the set-up used in the study, the length of the tube was 375 mm the outside diameter was 114 mm and the thickness 1.6 mm Figure 4 (a). As the mass of the trolley was 576 kg and the velocity was 35 km/h, the impact energy was large, so we could use two tubes in the test to absorb energy. To register the acceleration response of the trolley，the signal from one accelerometer attached to it was used. The trolley is shown in Figure 4 (b).

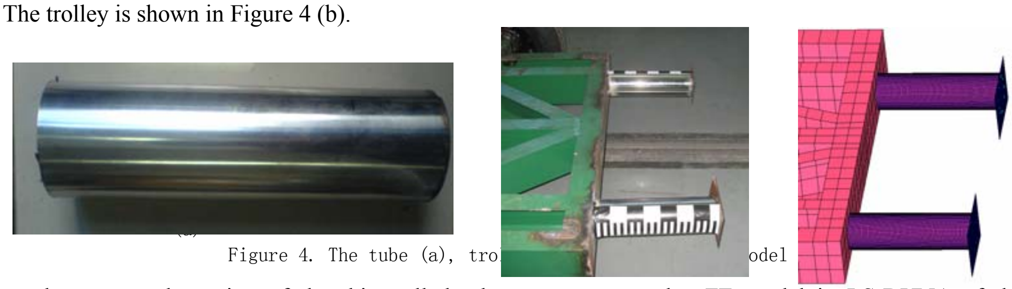


Figure 4. The tube (a), tro odel

To analyze the energy absorption of the thin-walled tube, we constructed a FE model in LS-DYNA of the trolley and thin-walled tube based on the test setup. The geometries, velocity and mass of the FE trolley was consistent with test setup. During the test the trolley wasn't deformed, so in the model we used rigid bodies to connect the parts of the trolley. Table 1 lists all the related conditions and properties of the material for the FE model.

Table1. Properties and impact conditions

Material Properties		Tube Geometry	
Young's modulus	207 GPa	Total length	375 mm
Yield Stress	250 MPa	Diameter	114 mm
Density	7830 kg/m3	Wall thickness	1.6 mm
Ultimate Stress	448 MPa	Impact conditions	
Hardening modulus	630 MPa	Initial velocity	9.7 m/s
Poisson's ratio	0.3	Sampling time	100 msec

The thin walled tube was modeled using shell elements of type quad4, which is interpreted as a quadrilateral element with four nodes, suitable for simulation of large deformations. After convergence an element size of 4 mm was found to produce appropriate results. The simulated thin walled tubes were fully fixed to the trolley using rigid body connection. The FE model has 21 744 elements, the thin-walled tube has 16 205 elements. The contact between the trolley and the thin-walled tube, the trolley and ground was defined. From the simulation we output the data for validation in form of acceleration that was filtered according to 60 SAE class filter and deformation of the tube, see Figure 5. The FE model is shown in Figure 4 (c).

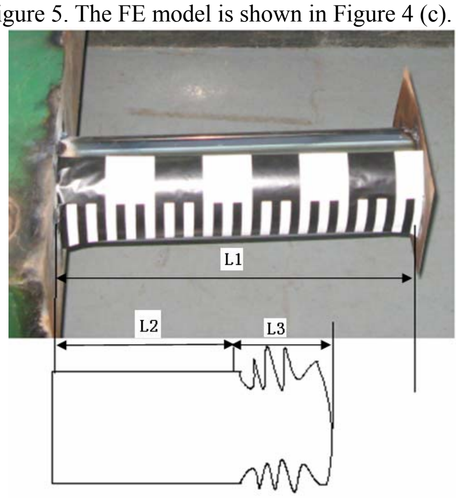


Figure 5. Measurement of tube deformation (L1- initial length, L2- uncompressed length, L3- remain compression length,)

In the validation of the model we compared the acceleration of the trolley in simulation and the test. We compared the deformation mode of integrated overlapping observed in the test with periods on acceleration curve under assumption that one buckle corresponds to one period on acceleration maximum. We also measured and calculated the remain compression length L3 and uncompressed length L2 of the tube. According to [17] if the difference between test and simulation is less than 20 % than we can accept the FE model as validated. Due to simple geometry of the model the target difference of validation in our study was set to 10%

2.3 The characteristic of energy absorption

To know all the characteristic of the energy absorption, we need to analyze in depth the process of joint creation. According to Jones et al. [10] during dynamic progressive buckling, on the FE model we can analyze the energy absorption by calculation of the displacements of certain points of the model. We can define the three key points of a joint as A1, B1, C1 in Figure 6, the time period to develop the first overlapping shrinkage. The following periods are defined the same way. In the study we decided to use this method to analyze the energy absorption. We also decided to compare displacements of points A1, B1, C1 during simulations.

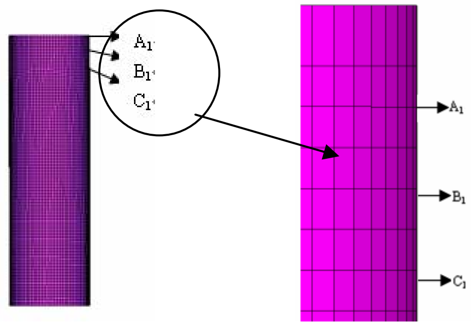


Figure 6. FE tube model

2.4 Parameter of pre-deformation

In the car manufacturing process the plates are formed to different shapes, especially for longitudinal beams or energy absorbing parts of car chassis. This form can be a trigger of buckling or other deformation types in the crush. In the study, to investigate the effect of the different shapes of the thin walled tube we used the pre-deformation with various number of crash trigger as is shown in Figure 7. We used 3, 2, 1 and 0 number of crush trigger in the simulations. The output was the acceleration response and amount of energy absorption as summarized in Table 2.

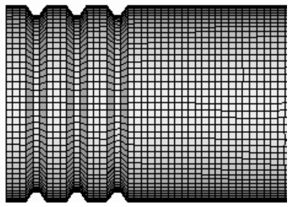


Figure 7. The FE model with three crush triggers

Table 2. Evaluation of the effect of crush trigger's number

Input	Output	
3 crush triggers 2 crush triggers 1 crush trigger 0 crush trigger	Acceleration response	Amount of energy absorbing

Table 3.Evaluation of the effect of crush trigger's geometry

Experiment	Depth [mm]	Width [mm]	Gap [mm]	Output
NO1	4 6 8	12	16	Amount of energy absorption
N02	6	12 16 20	16	
N03	6	16	12 16 20	

The next question is how the geometry of the crush triggers affects the characteristic of the thin walled tube? Designer of mechanical structures needs to know in the first hand the trend of parameters of the triggers. So we decided to investigate the trend of energy absorption in relation to depth, width and gap of the crush trigger. To investigate these parameters, we used a thin walled tube, which has three crush triggers. This thin walled tube was 375 mm, the outside diameter 114 mm and the thinness 1.6 mm. Based on laboratory experience, the depth was selected from 4 mm to 8 mm, because if this value is too big the energy absorption is not efficient, and if it is too small the influence on deformation is not obvious. The width of trigger has effect on the overlapping, which is usually one wavelength of the overlapping, so we varied it from 12-20 mm. The gap between two crush triggers was set from 16-20 mm.

3 Results

3.1 Validation results

Figure 8 shows the acceleration curve from test and simulation that showed the best agreement. The comparison of the deformations between test and simulation is showed in Figure 8 and Table 4 lists the value of these parameters.

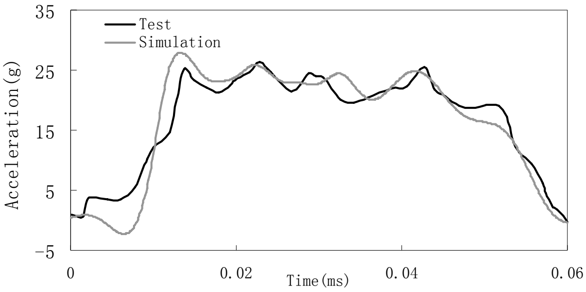


Figure 8. Sled acceleration in the test versus simulation

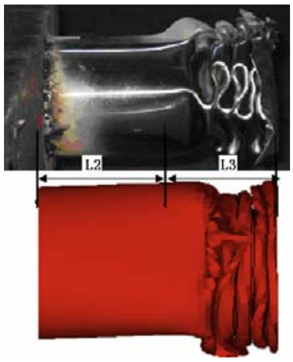


Figure 9. Deformation of test versus simulation

Table 4. Comparison of deformation

Parameters	Test value	Simulated value	Difference %
Uncompressed length L2 (mm)	121	130	7.4
Compress remain length L3 (mm)	108	99	8.3

From the Figure 8 we can see that time duration of the acceleration pulse in the simulation match very well that from the test. The levels of local maximum values are similar. From Figure 9 we can see that the deformation mode of thin walled tube in the test and simulation is also similar. The compressed remain length and uncompress length is nearly the same. There is four and a half of integrated overlapping that correspond to the four and a half periods of acceleration curve (Figure 8). The difference between measured values in the test and calculated from simulation, see Table 4, Are below the target values for current study.

According to the analysis above we can see that not only the acceleration response but also deformation mode, is fitting reasonable, the difference is below the target one, so we can say that the FE model is properly validated and reliable, and can be used in the parameters analysis of thin-walled tube.

3.2 Analysis of energy absorption characteristics

Figure 10 shows the displacement-time curve of points A1, B1, C1 defined according to Figure 4. During 0 to t_1 the process is elastic; the displacements of A1, B1, C1 are nearly the same. From t_1 to t_2 the displacement of A1 is large, but the displacement of B1 and C1 is smaller; this is the creation of first overlapping shrinkage. From t_2 to t_3 the displacement of B1 is large and this is the beginning of second one. At the same time the displacement of C1 is small, and A1 is also moving. Because point A1 is before B1, and the slope of A1 is the same as B1, we can say that there is none relative displacement between A1 and B1, and the energy is mostly absorbed by the overlapping of B1. The third overlapping will form when after t_3, the process of deformation is the same as above. From this analysis we can see that the energy absorption was mainly in the place of overlapping position. The rest of the tube shows only small displacements.

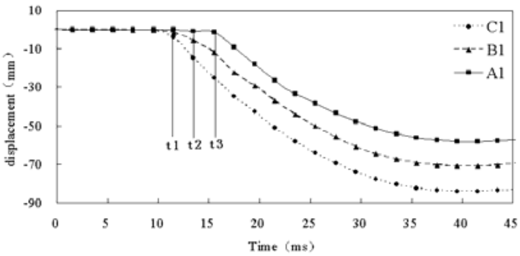
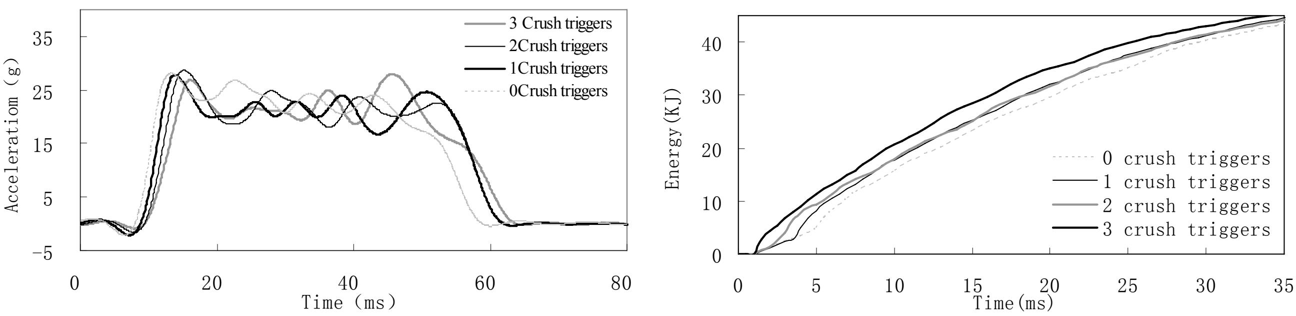


Figure 10. Displacement-time curves of A1, B1, C1

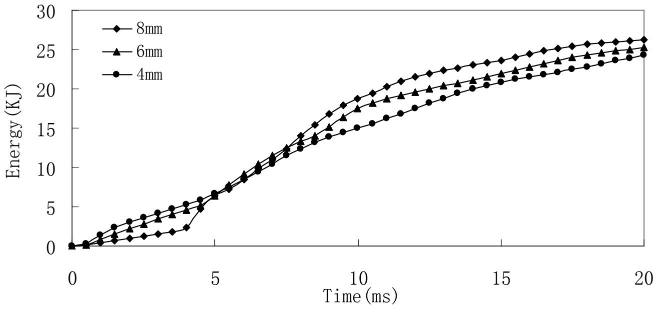
3.3 The effect of pre-deformation

The results of four kinds of thin walled tubes' are showed in Figure 11 a and b. If the number crush triggers increase the time duration of acceleration is also increased, but the changes of its first peak value are only small. The increase of number of crush trigger will increase the energy absorption, especially in the case of three triggers.

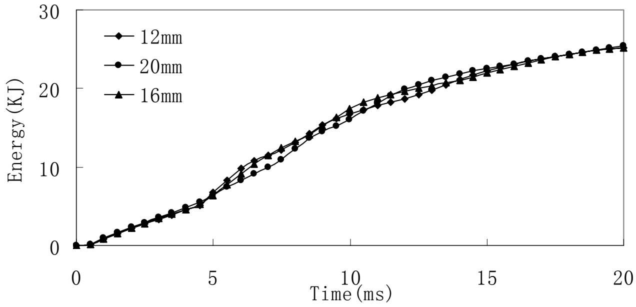


(a) (b)

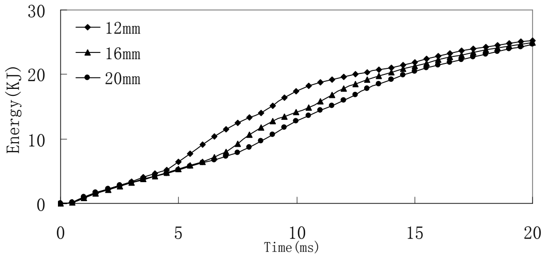
Figure 11 Comparison of acceleration (a) and energy-absorption (b) by number of crush trigger



(a) Depth effect on energy absorption



(b) Gap effect on energy absorption



(c) Width effect on energy absorption

Figure 12. The effect of trigger geometry on energy absorption

From Figure 12 we can see that if the depth of the trigger increase the energy absorption is increasing, but from 0 to 8 milliseconds the energy is nearly the same. If the trigger gap is changing the energy absorption remain at the same level. However the energy absorption is increasing with the width of the trigger.

4 Discussion

From analysis of energy absorption characteristic of thin walled tube we know that the overlapping position is important for energy absorption. It is easy to create overlapping in such structure partly due to its regular and symmetrical shape. So the thin-walled tube is an efficient energy-absorbing structure.

When the number of triggers is increased the duration of acceleration is also increased, but the number of triggers influence on its first peak value is small. Moreover if the number of triggers is three the influence is more clearly, it seems that the number of triggers has strong influence on the duration of acceleration. Sometimes we need to extend the duration of the acceleration, for example when considering passenger protection we want to decrease the peak value of the acceleration during the vehicle crash. Maybe we can use this idea regarding the number of triggers in car design.

We concluded previously that the increase of number of triggers on thin walled tube increases energy absorption. It is because the triggers form easily overlapping.

From the energy absorption theory [16] we know according to the Equation 6 that if the length (l) is increased the energy absorption will be also increased. If we make the triggers wider, at the same time the length (l) will also increase, the energy absorption will increase. This finding from the simulation is in agreement with the theory.

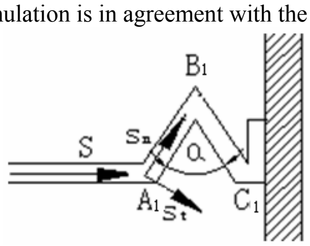


Figure 13. Force during overlapping

The force is an important parameter for energy absorption, so we need to discuss this force in simulation of the pre-deformed thin walled tube. Figure 13 shows the force state during overlapping where S is the impact force and it consists of two components: the pressure force (Sn) and shearing force (St).We can see if the angle α is decreased then St is increased and the energy absorption is also increased. As shown in Figure 12 (c) when we increase the depth it means decrease the angel α, we can see that results from our simulation study is consistent with the finding from literature [16].

The analysis is based on the introverted pre-deformation of trigger, but whether the trigger out forward and the parameters of this trigger have the same trend on energy absorption is not clearly. We should do more investigations about it. Moreover, in our study the analysis of the parameters of trigger is performed just by simulations, we should do some laboratory tests to validate them.

5 CONCLUSIONS

The paper describes the investigation of the axial crushing of the thin-walled tube, using a validated FE model. From the analysis of the parameters performed with FE model we found that: pre-deformation is important for thin-walled tube to absorb energy. To increase the energy absorption we can increase the depth of the trigger or change the width reasonable, however the gap between two triggers affects the energy absorption unremarkable.

The theory and energy absorption analysis of the thin-walled tube confirmed that the thin-walled tube can absorb energy efficiently, as it easy to be overlapped, so it can be used to absorb energy or assistant to absorb energy in engineer area.

Acknowledgements

The study is sponsored by the National “863 Program” (No. 2006AA110101) , the Ministry of Education of China “ the Program for Changjiang Scholars and Innovative Research Team in University(PCSIRT) ” (No. 531105050037) and the Autonomous subject Program of the State Key Laboratory of Advanced Design and Manufacturing for Vehicle Body, Hunan University (No. 60870004)

References

[1] Wierzbicki T., Recke L., Abramowicz W., Gholami T.. (1994). Stress Profiles in Thin-walled Prismatic Columns Subjected to Crush Loading in Compression. Computers & Structure, Vol.51,No6 611-632

[2] Abramowicz W., Wierzbicki W.. (1989).Axial Crushing of Multicorner Sheet Metal Column, Journal of Applied Mechanisms, 53.113 – 120

[3] Abramowicz W., Jones N.. (1986). Dynamic Progressive Buckling of Circular and Square Tubes. International Journal of Impact Engineering, Vol. 4, No. 4 243 –270

[4] Mamalis A., Manolakos D., Baldoukas A.. (1991).Energy Dissipation and Associated Failure Modes When Axially Loading Polygonal Thin-Walled Cylinder. Thin-Walled Structures, 12 17 – 34

[5] Liu Y.C. Day M.L..(2006). Simplified Modeling of Thin-Walled Box Section Beam. International Journal of Crashworthiness, Vol. 11, No. 3 263 – 272

[6] Kim H.S., Kang S.Y., Lee I.H., Park S.H., Han D.C.. (1997). Vehicle Frontal Crashworthiness Analysis by Simplified Structure Modeling using Nonlinear Spring and Beam Elements. International Journal of Crashworthiness, Vol. 2, No. 1 107–117

[7] Drazetic P., Markiewicz K., Ravalard Y.. (1993).Application of Kinematic Models to Compression and Bending in Simplified Crash Calculation. International Journal of Mechanical Science, Vol. 35, No. ¾ 179 – 19

[8] Mamalis A., Manolakos D., Loannidis M., Kostazos P., Dimitriou C.. (2003). Finite Element Simulation of the Axial Collapse of Metallic Thin-Walled Tubes with Octagonal Cross-Section. Thin-Walled Structures, 41 891 – 900

[9] Alexander J.L.. (1960).An approximate analysis of the collapse of thin cylindrical shells under axial loading. J Mech. Appl. Math, 13:10-5

[10] Karagiozova D.. (2001). Norman Jones Dynamic effects on buckling and energy absorption of cylindrical shells under axial impact. Thin-Walled Structures, Volume 39, Issue 7, Pages 583-610

[11] Langseth M., Hopperstad O.S., Berstad T.. (1999). Crashworthiness of aluminum extrusions: validation of numerical simulation, effect of mass ratio and impact velocity. International Journal of Impact Engineering, Volume 22, Issues 9-10, Pages 829-854

[12] Karagiozova D., Norman J.. (2004). Dynamic buckling of elastic–plastic square tubes under axial impact—II: structural response. International Journal of Impact Engineering, Volume 30, Issue 2, Pages 167-192

[13] Tai Y.S., Huang M.Y., Hu S.T.. (2009). Numerical Study in Axial Impact of Thin-walled Circular Tubes Made of High Strength Steel. Proceedings of the Nineteenth International Offshore and Polar Engineering Conference.

[14] Yu Z., Lei Z.B., Yang Z.. (2004). Simulative Design of Automobile’S Front Rails Based on the Pre-distortion Control Theory. College of Automobile and Mechanical Engineering，Changsha University of Science and Technology .Vo1．2 No.4.Pages.34-38(in Chinese)

[15] Jiang Z.G., Lin Z.G., Xu F.. (2004). Energy absorbing characteristic of soft steel and stainless tube under axial dynamic loads. Journal of Donghua University, Vo1. 3O, No. 5 16-20 (in Chinese)

[16] Jia D.Y., Jun R.J.. (2006). Energy-absorption of the end of cylindrical shell exposed to axial dynamic loads. Mechanical Management and Development (in Chinese)

[17] Wang C., Han Z.H., Yuan L., Liu S.. (2006) Simulation of Car Safety in Full scale Frontal Impact. Journal of Liaoning Institute of Technology. Vo1. 26, No. 4, Aug. (in Chinese)