Reduction of Impulsive Noise Caused by Unsteady Compression Wave*

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When an unsteady compression wave propagating in a tube arrives at its exit, an impulsive wave is emitted toward the surrounding, which causes an impulsive noise. The objective of this work is to investigate the appropriate silencer configuration for reducing this impulsive noise. Five kinds of silencers were installed at a shock tube exit, and their effects on the impulsive noise were investigated numerically and experimentally. Two-dimensional unsteady conservation equations were solved by means of a PLM. Experiments were carried out using a shock tube with an open end. As a result, the effect of silencer configuration on noise reduction was clarified, and it was found that a suitable choice of the silencer could reduce the strength of the impulsive wave by about 25%.

Key Words: Compressible Flow, Compression Wave, Impulsive Noise, Shock Tube, High-Speed Train, Passive Control, Silencer

1. Introduction

As a high-speed train enters a tunnel, a compression wave is generated ahead of the train and propagates along the tunnel. At the exit of the tunnel, part of the compression wave is reflected backward as an expansion wave and propagates toward the tunnel entrance. The expansion wave interferes with the moving train and consequently causes pressure transients in the tunnel and ear discomfort to passengers riding the train. On the other hand, part of the compression wave emerges from the tunnel exit and becomes an impulsive wave, which radiates toward the surroundings, creating an impulsive noise resembling the sonic boom caused by a supersonic aircraft. The impulsive noise has been known to be of low frequency over the range of several hertz to several

ten hertz and to have high sound pressure levels. In fact, the impulsive noise was not an issue before the 1980's when train speed was not very high. Recently much effort has been expended to speed up trains in many countries. This has caused impulsive noise to become a new type of environmental problem, because train speed is already exceeding the speed of some transport planes of a few years ago. Thus impulsive noise is one of the engineering problems that must be resolved before attempting to further increase the speed of trains.

According to the aeroacoustic theory⁽¹⁾, impulsive noise is closely related to the compression wave propagating in a tunnel. Therefore, it is important to investigate the characteristics of the compression wave to resolve the impulsive noise problem. Many investigations of a compression wave propagating in a tunnel have been performed so far, and consequently, it was found that the compression wave is related to factors such as the train speed, the ratio of the cross-sectional areas of the train and the tunnel, and the characteristics of the train track^{(2),(3)}. Furthermore it was found that the magnitude of the impulsive wave is proportional to the pressure gradient of the

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compression wavefront at the tunnel exit⁽⁴⁾.

In order to reduce impulsive noise, many attempts have been concentrated on alleviating the pressure gradient of the compression wave arriving at the tunnel exit. Some examples of noise control methodologies are the use of tunnel entrance hoods and side branches or bleed holes in the tunnel^{(5),(6)}, and some effects were estimated experimentally or numerically. As a result, such passive structures have been adopted at the entrance or inside actual tunnels. However, no significant reduction of impulsive noise has been achieved so far, and it is thus necessary to investigate other possible methods.

For the reduction of impulsive noise, we tested new passive control devices which are applied at the exit portal of the tunnel. Such devices have not previously been examined because of the large wavelength of a compression wave propagating in a real tunnel or the long time scale of impulsive noise. However, with the advent of the recent high-speed trains, the frequency of the impulsive noise has increased and passive controls at the exit portal of the tunnel may be effective for impulsive noise reduction.

In the present study, a simple open-ended shock tube was employed as a model tunnel. We installed five kinds of silencers with different configurations at the exit of the model tunnel and tested the effects of the profile of the incident compression wave on impulsive noise reduction. The effect of the present passive control on impulsive noise was also evaluated using a numerical simulation. Two-dimensional unsteady conservation equations were differenced and solved by the piecewise linear method.

2. Numerical Method

A numerical simulation on the discharge of a compression wave from a shock tube which has a

Fig. 1 Silencer and calculation area

rectangular or triangular box at its exit was carried out. The tube configuration and computational domain are shown in Fig. 1. Two-dimensional inviscid unsteady compressible equations were converted to dimensionless forms by using the pressure, density, sound speed and under atmospheric conditions and the size of the silencer. The governing equations are written in terms of the following conservative equation:

$$\frac{\partial U}{\partial \tau} + \frac{\partial F}{\partial x} + \frac{\partial G}{\partial y} = 0$$

$$U = \begin{bmatrix} \rho \\ \rho u \\ \rho v \\ e \end{bmatrix}, \quad F = \begin{bmatrix} \rho u \\ \rho u^2 + p \\ \rho uv \\ (e+p)u \end{bmatrix}, \quad G = \begin{bmatrix} \rho v \\ \rho uv \\ \rho v^2 + p \\ (e+p)v \end{bmatrix} \quad (1)$$

$$e = \frac{p}{\gamma - 1} + \frac{1}{2} \rho (u^2 + v^2), \quad (2)$$

where ρ is the density of gas, u and v the velocities, τ the time, x and y the spatial coordinates, p the pressure, γ the specific heat ratio of gas, and e the total energy per unit volume.

The above Eq. (1) was solved by the piecewise linear method, which is known to be effective in flows with a pressure discontinuity, such as a shock wave⁽⁷⁾. The boundary conditions employed in the calculation are the outflow condition at the downstream flow field and the slip wall conditions at the solid boundaries. The incident compression waves in the tube were given as the initial condition in the range of $x/D \leq -2.0$ for the comparison with the results of shock tube experiments. The present calculation uses three model profiles expressed by Eqs. (3) to (5).

$$\frac{\Delta p_i}{p_1} = -\frac{\Delta p^*}{p_1 W} x \quad (3)$$

$$\frac{\Delta p_i}{p_1} = \frac{\Delta p^*}{p_1} \left\{ -\frac{2}{\pi} \tan^{-1} \left(\frac{\pi}{2.5 W} x \right) \right\} \quad (4)$$

$$\frac{\Delta p_i}{p_1} = \frac{\Delta p^*}{p_1} \left[0.5 - \frac{1}{\pi} \tan^{-1} \left\{ \frac{\pi D}{W} \left(\frac{x}{D} + \frac{16}{3} \right) \right\} \right] \quad (5)$$

Here p_1 is the atmospheric pressure, x is the distance from the tube exit, and D and W are the height of the tube and length of the compression wave, respectively. Δp_i and Δp^* indicate the gauge pressure, as shown in Fig. 1. In the present calculation, Δp^* is fixed at 2 kPa, except in some cases, and the value of W/D is varied in the range of 0.1–2.0. The compression waveforms given by the above Eqs. (3) to (5) are illustrated in Fig. 2. The waveform of Eq. (5) is especially interesting because the real waveforms measured in a high-speed railway tunnel are similar to it. The maximum rate of pressure change of the compression wavefront, $(\partial \Delta p_i / \partial t)_{sm} = a_1 (\partial \Delta p_i / \partial x)_{sm}$ (a_1 is the speed of sound under atmospheric conditions) is the same for Eqs. (3)–(5), but the position at which

the maximum value occurs is different, i.e., it is located at the head and near the midpoint of the compression wave for Eqs.(3) and (4) and for Eq.(5), respectively.

Figure 3 shows the schematics of the passive control structures installed at the shock tube exit. Five types of silencers are adopted here as impulsive noise control devices. They are defined as the models (a) to (e).

Model (a) : reference geometry

Model (b) : $L=H=D$, and $L=2D$, $H=D/2$ (with or without a baffle plate of length l)

Model (c) : $L=H=D$, and $L=2D$, $H=D/2$ (with a porous plate of porosity P)

Model (d) : $L=2D$, $H=D$ (with or without a baffle plate of length l)

Fig. 2 Initial compression waveforms ($W=1.5D$)

Fig. 3 Silencer configurations

plate of length l)

Model (e) : $L=2D$, $H=D$ (with or without a baffle plate of length l)

For the models (b) to (e), the cross-sectional areas of abrupt enlargement at the tube exit are the same as D^2 . Some models have a baffle plate with its length of l in order to alleviate the pressure gradient of the compression wave. In particular, the model (c) has a porous plate with a porosity of P in order to evaluate the effect of configuration of the baffle plate on the pressure gradient of compression wave.

3. Experimental Apparatus and Method

The experimental apparatus used in the present study is a simple open-ended shock tube. It has a cross-sectional area of $D=60 \times 60 \text{ mm}^2$ with a total length of 4 635 mm. The high-pressure tube, which has a length of 1 500 mm, is filled with compressed dry air, and the abrupt rupture of a cellophane diaphragm 0.03 mm thick induces a high-pressure airflow into the test tube. A compression wave is generated by the abrupt inflow of the high-pressure air and propagates toward the tube exit. Not all the silencers shown in Fig. 3 are employed in the present experiments; only models (b) and (c) are selected for comparisons with the numerical calculations. Two pressure transducers (PCB 112 A 21) were mounted flush on the bottom wall of the tube at $x/D=-4.0$, $y/D=0.0$ and $x/D=2.0$, $y/D=0.0$ to measure the unsteady pressure waves. The output of the pressure transducers is recorded by an X-Y recorder via a wave memory,

Fig. 4 Comparison between calculated and measured pressure variations

after they are amplified by D.C. amplifiers. The uncertainty in the pressure measurements is estimated to be $\pm 3\%$.

4. Results and Discussions

4.1 Effect of silencer configuration

Figure 4 shows the comparisons between calculated and measured pressure variations. Figure 4(a) shows the compression waveform measured at $x/D = -4.0$, $y/D = 0.0$, and Figs. 4(b), (c), and (d) the impulsive waveforms measured and calculated at $x/D = 2.0$, $y/D = 0.0$ for models (a), (b), and (c), respectively. In the present calculations, the model waveform expressed by Eq. (5), which is the same as the waveform measured at $x/D = -4.0$, was employed as the initial compression waveform. It is evident from Figs. 4(b) - (d) that the numerical waveforms agree well with the experimental ones, and moreover, the maximum overpressures of the impulsive wave are nearly the same for both the numerical calculations and the experiments. For model (b), Fig. 5 shows the effects of the present passive control on the impulsive wave, where Eq. (5) was employed as the initial compression wave. The calculation parameter is the length of baffle plate l , and the length of the compression wave W is the same as the equivalent diameter of the tube (D), and is fixed at 9 m. The maximum overpressure of the impulsive waves in model (b) is lower than in model (a). This is obvious due to the reduction in the pressure gradient of the compression

Fig. 5 Effect of passive control on pressure variation in model (b) (Initial compression wave: Eq. (5) and $W = 1.5D$)

wave passing through the region of abrupt change in the cross-sectional area at the tube exit. It is particularly interesting to note that only one peak for model (a) appears in the impulsive waveform, but two peaks appear for model (b). This is caused by diffraction of the compression wave at the region of abrupt change in the cross-sectional area. For impulsive waves with such two peaks, the optimum geometry for impulsive noise reduction is likely to be $l/D = 1/2$. It is reasonable to consider that the baffle plate delays the time until the diffraction wave in the region of abrupt change in the cross-sectional area affects the main flow. Here it should be noted that for a large l , the first peak becomes large due to the limitation of the diffraction wave. On the other hand, the same results were obtained for model (b) with $L = 2D$ and $H = D/2$, but the impulsive waveforms were more complicated. The optimum length of the baffle plate

Fig. 6 Effect of passive control on pressure variation in model (d) (Initial compression wave: Eq.(5) and $W=1.5D$)

Fig. 7 Effect of baffle plate porosity on pressure variation in model (c) (Initial compression wave: Eq.(5) and $L=H=D$, $W=1.5D$)

was $l/D=1$ and $l/L=1/2$. Similar effects were also obtained for models (c) to (e). For example, Fig. 6 shows the impulsive waveforms in model (d). The optimum length of the baffle plate is $l=D/2$ and the reduction of maximum overpressure in this case appears to be about 20%. The effect of the use of the porous baffle plate is shown in Fig. 7, where model (c) with $L=H=D$ was tested. For a given silencer, it is seen that there is an optimum porosity of the baffle plate for reducing the maximum overpressure of the impulsive wave. It can be observed from Fig. 7 that the porosity of 30% is the most effective for reducing impulsive noise. From the viewpoint of impulsive noise reduction, the optimum geometries of silencers are summarized in Table 1.

Figure 8 shows the calculated impulsive waveforms for the optimum geometries at the exit of the tube. The optimum geometries can be seen in model (b) which has $L=2D$, $H=D/2$, $l=D$, and $W=1.5D$ and model (e) having $l=D/2$, and the maximum overpressure of the impulsive wave is reduced by about 25% in comparison with that of model (a).

4.2 Effect of compression waveform

Figure 9 shows the pressure variations $\Delta p/p_1$ calculated at $x/D=2.0$ for the three kinds of compression waves expressed by Eqs.(3) to (5), in which the nondimensional time τ is defined by $(ta_1\sqrt{\gamma})/D$ (where γ is the ratio of specific heat). It is observed that the

Table 1 Optimum silencer parameters

Model (c), $L=H=D$	Porosity $P=30\%$
Model (c), $L=2D$, $H=D/2$	Porosity $P=30\%$
Model (d)	$l=D/2$ ($l/L=1/4$)
Model (e)	$l=D/2$ ($l/L=1/4$)

Fig. 8 Comparison of pressure variations for effective silencers (Initial compression wave: Eq.(5) and $W=1.5D$)

Fig. 9 Impulsive waveforms for model (a) at $x/D=2.0$

maximum overpressure Δp_m of the impulsive waves is obtained for the compression wave of Eq.(3), for which the pressure gradient is largest among the three types of the initial compression waves.

Figure 10 shows the relationship between the maximum overpressure of the impulsive waves $\Delta p_m/p_1$ and the excess pressure of the initial compression waves with various pressure gradients. The maximum pressure gradient of the initial compression waves is represented by $a_1(\partial \Delta p_i / \partial x)_{sm}$, where Eq.(3) was employed as the initial compression waveform because it is relatively straightforward. For the compression waves with pressure gradients that are very large, like a shock wave, the maximum overpressure of the impulsive wave $\Delta p_m/p_1$ increases with

Fig. 10 Relation between impulsive wave strength and maximum overpressure of initial compression wave at $x/D=2.0$ (model (a)), Initial compression waveform of Eq.(3)

Fig. 11 Effect of initial compression wave on impulsive wave of model (b) ($L=2D$, $H=0.5D$, $l=D$, and $W=1.5D$)

increasing excess pressure of the initial compression waves. However for a smaller value of $a_1(\partial \Delta p_i / \partial x)_{sm}$, $\Delta p_m/p_1$ approaches a constant value when $\Delta p^*/p_1$ becomes larger. The critical values of $\Delta p^*/p_1$ indicating the asymptotic values in $\Delta p_m/p_1$, which are marked by the arrows, decrease with decreasing pressure gradients of the compression wave $a_1(\partial \Delta p_i / \partial x)_{sm}$.

In the case of model (b) with $L=2D$, $H=D/2$, $l=D$, and $W=1.5D$, the three initial compression waves expressed by Eqs.(3) to (5) result in the three impulsive waveforms at $x/D=2.0$, as shown in Fig.11. The maximum overpressure value of the impulsive wave decreases in comparison with that for model (a), while it increases for the initial waveform of Eq.(4) (see Fig.9). It is thought that the diffracted wave at the part of the abrupt area change strongly affects the main flow.

(a) $W/D=0.1$

(b) $W/D=1.0$

(c) $W/D=2.0$

Fig. 12 Effect of initial compression wavelength on impulsive wave of model (b) ($L=2D$, $H=0.5D$, and $l=D$)

Fig. 13 Relation between compression wave length and maximum overpressure of impulsive wave of model (b) ($L=2D$, $H=0.5D$)

4.3 Effect of compression wavelength

Figure 12 shows the effect of the wavelength of the initial compression wave on impulsive noise reduction. For a given silencer, there seems to be an optimum wavelength for noise reduction. In the case of model (b), the variation of the maximum over-

Fig. 14 Relation between compression wave length and maximum overpressure of impulsive noise reduction of model (b) ($L=2D$, $H=0.5D$)

pressure of the impulsive wave with the initial compression wavelength is shown in Fig. 13, in which the maximum pressure gradient of the initial compression waves $a_1(\partial \Delta p_i / \partial x)_{sm}$ is the same for all wave lengths. As W/D is increased, the maximum overpressure of the impulsive wave is decreased. For the baffle plate of $l=1.5D$, the rate of decrease of the maximum overpressure against W/D is seen to be the same as that for model (a). However, for the baffle plate length with l less than $l=1.5D$, the reduction of the maximum overpressure is not great as W/D becomes larger than 1.0. Therefore, the present passive control is not effective for $W/D > 1.0$.

Impulsive noise reduction for model (a) is shown in Fig. 14. The benefit parameter ($B\%$) is plotted against the initial compression wavelength. In numerical calculations, Eq. (3) was employed as the compression waveform for model (b). Here the benefit parameter is defined as the maximum overpressure reduction ratio, i.e.

$$B(\%) = \frac{\Delta p_m - (\Delta p_m)_{\text{model(a)}}}{\Delta p_m} \times 100(\%) \quad (6)$$

where Δp_m and $(\Delta p_m)_{\text{model(a)}}$ denote the maximum overpressures of the impulsive waves for the testing model and model (a), respectively. The strength of the impulsive wave is reduced by about 15% for $W/D = 2.0$ and about 25% for $W/D = 1.0$. This means that the passive device is the most effective when its physical size is of the same order as the initial compression wavelength. This result is somewhat surprising in that the present calculations do not include the viscous dissipation mechanism in the passive devices. It can also be seen that there exists an optimal W/D

for each passive device. This is inferred from the finding that the frequency of the impulsive wave becomes lower as W/D increases. In general, W/D decreases with increasing train speed, and the frequency increases with decreasing W/D . This means that a large box is necessary for effective control of impulsive noise. On the other hand, for model (b) with $L=2D$, $H=D/2$, the maximum reduction of impulsive noise is nearly the same for both cases of $l=0$ and $D/2$. For smaller W/D , the noise reduction is negligibly influenced by the baffle plate, but as $W/D > 1.0$, the noise reduction of the baffle plate of $l=D/2$ is estimated to be about 6% in comparison with that in the case of $l=0$.

5. Conclusions

The effect of various types of silencers on impulsive noise emitted from an open end of a shock tube was investigated numerically, together with shock tube experiments. The effect of the silencer geometries on impulsive noise reduction was discussed and the design factors of the silencer were suggested. The interesting result is that the strength of the impulsive wave is reduced by about 25% using an optimum silencer. The results obtained from the present study are promising in designing a high-speed railway tunnel.

References

- (1) Blake, W.K., *Mechanics of Flow-Induced Sound and Vibration*, Vol. 1, (1979), Academic Press.
- (2) Ozawa, S., *Study of Micro-Pressure Wave Radiated from a Tunnel Exit*, Railway Technical Research Report of Japanese National Railways, (in Japanese), No. 1121, (1979).
- (3) Matsuo, K. and Aoki, T., *Wave Problems in High-Speed Railway Tunnels*, Proc. 18th Intl. Symp. on Shock Tube and Waves, K. Takayama, ed., (1991), p. 95, Springer-Verlag, Sendai, Japan.
- (4) Matsuo, K., Aoki, T., Kashimura, H. and Nonaka, Y., *Emission of a Propagating Compression Wave from an Open End of a Tube (1st Report, Analytical Study)*, Trans. Jpn. Soc. Mech. Eng. (in Japanese), Vol. 58, No. 546, (1992), p. 331.
- (5) Vardy, A.E., *Reflection of Step-Wavefronts from Perforated and Flared Tube Extensions*, Jour. Sound and Vibration, Vol. 59 (1978), p. 785.
- (6) Dayman, B. and Vardy, A.E., *Alleviation of Tunnel Entry Pressure Transients: 1. Experimental Program*, Proc. 3rd Intl. Symp. on Aero. and Ventilation of Vehicle Tunnels, (1979), p. 343.
- (7) Collela, P. and Galz, H.M., *Efficient Solution Algorithms for the Riemann Problem for Real Gases*, LBL-15776, Lawrence Berkeley Laboratory, Univ. of California, (1983).