Dynamic Response and Energy Absorption Characteristics of Expansion Tubes Under Axial Impact

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This work was supported in part by the Natural Science Foundation of China under Grant 11672329, and in part by the Foundation of National University of Defense Technology.

ABSTRACT
The expansion tube is normally used in impact-resistant components, owing to the stable expansion force, and high specific energy-absorption capacity. This paper reports the deformation mechanism and energy absorption characteristics of expansion tube subject to impact velocity ranging from 0.083 m/s to 48.84 m/s. The effects of cone piston semi angle, tube wall thickness, and impact velocity on expansion tube performance were studied via experimental measurements and numerical modeling. The mechanical responses of the expansion tube were measured using a universal gas gun setup. The energy absorption capacity of the expansion tube absorber was identified to be different under quasi-static and dynamic compressions. The dynamic expansion force is lower than the quasi-static expansion force, which is about 62.2%–76.6% of the static expansion force, but the deformation mechanisms of the tube under quasi-static loading and dynamic impact are same. A finite element numerical model was built and validated with the experimental data. The finite element predictions were in good agreement with the experimental measurements. It was shown that the decrease in the friction coefficient is the main reason for the dynamic expansion force lower than the quasi-static expansion force. The influence of cone piston semi angle and tube wall thickness are significant on the energy absorption capacity of the tube. The dynamic expansion response does not change significantly when the impact velocity is less than 50 m/s. Under dynamic impact, the change of energy absorption efficiency is negligible, and the plastic deformation energy is about 64%–71.5% of the total kinetic energy of striker.

INDEX TERMS
Dynamic impact, energy absorption characteristics, expansion response, expansion tubes, friction coefficient.

I. INTRODUCTION
As the rapid increase in the number of vehicles leads to an increase in traffic accidents, many kinds of energy absorption components are employed in vehicles to lessen the potential danger of impact accidents [1], [2]. Lightweight thin-walled structures under axial compression are widely used as energy absorbers which have drawn much attention due to their good energy absorption characteristics [3]–[5]. Most of the metal energy absorbing components for which the main energy-dissipating mechanisms are plastic deformation, fracture and tearing [6]; the conversation of energy is irreversible [7]. Different from conventional energy absorption structures, the expansion tube has long-stroke, stable deformation mode and high specific energy-absorption capacity [7], [8]. Moreover, the expansion force of the tube is controllable and stable [9]–[11].

The schematic diagram of the expansion tube is shown in Fig. 1. When a load is transmitted to the component, the cone piston is pressed into the expandable tubes and the radius of tubes is greater than that of the origin tube. After the loading process, the circular tube will be expanded into a coaxial tube with a larger final radius [12]. The loading energy will be dissipated by the tubes’ plastic deformation and friction between the tubes’ inner wall and the cone piston surface.

The previous researchers studied the deformation mechanisms and the energy absorption characteristics of expandable
tube absorbers by means of theoretical analysis, experiment and simulation. It has been proven that the expansion ratio, semi angle of cone piston, wall thickness of the tube, and friction considerably influence the energy absorption capacity of expandable tube. Lu [13] investigated the process of tube expansion and presented theoretical expressions for the relationship among the geometrical parameters, the stroke of cone piston and the deformation at the tube end under quasi-static loading condition. Fischer et al. [14] proposed an analytical model to calculate the expansion force as a function of the displacement of the cone piston. In this model, the wall thickness within the deformed region of the tube was assumed to be distributed linearly, and the effects of strain hardening and friction were neglected. Seibi et al. [15] derived a theoretical model to predict the steady stage of the expansion under quasi-static loading condition. In their model, the friction and thickness variation were considered, the strain hardening was neglected, and the contact pressure was also assumed to be uniformly distributed at the interface. Liu and Qiu [12] proposed a new deformation theory, which considered the effects of strain-hardening and friction, to predict the steady expansion force. They concluded that the plastic deformation and friction are equally important in dissipating impact energy during the process of tube expansion. Almeida et al. [16] studied the deformation mechanism of expansion tubes by changing the geometrical parameters and friction through experiments and finite element simulation under quasi-static loading condition. They observed ductile fracture, local buckling and wrinkling in the tests, and concluded that the interfacial friction was the key role in determining arise of instability of tube. By means of a comprehensive numerical and experimental investigation under quasi-static loading condition, Yang et al. [11] studied the expansion process; the contact force was found to rise with oscillation and reach a steady-state for long stroke. Depending on geometrical parameters, they classified the expansion process into three modes by the contacting situation. They also discussed the specific energy absorption capacity, which was found to be higher for a larger semi angle. To investigate the effects of the semi angle of cone piston on energy absorption characteristics of tubes, Choi et al. [17] conducted quasi-static experiments with three types of semi angle and carried out a finite element analysis under the same conditions as those of the experiments. They found that the semi angle had a significant effect on the frictional coefficient and the plastic deformation energy of the tube, and on the frictional energy and expansion ratio. Choi et al. [18] also investigated the influence of impact velocity on energy absorption less than 10 m/s. It is observed that energy absorbed into the tube was increased due to the strain rate effect of the tube under the dynamic impact. They used the least square method to calculate the friction coefficient and found that the friction coefficient in quasi-static state was higher than that in dynamic state.

Expansion tube is normally used as an energy absorption component under dynamic impact, such as buffer of separating device of spacecraft or striking prevention device of the train, and it is loaded at impact velocities below 50 m/s. The force is therefore considered only due to the negligible effect of temperature and fluid flow in this range of impact velocity. Compared to the quasi-static loading conditions, the expansion process would be affected by more factors under the dynamic impact conditions, such as the strain rate effect of parent materials and friction coefficients between surfaces of tube and cone piston. However, the dynamic response of the expandable tube has been rarely investigated. Previous studies were all under the condition of quasi-static load or low-speed impact (less than 10 m/s). Hence, in order to fill the research gap, the current study is carried out to research the deformation mechanism and the energy absorption characteristics of expansion tube absorbers under the dynamic impact (impact velocity ranging from 15 m/s to 50 m/s), and provide guidance for the design of this energy absorbers.

II. EXPERIMENTS
A. EXPANSION TUBE SPECIMEN
In this study, the inner diameter of expansion tubes is designed to be fixed, i.e. $D_0 = 30$ mm, and the wall thicknesses ($h$) of expansion tubes are ranged from 0.8 mm to 2.0 mm. All the expansion tubes have a length (L) of 50 mm. As shown in Fig. 1, the cone piston had three parts: guide rail, variable-sectional expanded (VSE) area and fixed-sectional expanded (FSE) area. The guide area is designed to guide the cone piston, and the diameter of the guide area is equal to the inner diameter of expansion tubes. The VSE area is cone-shaped with three different semi angles ($\gamma$): 5.71°, 16.70°, 26.57°. The height of every part of cone piston is fixed, i.e.
TABLE 1. Summary of experimental results for expansion tubes and cone pistons.

| Test no. | \( \gamma \) (deg) | \( D_1 \) (mm) | \( h \) (mm) | \( V \) (m/s) | \( \text{Smax} \) (mm) | \( F_{\text{p}} \) (N) | \( F_{\text{n}} \) (N) | \( \text{Fmax} \) (N) | \( W_m \) (kJ/kg) | I |
|----------|---------------------|---------------|--------------|-------------|-----------------|----------------|----------------|----------------|----------------|---|
| C1T2-1   | 5.71               | 31            | 1.4          | 0.083       | 33              | 3855           | 3649           | 4421           | 9.61           | 0.83 |
| C1T2-2   |                    |               |              | 15.06       | 16.77           | 2399           | 2622           | 5314           | 6.91           | 0.49 |
| C1T2-3   |                    |               |              | 16.75       | 19.08           | 2593           | 2906           | 6627           | 7.66           | 0.48 |
| C1T2-4   |                    |               |              | 18.52       | 22.08           | 2675           | 2871           | 5689           | 5.56           | 0.50 |
| C1T2-5   |                    |               |              | 20.06       | 25.15           | 2665           | 2912           | 5567           | 7.67           | 0.52 |
| C2T2-1   | 16.70              | 33            | 1.4          | 0.083       | 33              | 11704          | 10385          | 11918          | 27.56          | 0.87 |
| C2T2-2   |                    |               |              | 22.86       | 14.76           | -              | 6005           | 8420           | 15.82          | 0.71 |
| C2T2-3   |                    |               |              | 27.50       | 16.69           | 8261           | 7046           | 9629           | 18.56          | 0.73 |
| C2T2-4   |                    |               |              | 31.55       | 25.20           | 8020           | 7069           | 10135          | 18.62          | 0.70 |
| C2T2-5   |                    |               |              | 34.12       | 28.81           | 8074           | 7159           | 9819           | 18.86          | 0.73 |
| C3T2-1   | 26.57              | 35            | 1.4          | 0.083       | 6.93            | Fracture       | 7593           | 12101          | 20.0           | 0.63 |
| C3T2-2   |                    |               |              | 33.95       | 18.28           | -              | 10789          | 14078          | 28.42          | 0.77 |
| C3T2-3   |                    |               |              | 37.04       | 21.03           | 13404          | 11435          | 15799          | 30.12          | 0.72 |
| C3T2-4   |                    |               |              | 39.50       | 24.09           | 13652          | 11298          | 15486          | 29.76          | 0.73 |
| C3T2-5   |                    |               |              | 42.32       | 27.24           | 13484          | 12010          | 16920          | 31.63          | 0.71 |
| C4T2-1   | 16.70              | 33            | 0.8          | 0.083       | 33              | 5818           | 5272           | 5968           | 24.78          | 0.88 |
| C4T2-2   |                    |               |              | 16.73       | 13.41           | /              | 3231           | 4730           | 15.19          | 0.68 |
| C4T2-3   |                    |               |              | 19.34       | 19.06           | 3738           | 3292           | 4582           | 15.47          | 0.72 |
| C4T2-4   |                    |               |              | 22.71       | 23.46           | 4301           | 3763           | 5319           | 16.69          | 0.71 |
| C4T2-5   | 16.70              | 33            | 2.0          | 0.083       | 33              | 19289          | 16544          | 19893          | 29.94          | 0.83 |
| C4T3-2   |                    |               |              | 33.18       | 18.73           | /              | 11924          | 16031          | 21.58          | 0.74 |
| C4T3-3   |                    |               |              | 37.99       | 24.76           | 14223          | 12449          | 17078          | 22.53          | 0.74 |
| C4T3-4   |                    |               |              | 42.06       | 27.65           | 14712          | 12619          | 17031          | 22.83          | 0.74 |
| C4T3-5   |                    |               |              | 48.84       | 26.16           | 14778          | 13082          | 18471          | 23.67          | 0.71 |

S_1 = 10 \text{ mm}, S_2 = 5 \text{ mm}, S_3 = 5 \text{ mm}, \text{ see Fig. 1}. The diameter of the cone piston’s FSE area is marked as D_1. The cylindrical steel striker has a length and diameter of 100 \text{ mm} and 28 \text{ mm} respectively. The velocity of the striker is marked as \( v \). Detailed dimensions of all the specimens are listed in Table 1.

B. MECHANICAL PROPERTIES OF PARENT MATERIAL
The tubes tested were made of 6061 rust-proof aluminum alloy, which was heat treated. To identify the mechanical characteristics of expansion tubes, the standard static tensile tests were conducted on the parent material at room temperature at a constant strain rate of \( 3.3 \times 10^{-3} \text{ s}^{-1} \) using a WDW-500E universal test machine (UTM), as shown in Fig. 2(a). The dynamic tensile tests on parent material were conducted at room temperature at strain rates ranging from \( 6.0 \times 10^{-2} \text{ s}^{-1} \) to \( 2.5 \times 10^{-3} \text{ s}^{-1} \) using Split Hopkinson Tension Bar (SHTB) apparatus, as shown in Fig. 2(b).

Based on the repeat tests, the stress-strain curves of the parent material at each strain rate were similar, and their average value at each strain rate is shown in Fig. 3. According to the stress-strain curve shown in Fig. 3(a), the 0.2% proof stress \( \sigma_{0.2} \) of 316.93 MPa, ultimate stress \( \sigma_u \) of 383.63 MPa, and elastic modulus \( E \) of 71.3 GPa in the quasi-static tensile test can be obtained. Comparing the stress-strain curves at different strain rates shown in Fig. 3(b), it can be concluded that the tensile properties of this aluminum alloy are not sensitive to the strain rate.

The cone pistons were made from T10 carbon steel (T-10) with an elastic modulus of \( E_s = 210 \text{ GPa} \) and yield stress of \( \sigma_s = 990 \text{ MPa} \), which are much higher than the mechanical properties of the tubes. Hence, cone pistons can be regarded as rigid.

C. TEST SETUPS FOR EXPANSION TUBES
Quasi-static and dynamic experiments were implemented to identify the effects of impact velocity on the dynamic response of the tube. The quasi-static axial diameter-expansion tests were carried on a WDW-500E UTM. Before the test, the tube was adjusted to be aligned with the tube, cone piston, and test machine. The molybdenum disulfide grease, which can not only avoid adhesive wear but also dissipates the kinetic energy of strikers, was used between the tube and cone piston. The steel striker was loaded by cross-head of the UTM at a constant speed of 0.083 m/s, then the cone piston was pressed into the tube. The load-displacement curves are recorded by the test machine. Besides, the expansion process was recorded by a camera.

The dynamic experimental setup was performed using a gas gun, as shown in Fig. 4. The tube specimen was inserted onto the circular table, and fast glue was used to adhere to the tube and circular table together. The circular table was fixed to the stationary base through the internal thread, and the pressure transducer (KD3020) was used to detect the pressure, see Fig. 4(b). The pressure history at the bottom of the tube was recorded as a voltage change, and was amplified by a KD5009 signal conditioning amplifier system and then output onto a Tektronix DPO4054 500 MHz Digital Phosphor Oscilloscope at a sampling rate of 10 MHz. A high-speed camera was used to record how the tube was expanded during the entire dynamic impact. Typically, the frame rate and exposure time were 50,000 fps and 20 \( \mu \text{s} \), respectively. The striker with different velocities \( v \) (15–50 m/s) fired by gas gun impacted the cone piston firstly, and then the cone piston was pressed into the tube. Molybdenum disulfide grease was also used between the tube and the cone piston in the dynamic impact.
D. EXPERIMENTAL RESULTS

Fig. 5 shows the response of the expansion tube specimen \((\gamma = 16.70^\circ, h = 1.4 \text{ mm})\) expansion during a quasi-static test, and Fig. 6 shows the response of the expansion tube specimen \((\gamma = 16.70^\circ, h = 1.4 \text{ mm}, v = 34.12 \text{ m/s})\) expansion during a dynamic test. The experimental results for all specimens have been listed in Table 1, where \(S_{\text{max}}\) represents the displacement of the cone piston after tests, \(F_P\) represents the stable-state expansion force, \(F_S\) represents the average expansion force, \(F_{\text{max}}\) represents the maximum expansion force of the entire energy absorption process, and the blank means that the stable-state has not been achieved.

To evaluate the effect of structural parameters and impact velocity, a series of tests on tubes with different semi angles \(\gamma\), tube thickness \(h\) and impact velocities \(v\) of strikers were conducted. The expansion force-displacement curves, obtained from the tests are shown in Fig. 7. It indicates that the expansion force increases significantly with the increasing of cone piston semi angle and tube wall thickness. Overall, the expansion process can be divided into two phases both in quasi-static tests and dynamic tests, i.e. oscillation-stage and stable-stage, corresponding to “I” and “II” in Fig. 7(b), respectively. The stable-state stage occurred earlier in dynamic tests than that in quasi-static tests, and the tubes with thinner wall thickness, i.e. \(h = 0.8 \text{ mm}\), the stage “II” occurred earlier. As for tests with small semi angle of cone piston or thick wall thickness of the tube, i.e. the cases with \(\gamma = 5.71^\circ\) or \(h = 2.0 \text{ mm}\), the fluctuations are significant in the initial raising stage. Although the loading velocity decreases gradually during impact which is an unavoidable phenomenon and happens in the real engineering
applications, the elastic stage and stable crushing stage of expansion tube would not be affected by the decreases of loading velocity significantly. Comparing with the quasi-static tests, the stable-state expansion force in dynamic tests is lower, approximately 62.2%–76.6%. The difference of the expansion force in quasi-static tests and dynamic tests will be discussed in detail in section IV-A. As shown in Figs. 7 and 8, the stable-state expansion force $F_P$ at different impact velocities in dynamic tests is stable, demonstrating that the performance of this buffer is no sensitive to velocity when the impact velocities are between 15 m/s and 50 m/s.

It should be noted that except for the tube of C3T2-1, other tubes used in experiments were expanded uniformly, and no fracture, local buckling or wrinkling was observed in the tests. It was measured that when the tube C3T2-1 gave rise to fracture (see Fig. 9), the displacement of the cone piston corresponded to 6.93 mm and the tensile strain in the circumferential direction of tubes equal to 17.8%. Recall the stress-strain curve of the aluminum alloy material in Fig. 2, the tensile fracture strain of this tube is consistent with the fracture strain of parent material in quasi-static tensile tests, which can be concluded that the fracture of tubes in the quasi-static test is governed by the fracture tensile strain of parent material.

III. FINITE ELEMENT SIMULATION

A. FINITE ELEMENT MODEL

Numerical simulations were conducted to simulate the quasi-static and dynamic mechanical response and of the
expansion tubes using the commercial finite element package ABAQUS [19]. The primary aims of the numerical investigation are [20]–[22]: (1) develop an accurate three-dimensional finite element model to predict the mechanical response and energy absorption characteristics of expansion tubes under quasi-static and dynamic loading; (2) understand the effects of the key parameters on the mechanical response and energy absorption characteristics of the expansion tubes.

Owing to the symmetry of the structure, a quarter model was created to perform the finite element analysis, as shown in Fig. 10. The tube, cone piston, and striker were modeled with 8-node 3D linear solid elements (C3D8R in ABAQUS...
The constitutive model of aluminum adopted in the simulation is elastoplastic material model which is based upon the von Mises isotropic plastic algorithm. The material mechanical properties (0.2% proof stress $\sigma_{0.2}$ of 316.93 MPa, ultimate stress $\sigma_u$ of 383.63 MPa, Young’s modulus $E$ of 71.3 GPa, density $\rho$ of 2.7 g/cm$^3$, Poisson’s ratio $\nu$ of 0.33) input were obtained based on the test results of parent material shown in section II-B. The plastic hardening model based upon Johnson-cook algorithm ($C_0 = 0.0118$, $\dot{\varepsilon}_0 = 0.0033$ s$^{-1}$) was adopted for aluminum. $C_0$ is the material constant that was obtained via fitting the data of dynamic material testing of the tube, and $\dot{\varepsilon}_0$ is the strain rate of the material specimen in the quasi-static material testing. To model the cracking failure (shear band localization) of aluminum tubes, the shear damage criterion was adopted and the fracture strain $\varepsilon_f$ was set to be 0.17. Also, the damage initiation was dependent on the displacement and the displacement at failure $S_P$ was set to be 0.095 mm. The parameters of material constitutive model for expansion tube have been listed in Table 2.

To determine the element size of the tube, the mesh sensitivity analysis was carried out. As shown in Fig. 11, the number of elements in the thickness direction of the tube has a significant effect on the expansion force. It is demonstrated that 5 elements (named as n = 5 in Fig. 11) in the thickness direction of the tube were adequate to achieve converged results. The mechanical behavior of parent material is modeled with an elastic-plastic material model using data from Fig. 3, which is based on the tension test. A total of 11,000 elements and 12,000 nodes meshed for cone piston, and 31,000 elements and 33,000 nodes meshed for the striker.

For the quasi-static expansion simulation, the kinetic energy was controlled to be under 5% of the total energy in the system, and the friction coefficient between the cone pistons and expansion tubes is considered to be 0.17. For the dynamic expansion simulation, the strikers moved at an initial impact velocity that were identical to those of the strikers employed in the impact tests, and the friction coefficient between the cone pistons and expansion tubes is considered to be 0.02. In addition, this value is assumed to be 0.17 for the contacting surfaces between the expansion tubes and the rigid plate. The reason for the difference of friction coefficients in quasi-static tests and dynamic tests will be discussed.

### B. SIMULATION RESULTS

Fig. 12 shows the predicted deformation process and the stress field distribution of the expansion tube C2T2-5 ($\gamma$ = 16.70°, $h$ = 1.4 mm, $v$ = 34.12 m/s) at different stages. The deformation process can be divided into three stages, which are I, II and III. When the cone piston starts to widen the tube, the wall of the tube spread outward and the expansion force starts to increase (stage I). The increase in expansion force is caused by the continuing expansion of the tube and increasing of the contact area. Next, the expansion force reaches a level where the radial displacement of the tube leading edge reduces; then, with the increasing of cone piston displacement, the expansion force increases again and reaches the stable state (stage II). Finally, the tube is expanded in a steady model, and the expansion force develops at a steady level (stage III).

Figs. 13 and 14 show numerically predicted expansion force as a function of displacement of cone pistons at different impact velocities. Comparing the expansion force-displacement curves, it is found that although the values of expansion forces are different in quasi-static and dynamic loading conditions, the deformation process of tubes and

| $\sigma_{0.2}$ (MPa) | $\sigma_u$ (MPa) | $E$ (GPa) | $\rho$ (g/cm$^3$) | $\nu$ | $C_0$ | $\dot{\varepsilon}_0$ (s$^{-1}$) | $\varepsilon_f$ | $S_P$ (mm) |
|----------------------|-----------------|-----------|-----------------|-----|------|-----------------|-------------|----------|
| 316.93               | 383.63          | 71.3      | 2.7             | 0.33| 0.0118| 0.0033 s$^{-1}$ | 0.017       | 0.095    |

**TABLE 2. Parameters of material constitutive model for expansion tube.**
IV. DISCUSSION

Fig. 15 shows the experimentally measured and numerically predicted expansion force-displacement relationships of tubes under different loading conditions. It can be seen that the numerical expansion force-displacement curves show fluctuations with small amplitudes, but the experimental expansion force-displacement curves illustrate fluctuations with large amplitudes. This is because the loading conditions and constraint boundary conditions are both completely ideal cases in the numerical simulation. However, the experimental measurements are affected by various factors, for example, the striker is not completely normal incidence due to its own weight, and the expansion tube and the base are not completely consolidated. These factors will cause structural vibrations, thus increasing the fluctuations of the expansion force-displacement curves. In addition, the surfaces of the tube, as well as cone piston, are not absolutely smooth, which also lead to the fluctuation. However, the fluctuations of the experimental expansion force-displacement curves are within the acceptable range, and the characteristics of the numerical and the experimental expansion force-displacement curves are very similar. Therefore, the numerical simulation results are considered to be valid. Table 3 summarizes the comparison of the finite element analysis results with the experimental data in terms of the stable-state expansion force $F_P$.

In some tests, the initial kinetic energy of the striker is not sufficient to cause cone piston to have a large embedded displacement, which prevents the expansion force rising to the stable-state expansion force, for example, the test of $\gamma = 16.70^\circ$, $h = 0.8$ mm, $v = 22.71$ m/s. In addition, in the test of $\gamma = 26.57^\circ$, $h = 1.4$ mm, $v = 0.083$ m/s, the stable-state expansion force was not achieved due to the fracture of the tube, as discussed in Section II-D. In this table, ‘Diff’ represents the difference between simulated $F_P$ and measured $F_P$. According to the comparison, the differences for the stable-state expansion force $F_P$ were approximately 0.06%–10.41%, reflecting that the simulated results are in good agreement with the experimental results.

The dynamic response and energy absorption characteristics of expansion tubes will be discussed from the following five aspects: friction, semi angle and wall thickness, impact velocity, specific energy absorption capacity, and energy absorption efficiency.

the trend of expansion force-displacement curves are similar. It is observed that in the simulation with lower semi angle or thinner wall thickness, i.e. $\gamma = 5.71^\circ$ or $h = 0.8$ mm, stage III occurs earlier. Also, in dynamic tests of initial impact velocity ranging from 15 m/s to 50 m/s, the expansion force-displacement curves of the tube with the same structural parameter are coincident, which shows the influence of impact velocities on the steady-state expansion force of expansion tube is negligible.
A. FRICTION
The previous researches point out that under the same surface lubrication condition, the quasi-static friction coefficient of the tube is significantly lower than the dynamic friction coefficient of the tube [24]–[26]. The reason is due to the heating at the slide-contact interface under dynamic impact, which would cause thermal softening at the slide-contact interface and the reduction of friction coefficient. Based on the numerical calibration against tests, the friction coefficient at the interface under quasi-static loading condition is set to be 0.17 [23]. In the simulation of dynamic impact loading, the friction coefficient at the interface is set to be 0.02, which is consistent with the range of 0.02–0.05 reported by Choi et al. [18]. In addition, higher friction coefficient results in higher expansion force and lower friction coefficient results in lower expansion force. Recall section II-B, the strain rate sensitivity of parent material is negligible. Hence, it can be concluded that different steady-state expansion forces under different loading conditions stem from different friction coefficients.

B. AVERAGE CRUSHING FORCE $F_S$
The average crushing force $F_S$ of the expansion tube is an important indicator to evaluate the energy absorption characteristics, and it can be calculated as follows:

$$F_S = \frac{W}{S_{\text{max}}} = \int_0^{S_{\text{max}}} F(S) dS$$

where $S_{\text{max}}$ is the maximum displacement of cone pistons, which has been summarized in Table 1. $W$ is the energy absorbed by the expansion tube, and it can be obtained by integrating expansion force along with the displacement of cone piston from 0 to $S_{\text{max}}$. $F(S)$ is the expansion force when the displacement of the cone piston is $S$. 

TABLE 3. Comparison of $F_p$ between experiment and FEM.

| Conditions          | $F_p$ (N) | Diff (%) | Error (%) |
|---------------------|-----------|----------|-----------|
| $\gamma = 5.71^\circ, h = 1.4$ mm | 15.06     | 4.12     | 10.41     |
| Exp                 | 3855      | 3.29     | 8.97      | 3.30  
| FEM                 | 2399      | 4.29     | 9.41      | 3.60  
| Diff (%)            | 10.41     | 8.97     | 3.30      |
| $\gamma = 16.70^\circ, h = 1.4$ mm | 19.34     | 4.14     | 9.41      |
| Exp                 | 11704     | 4.34     | 9.41      |
| FEM                 | 8266      | 4.34     | 9.41      |
| Diff (%)            | 4.34      | 9.41     | 3.78      |
| $\gamma = 26.57^\circ, h = 1.4$ mm | 34.12     | 4.14     | 9.41      |
| Exp                 | 13404     | 4.34     | 9.41      |
| FEM                 | 13652     | 4.34     | 9.41      |
| Diff (%)            | 4.34      | 9.41     | 3.78      |
| $\gamma = 16.70^\circ, h = 0.8$ mm | 22.71     | 1.92     | 3.53      |
| Exp                 | 5818      | 3.36     | 3.83      |
| FEM                 | 4107      | 3.53     | 3.26      |
| Diff (%)            | 3.53      | 3.83     | 2.61      |
| $\gamma = 16.70^\circ, h = 2.0$ mm | 48.84     | 3.53     | 3.83      |
| Exp                 | 19289     | 3.36     | 3.83      |
| FEM                 | 14223     | 3.53     | 3.26      |
| Diff (%)            | 3.53      | 3.83     | 2.61      |
FIGURE 15. Experimentally measured and numerically predicted expansion force-displacement relationships under different loading conditions: (a) quasi-static, (b) $v = 22.86$ m/s, (c) $v = 27.51$ m/s, (d) $v = 34.12$ m/s of expansion tube ($\gamma = 16.70^\circ$, $h = 1.4$ mm).

FIGURE 16. Average crushing force $F_S$ of expansion tubes at different impact velocities.

These experimentally measured average crushing forces $F_S$ of the tubes as a function of impact velocities are shown in Fig. 16. It can be seen that the average crushing forces under dynamic loading condition is lower than the average crushing forces under quasi-static loading condition, approximately 61.8%–79.8%. The average crushing force $F_S$ does not change significantly with the increase of the initial impact velocity of the striker. However, in the research conducted by Choi et al. [18], the expansion force was increased with the increase of impact velocity. The difference can be explained as follow: on the one hand, the parent material of the tube in reference [18] was steel that had significant strain rate effect, whereas that in the present study was aluminum alloy that had negligible strain rate effect; on the other hand, the reference [18] (0-10 m/s) reported that the friction coefficient decreases with the increase of impact velocity, which led to the more significant decrease of expansion force of tubes at higher impact velocity (15-50 m/s). Based on the above two reasons, the dynamic expansion force is lower than the quasi-static expansion force in this study. As shown in Fig. 16, the variation trend of the curve (green triangles) is different from the other four curves. This is because the wall of the expansion tube ($\gamma = 26.57^\circ$, $h = 1.4$ mm) fractured under the quasi-static loading condition, but did not fracture...
under the dynamic loading condition. This leads to the fact that the stable-state expansion force of this expansion tube (γ = 26.57°, h = 1.4 mm) has not been reached when the wall fractured under quasi-static loading condition, as shown in Fig. 7c. Therefore, the quasi-static average crushing force of this type of expansion tube is lower than the dynamic average crushing force. Also, the average crushing force FS increases significantly when the semi angle (ranging from 5.71° to 26.57°) or wall thickness (ranging from 0.8 mm to 2.0 mm) increases, as shown in Fig. 17, which contributes to the increase of energy absorption capacity of the expansion tube.

C. INFLUENCE OF IMPACT VELOCITY ON DEFORMATION OF TUBES

Fig. 18(a) shows the finite element model after the tube is expanded, and Fig. 18(b)–(d) show the outline of the deformed tube after the tube is expanded under different impact velocities. It is can be seen that with the increase of impact velocity, the deformation of tubes in radial direction does not change significantly. Also, the radical displacement of tubes with different semi angles, wall thicknesses and loading conditions at points A and B were calculated. The tensile strain at point A is 8.67%–9.82%, and the tensile strain at point B is 11.08%–12.41%. Hence, the tensile strain at point B is 2.28%–2.94% higher than the tensile strain at point A, which approximately equals to the elastic strain of parent material of tubes, as shown in Fig. 3. It can be concluded that the rebound part of the tube is caused by the elastic deformation of tubes.

D. ENERGY ABSorption CAPACITY OF TUBES

1) PLASTIC DEFORMATION ENERGY OF TUBES

The kinetic energy (i.e. total energy) of striker does not convert into the plastic deformation energy of the tube totally, it is also dissipated by the frictional heat and structural vibration of the expansion tube. The total energy and plastic deformation energy in simulations with different semi angles, wall thicknesses, and initial striker velocities are listed in Table 4 which also includes the percentage of plastic deformation energy to the total energy.

As shown in Table 4, the tube with a lower semi angle of cone piston, i.e. γ = 5.71°, the percentage of plastic deformation energy in total energy is lower, demonstrating that more energy is dissipated by the frictional heat and structural vibration of the expansion tube. In addition, with the increase of striker velocity, the Plastic deformation energy increases, approximately 64.0% to 71.5% of total energy.

2) ENERGY ABSorption PER UNIT MASS Wm OF TUBES

The energy absorption per unit mass Wm of tubes is normally used to evaluate the energy absorption capacity of an absorber, and Wm of the tubes can be defined as:

\[ W_m = \frac{W}{m} = \frac{\int_{0}^{\max} F(S) dS}{\rho S_{max} \pi (D_0 + t)^2} \]
For aluminum alloy, the density $\rho$ is 2750 kg/m$^3$. Fig. 19 shows the energy absorption capacity per unit mass $W_m$ of expansion tubes with different semi angles and wall thicknesses as a function of impact velocity $v$. It can be found that the energy absorption per unit mass $W_m$ increases significantly when the cone piston semi angle or tube wall thickness increases. The energy absorption per unit mass $W_m$ decreases when the load conditions switch from quasi-static to dynamic, then it does not change significantly with the increase of impact velocity. However, for the tube with cone piston semi angle $\gamma = 26.57^\circ$, the energy absorption per unit mass $W_m$ under quasi-static loading condition is less than that under dynamic loading condition. This is due to the tube fractured under the quasi-static test which has been discussed in section II-D. Hence, it can be concluded that the energy absorption per unit mass $W_m$ increases significantly when the cone piston semi angle or tube wall thickness increases, but it does not change significantly when the impact velocity increases.

**E. ENERGY ABSORPTION EFFICIENCY**

Energy absorption efficiency $I$ is used to evaluate the stability of the expansion tube. The higher the energy absorption efficiency $I$ is, the less fluctuation and better energy absorption capacity can be obtained. Energy absorption efficiency $I$ is defined as:

$$I = \frac{\int_0^{S_{max}} F(S) dS}{F_{max} \cdot S_{max}} = \frac{W}{F_{max} \cdot S_{max}} = \frac{F_S}{F_{max}}$$  \hspace{1cm} (3)

Fig. 20 shows the experimentally measured energy absorption efficiency $I$ of tubes as a function of striker impact velocity $v$. For the expansion tube with the same cone piston semi angle and tube wall thickness, energy absorption efficiency is less in dynamic loading conditions than that in quasi-static loading conditions. For the expansion tubes in dynamic loading conditions, the changes in energy absorption efficiency are very small when the cone piston semi angle or tube wall thickness changes. As for the impact velocity, energy absorption efficiency does not change significantly.
FIGURE 19. Energy absorption per unit mass $W_m$ of expansion tubes with different (a) semi angles and (b) wall thicknesses as a function of impact velocity $v$.

FIGURE 20. Experimentally measured energy absorption efficiency $I$ of tubes as a function of striker impact velocity $v$.

when the impact velocity increases (ranging from 15.06 m/s to 48.84 m/s). It should be noted that the expansion tubes with semi angle $\gamma = 5.71^\circ$, the energy absorption efficiency is significantly less than that of expansion tubes with semi angle $\gamma = 16.7^\circ$ or $\gamma = 26.57^\circ$, which means that the anti-impact process is unstable when the semi angle $\gamma = 5.71^\circ$.

V. CONCLUSION

The quasi-static and dynamic mechanical responses of expansion tubes have been reported. A gas gun was employed at the impact velocity up to 50 m/s to obtain the characteristics of expansion tubes with different geometrical parameters. Finite element simulation was conducted to interpret the experimental measurements and the process of tube expansion. It is found that the semi angle of cone piston, the wall thickness of the tube and friction coefficient have a considerable influence on the expansion force as well as the energy absorption capacity of tubes. The experimental results suggest that the tensile fracture strain of the expansion tube is consistent with the fracture strain of parent material in quasi-static tensile tests, which can be concluded that the fracture of tubes in the quasi-static test is governed by the fracture tensile strain of parent material. The average crushing force $F_S$ increases significantly when the semi angle of cone piston or wall thickness of the expansion tube increases. The average crushing force decreases when the loading condition switch from quasi-static to dynamic, whereas the average crushing force does not change significantly when the impact velocities changes. Finite element simulations were seen to be in good agreement with experimental measurements. The numerical results demonstrate that the friction coefficient is the critical factor to the mechanical characteristics of expansion tubes and the values are different in quasi-static and dynamic loading conditions. The rebound tensile strain at the leading edge of the expansion tube is approximately equal to the elastic strain of the parent material of the tube, which can be concluded that the rebound part of the tube is caused by elastic deformation of tubes. The plastic deformation energy of the expanded tube is about 64%–71.5% of the total kinetic energy of strikers. The energy absorption capacity per unit mass and energy absorption efficiency decrease when the loading condition switches from quasi-static to dynamic, and it does not change significantly with the increase of the velocity of the striker.

The strain rate effect has an important influence on the mechanical response of expansion tube. Different materials have different strain rate effects. In future works, the influence of materials on the mechanical response of the expansion tube will be studied.

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