Abstract
The dynamic behavior of a thin-walled circular tube under high-velocity impacts was investigated by performing a finite element analysis (FEA) and an experiment. The FEA for a thin-walled circular tube subjected to an axial impact revealed the effect of the impact velocity. The results suggested that an increase in the initial impact velocity enhances the absorbed energy through compressive deformation. High-velocity impact tests on aluminum alloy circular tubes confirmed the energy absorption characteristics indicated by the FEA.

Key words
Dynamic Behavior, Impact Load, Absorbed Energy, Circular Tube, Axial Impact

1. Introduction
Higher safety requirements for vehicles are being demanded as measures to protect passengers involved in traffic accidents. The structural components of the vehicles must absorb sufficient energy and reduce the impact load even when vehicles collide with each other or other object at high velocities.

Thin-walled tubes have been used as impact energy absorbers because they progressively buckle like bending plastic hinges to absorb energy under axial low-velocity impacts [1-3]. The dynamic behavior of thin-walled tubes under high-velocity impacts has been experimentally [4] and numerically [5-11] investigated in terms of the buckling modes and energy absorption.

It has been suggested by several numerical studies [6-8,10] that a higher impact velocity enhances the energy absorbing capability of a thin-walled tube because a large portion of the initial impact energy can be absorbed by compressive deformation before progressive buckling begins. One should thus clarify the effect of impact velocity on the impact load history of a thin-walled tube as well as the characteristics of energy absorption associated with the impact load in order to use thin-walled tubes as vehicle structures.

In this study, the dynamic behavior of a thin-walled circular tube subjected to high axially impact loading was studied in finite element analyses (FEA) and experiments. The FEA, for a circular tube made of an elastic-plastic material subjected to high and low velocity impact, detailed the effect of the impact velocity on the peak and average load as well as the energy absorption characteristics during compressive deformation. Absorbed energy-displacement curves obtained by the FEA indicated that a higher initial impact velocity could enhance the absorbed energy due to the compressive deformation just after impact.

The high-velocity impact tests were performed to verify the absorbed energy of aluminum alloy circular tubes through the compressive deformation based on the quantitative result in the FEA.

2. Finite Element Analysis
2.1 Analytical conditions
The FEA was conducted by using the explicit finite element code, Hyperworks Altair Radioss, ver. 10, to investigate the dynamic behavior of a circular tube under high velocity impact.

A circular tube was made of aluminum alloy which is insensitive to the strain rate. The aluminum alloy tube used in the analysis was assumed to be an elastic-plastic material with linear strain hardening and obeying the von Mises yield criterion. The material properties, i.e., Young’s modulus, $E$, hardening modulus, $E_h$, yield stress, $\sigma_y$, Poisson’s ratio, $\nu$, and density, $\rho$, of the circular tube are listed in Table 1.

The circular tube was modeled with four-node thin-shell elements (QEPH) allowing the thickness to be changed by large in-plane deformations. Figure 1 shows the geometry of the circular tube and loading condition. The radius, $R$, thickness, $h$, and length, $L$, of the circular tube were 15, 0.6 and 100 [mm], respectively. The circular tube was fixed to a rigid wall, and was struck axially by a rigid body with mass, $m$, and an initial impact velocity, $V_0$, ranging from 5 to 140 [m/s]. The mass of the rigid body was selected depending on $V_0$, so as to keep the rigid body’s kinetic energy constant, 300 [J]. The rigid body moved along the central axis of the circular tube to simulate the impact load as a virtual load.

Table 1 Material properties of Al alloy circular tube used in FEA

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E$ [GPa]</td>
<td>68.3</td>
</tr>
<tr>
<td>$E_h$ [MPa]</td>
<td>683</td>
</tr>
<tr>
<td>$\sigma_y$ [MPa]</td>
<td>195</td>
</tr>
<tr>
<td>$\nu$</td>
<td>0.3</td>
</tr>
<tr>
<td>$\rho$ [kg/m$^3$]</td>
<td>2690</td>
</tr>
</tbody>
</table>

![Fig. 1 Geometry of a circular tube and loading condition](image-url)
tube. A friction coefficient of 0.2 was assumed in the friction contact conditions between the circular tube and rigid walls and the self-contact on the inner and outer surfaces of the circular tube.

The finite element analysis computed not only the impact load on the impacted end, $P_I$, of the circular tube but also the load at the opposite fixed end, $P_F$. The energy absorbed by the circular tube, $K$, was defined as the external work imparted from the rigid body to the circular tube, i.e.,

$$K(\delta(t)) = \int_0^{\delta(t)} P_I(\delta(t)) d\delta(t)$$

where $\delta(t)$ is the displacement history at the impacted end.

### 2.2 Results of finite element analysis

Figures 2 and 3 show the results of the finite element analysis with initial impact velocities of $V_0 = 100$ and 5 [m/s] as typical cases for high- and low-velocity impacts: (a) histories of the axial loads and the displacement and (b) the axial load- and absorbed energy-displacement curves obtained from the data in each Fig. (a). In the figures, $t_\delta$ is the time when the axial displacement reaches the maximum value, $\delta_{\text{max}}$. For all initial impact velocities, the absorbed energies at the maximum displacement, $K(\delta_{\text{max}})$, agreed with the initial kinetic energy of the rigid body (300 [J]).

The maximum displacement, $\delta_{\text{max}}$, at $V_0 = 100$ [m/s] was smaller than that at $V_0 = 5$ [m/s] as shown in Figs 2 and 3.

Figure 4 shows the crushed shapes of the circular tubes at a maximum displacement, $\delta_{\text{max}}$. The deformation mode of the tubes for impacts with $V_0 = 100$ and 140 [m/s] varied from axisymmetric to non-axisymmetric, while the modes for impacts with the other initial velocities were axisymmetric throughout the whole deformation.

Figure 5 shows the relation between the maximum displacement, $\delta_{\text{max}}$, and the initial impact velocity, $V_0$. The maximum displacement, $\delta_{\text{max}}$, decreased as the initial impact velocity, $V_0$, increased independently of the buckling mode, which indicates that the average load, $P_{\text{ave}} = K(\delta_{\text{max}}) / \delta_{\text{max}}$, increases with $V_0$. Consequently, the increase in the initial impact velocity enhances energy absorbing capability of the circular tube. As reported in Ref. [6-8,10], a large portion of the initial impact energy can be absorbed by compressive deformation before the progressive buckling wrinkles the tube in the higher velocity impact.

The axial loads at the impacted and fixed ends did not coincide with each other under the high-velocity impact (Fig. 2(a)), in contrast to the low-velocity case (Fig. 3(a)), suggesting that the inertia effect of the circular tube becomes significant in the high-velocity impact, which causes a difference in the peak loads at the impacted and fixed ends.

In the high-velocity impact, the axial load at the impacted end reached the first peak just after the impact, as shown in Fig. 6(a), which is redrawn from Fig. 2(a) on a time scale. The first peak load at the impacted end can be theoretically predicted as follows [6,7]:

![Figure 2 Results of finite element analysis with initial impact velocity, $V_0 = 100$ [m/s]](image)

![Figure 3 Results of finite element analysis with initial impact velocity, $V_0 = 5$ [m/s]](image)
Equation 2 is for an elastic-plastic circular tube, in the plane stress state, made of a strain-rate-insensitive material with linear strain hardening and obeying the von Mises yield criterion.

\[ P_{r,\text{max}} = 2\pi Rh\left(\sigma_y + V_0 \sqrt{\rho E_n}\right) \frac{2}{\sqrt{3}} \]  

(2)

In the case that the yield stress is much smaller than an elastic buckling stress as calculated by Timoshenko’s formula: \( \sigma_{\text{cr}} = E/(3(1-\nu^2))(t/R) \) [12], Note that the axial load at the fixed end reached a value of 12.73 [kN] (Eq. (3)), and then increased to the peak, after which it decreased with oscillations, even under the high-velocity impact of \( V_0 = 100 \) [m/s] (Fig. 6(a)).

For the low-velocity impact (Fig. 6(b)), after multiple reflections of elastic stress waves, the axial loads at both ends (the static yield load) are given by

\[ P_{s,yield} = 2\pi Rh\sigma_y \]  

(4)
A time-average load was calculated because of the difference between the axial loads at the impacted and fixed ends. To estimate the time-average load, the impulses at the impacted and fixed ends of a circular tube are calculated from

\[
 I_I(t) = \int_0^{t_d} P_I(t')dt', \quad I_F(t) = \int_0^{t_d} P_F(t')dt' \tag{5}
\]

Figure 7 shows the histories of the impulses at the impacted and fixed ends for \( V_0 = 100 \text{ [m/s]} \). Both impulses agreed with the initial kinetic momentum of the rigid body \((mV_0 = 6 \text{ [N·s]})\) at \( t_d \) when the rigid wall’s velocity becomes 0. Therefore, the time average load can be defined by

\[
 \bar{P}_{ave} = \frac{mV_0}{t_d} \tag{6}
\]

Figure 8 summarizes the first peak loads, the time-average load (Eq. (6)) against the initial impact velocity, \( V_0 \). The first peak load at the impacted end increases with \( V_0 \) as indicated by Eq. (2). On the other hand, the first peak load at the fixed end above \( V_0 = 20 \text{ [m/s]} \) remains constant and is larger than in the low velocity case \((V_0 = 5 \text{ [m/s]})\) or the estimation by Eq. (3). The time-average load is also insensitive to the initial impact velocity up to higher velocities.

Figure 9 plots the absorbed energy-displacement curves for \( V_0 = 5, 60, 100 \) and \( 120 \text{ [m/s]} \). The curves were approximated by bilinear forms, as shown by the dashed lines in the figure. The first slope means the energy absorption per unit displacement due to the compressive deformation before the progressive buckling of the plastic bending hinges. The absorbed energy due to compressive deformation increases as the initial impact velocity increases. The second slopes, which mean the energy absorption per unit displacement due to progressive buckling, of the high-velocity impact cases \((V_0 = 60, 100 \) and \( 120 \text{ [m/s]})\) were approximately parallel to the one of the low-velocity impact case \((V_0 = 5 \text{ [m/s]})\). This means that a considerable portion of the initial impact energy in a high-velocity impact can be absorbed by compressive deformation of a circular tube.

### 3. Experiment

#### 3.1 Experimental procedure

High-velocity impact tests were performed to qualitatively confirm the simulated energy absorption characteristics of a circular tube compressed by a high-velocity impact.

The specimens were aluminum alloy (JIS A6063-T6) tubes of 50 [mm] in length with outer diameters of 18 [mm] and thicknesses of 0.5 [mm].

A special experimental apparatus was built for the impact test, the schematic diagram of which is shown in Fig. 10. The apparatus mainly consisted of an air gun for launching an impactor and a load cell. Figure 11 shows the details of the impactor. The impactor of 52 [g] was composed of steel contacting the specimen and ultra-high molecular weight polyethylene (UHMWPE) for low friction. The impactor was launched by releasing compressed air through a gun barrel having bore of 20 [mm] and length of 1.5 [m]. The impact velocities were measured by a velocity pickup. The axial load, which corresponds to the load at the fixed end in the FEA, was measured using a load cell made of steel. The load cell had a detection part (20 [mm] in diameter and 20 [mm] long) mounted with semiconductor strain gages, at one end of the stress-transmitted sector (60 [mm] in diameter and 600 [mm] long). The load cell was designed to accurately measure the impact load [13]. In order to ensure a stable collapse of the specimen, steel plates (30 [mm] in length and width and 1.6 [mm] in thick) were attached to both ends of the specimen by a cyanoacrylate adhesive (Loctite LKK-020, Henkel). The specimen with plates was attached to the detection part of the load cell by the adhesive.
The displacement history at the impacted end of the specimen was calculated by using the following equation [13]:

\[
\delta(t) = V_0d - \frac{1}{m} \int_0^t \int P(t') dt'' dt'
\]

where \( P(t) \) is the axial load history measured by the load cell and \( m \) is the mass of the impactor.

### 3.2 Results and discussion

Figure 12 shows the histories of the axial load at the fixed end measured by the load cell, the impulse defined by Eq. (5) as well as the time average load calculated by Eq. (6) for \( V_0 = 37 \) and 55 [m/s]. Figure 13 shows specimen shapes after the impact tests with \( V_0 = 37 \) and 55 [m/s].

As shown in Fig. 12, the natural vibrations of the load cell were superimposed on the load histories. Although the peak value of the axial loads could not be detected by the load cell, the axial loads had high values just after the impact, then decreased when the plastic hinges formed, as shown in Fig. 13. Because the mass of the impactor, \( m \), was constant, i.e., the initial impact energy changed with the initial velocity, \( t_d \) and the numbers of wrinkles in the specimens increased as the initial impact velocity increased as shown in Figs. 12 and 13. The time-average loads of the impact tests whose impact velocities ranged from 37 to 55 [m/s] were not affected by changing the impact velocity.

Figure 14 shows the load-displacement curves obtained in the impact test with \( V_0 = 55 \) [m/s] and the static test that was carried out under a constant displacement rate of \( 8.3 \times 10^{-5} \) [m/s] on a universal testing machine (Autograph AGS-10kJ, Shimadzu). The static test curve (\( V_0 = 8.3 \times 10^{-5} \) [m/s]) showed the typical behavior of a circular tube under progressive buckling with axisymmetric mode. The impact test curve was above the static one until the displacement reached approximately a value of 4 [mm]. This is because a
impact tests on aluminum alloy circular tubes verified compressive deformation. Finally, static- and high-velocity deformation of the circular tube, so that a considerable velocity enhances the absorbed energy due to compressive deformation. Additionally, the FEA indicated that a higher initial impact velocity affected by changing the initial velocity above 20 \([m/s]\). The average loads measured at the opposite fixed end was not affected by the FEA results in Fig. 9.

The experimental results indicate that a higher initial impact velocity enhanced the absorbed energy due to compressive deformation just after the impact.

4. Conclusion
This paper described the dynamic behavior of thin-walled circular tubes subjected to high velocity impacts on the basis of a FEA and experiment.

The FEA of an aluminum alloy circular tube subjected to high- and low-velocity impacts, confirmed that while the peak load at the impacted end of a circular tube increased as the initial impact velocity increased, the peak and time average loads measured at the opposite fixed end was not affected by changing the initial velocity above 20 \([m/s]\). Additionally, the FEA indicated that a higher initial impact velocity enhances the absorbed energy due to compressive deformation of the circular tube, so that a considerable portion of the initial impact energy can be absorbed by compressive deformation. Finally, static- and high-velocity impact tests on aluminum alloy circular tubes verified qualitatively the effect of energy absorption by compressive deformation in the experiment. The absorbed energy-displacement curves obtained by the tests confirmed that a higher initial impact velocity could enhance the absorbed energy because of compressive deformation just after impact, as indicated by the FEA.

References