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Theoretical prediction and numerical studies of expanding circular tubes as energy absorbers

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ABSTRACT

This work investigated the energy absorption behavior of expansion circular tubes. This kind of energy absorption device dissipates the impact of kinetic energy through plastic deformation and friction. A finite element model (FEM) was established with reference to the size of the coupler, which was used for connecting two vehicles. A special case was designed to validate the accuracy of the FEM. Furthermore, a theoretical prediction model that took into consideration the additional shear deformation and expansion ratio enlargement was compared with the steady compressional force of the experiment and numerical simulation. The theoretical prediction was accurate. Based on this valid FEM, a series of parameter studies was developed. Specifically, a list of conical mandrels with different angles from 5° to 40° was established, and the coefficient of friction varied from 0 to 0.3. Both had great influence on energy absorption. The relationship between the expansion angle and steady compressional force is nonlinear, while the friction coefficient and steady compressional force are linear. With friction, the inflection point of force curve existed between 10° and 20°.

1. Introduction

Tube type energy absorbers dissipate impact energy through plastic deformation; the conversation of energy is irreversible [1]. Therefore, this process has been widely used over the past decades due to its successful performance in the field of passive vehicle safety. Various deformation patterns of these circle tubes have been carried out, under both axial and lateral loading [2]. Numerous studies on square and circular tubes were developed: tapered thin-wall square tubes for axial loading, nested circular tubes for lateral loading, circular tubes for free inversion, tapered circular tubes for axial crushing and thin walled aluminum tubes for expansion [3–7]. Furthermore, filler-filled tubes and combined structures that performed well were designed by engineers: foam-filled thin-walled tubes for static and dynamic loading, multi-cell square tubes for oblique loading, bitubal circular tubes for bending collapses [8–10].

In recent years, the expansion of circular tubes has drawn more and more attention in the field of energy-absorbing unit design. Compared to folding deformation, the deformation of expanding circular tubes is much more stable and the loading force is smoother. Many researchers have studied the influence of various parameters on the expanding tube process [11–14]. It has been proven by a mean of experiments and numerical simulations that the expansion ratio, expansion angle, thickness, and friction greatly influence energy absorption. Choi also has investigated the influence of impact velocity on energy absorption [15]. However, most of the level of the loading force is no greater than 10 t. Some experts also have proposed a theoretical model to predict deformation resistance [16–18]. However, these were all under the condition of a small expansion angle. Different boundary conditions and expansion purposes would result in varying prediction models. Based on the main stress analysis method, article [19] considered the kinematics and material flow, and developed a mathematical model to predict the drawing force required for expansion. The expandable tubes used for drilling and well repair were both very long. Then the expansion portion had a sharply contrasted stress–strain state [20]. The studies above all provided valuable experiences for this paper.

In this paper, we aimed to devise a structure appropriate for a railway vehicle. In light of the huge energy absorption requirements and the installation space, the steady compressional force of the tube expansion should be several times that of the others.

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Consequently, the expansion angle is usually large. However, Kopp pointed out that in the process of metal bulk forming, additional shear stress could not be neglected when the curvature radius of a circular tube changes [21]. Similar to tube-drawing, the drawing force was divided into three parts to overcome the resistances below – ideal deformation, additional shear deformation, and friction. In addition, because of the existence of big cone angle, the radial velocity component of the circular tube could not be rapidly reduced to zero when it moved out of the expansion zone.

Under this condition, a theoretical analysis model that considered the additional shear and expansion ratio enlargement was proposed to predict the deformation resistance. Furthermore, a finite element model was built and validated by the experiment. Finally, the effects of the main parameters were also studied based on the finite element model (FEM). A comparison of the deformation resistance was made between theoretical analysis and FEM, which agreed well with each other.

2. Finite element modeling

A finite element model of the tube expansion was modeled using HyperMesh. In order better simulate metal flow during deformation, 8 node 3D-brick element was adopted. Additionally, the thickness of the circular tube was divided into four units. This model is shown as Fig. 1. The shape and dimensions of the devices were in reference to the size of the coupler, which was used for connecting two vehicles. The initial input parameters of the model are listed in Table 1. There was a primary clearance of 1 mm between the circular tube and the mandrel. With respect to the large deformation, the material of the circular tube was considered plastic when the conical mandrel was rigid. The stress–strain data gained from the tensile test is shown in Table 2.

The bottom of deformable tube was fixed. A rigid wall with an initial speed struck the rigid mandrel and expanded the deformable tube. According to the scenario, the impact speed was set as 25 km/h. However, in order to compare the results with that of experiment, the speed was finally 6.39 m/s (23 km/h). This kind of energy absorption device dissipates the impact kinetic energy through plastic deformation as well as friction. Hence, an appropriate lubricant was added, which not only avoids adhesive wear but also dissipates the maximum kinetic energy. The friction

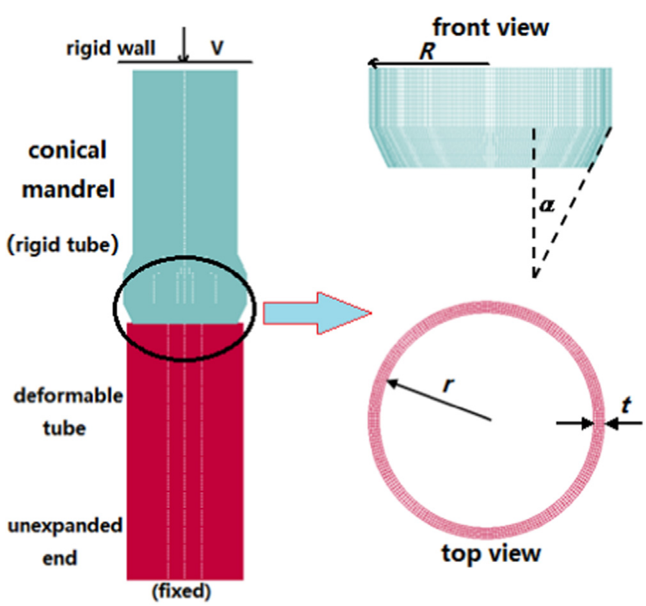

Fig. 1. Finite element model for analysis.

Table 1
Finite element model input parameters.

Part	Parameters	Value
Deformable tube	Inner radial r (mm)	72
	Thickness t (mm)	7
	Section length l (mm)	350
	Expansion angle α (deg)	32
	Density (kg/m^3)	7800
	Young modulus (MPa)	206,000
	Yield strength (MPa)	245
	Poisson ratio	0.3
	Stress–strain date	Table 2
	Coefficient of friction μ	0.14
Conical mandrel	Maximum outer radial R (mm)	84

Table 2
True stress–strain relationships for the deformable tube material.

ε	0.083	0.123	0.168	0.215	0.264	0.312
σ/MPa	401.5	477.4	532.4	573.1	605.0	629.2

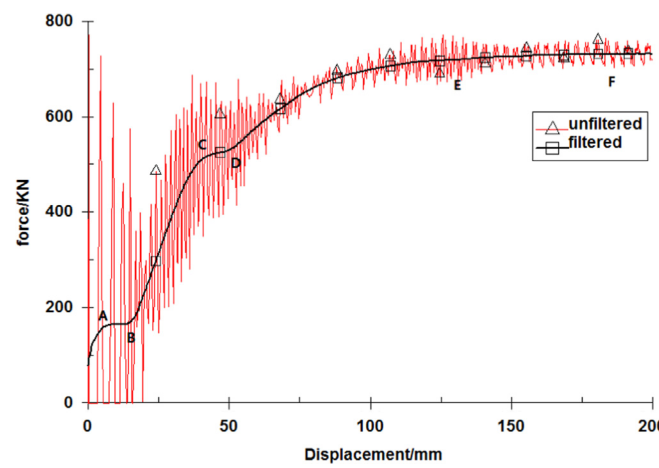

Fig. 2. Force–displacement of numerical simulation. (For interpretation of the references to color in this figure, the reader is referred to the web version of this article)

coefficient was gained from the experiment. A surface-to-surface contact algorithm was defined in the model, and the contact force can be obtained from the output file.

The force–displacement curve acquired from the output file is shown in Fig. 2. The red curve was unfiltered, while the black curve was filtered in the SAE channel with a frequency of 180. It can be seen from the figure that the peak force had been filtered. However, the growth process would be clearer. The whole process of the circular expansion tube can be learned as flowing.

In the present system, the deformation process can be divided into six parts. The first stage starts when the tube gets in touch with the mandrel, and bends. The axial load then rises to point A. Second, the tube bends until the end tube's surface is vertical in relation to the conical surface of the mandrel. During this time, the contact surfaces between the tube and the mandrel increase and the rate of the rising force decreases. Following this, the tube shifts into a state of uniform expansion along the conical surface of the mandrel. As a result, the load rises sharply and arrives at point C. Because of inertia, the free end of the tube still has radial velocity, which causes the gap between the tube and mandrel. Subsequently, the tube falls a little back to the mandrel. The load force remains almost unchanged, and in the end, reverse bending carries on. The force increases slowly until the radial velocity of the