Methods of Loss Estimation in Compressible Flow through Pipe Orifices and Nozzles*

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Compressibility effects on the characteristics of the flow through orifices and nozzles have been investigated, theoretically and experimentally, under the conditions of subsonic and supercritical flow regimes. A practical experimental formula is proposed by expressing the amount of loss as a ratio of loss coefficient of compressible flow case to that of incompressible flow case.

The results of energy flow measurement indicate that the major part (60-70%) of the total dissipation energy arises within one-diameter downstream of orifice. In addition, by measurements of the velocity and wall pressure, it is confirmed that the maximum pressure recovery occurs at 2 to 3 times the pipe diameter downstream of the reattachment point, and this location corresponds to the point where the irregularity of the velocity distribution due to the existence of orifice almost disappears.

Key Words: Compressible Flow, Orifice Meter, Flow Nozzle, Loss Estimation, Loss Characteristics

1. Introduction

It has been said that the amount of pressure loss of comparatively low-viscous liquid and gas flows through orifices and nozzles can be precisely estimated by the well known ASME experimental data† and/or conventional experimental formulae. In gas flows, however, even when the pressure ratio across orifices or nozzles decreases to 0.9, the throat Mach number increases to about 0.4. And compressibility effects on the pressure loss become no longer negligible. Therefore, for the loss estimation in compressible flows, corrections taking account of compressibility effects have to be made in the conventional incompressible loss data.

In various plants in which numerous orifice meters are installed, it is important for the optimum designs and operations to estimate the flow losses and mass flow rates precisely. However, at present, no investigation can be found relating to the prediction of the compressible flow losses through orifices and nozzles, except the authors' paper§ based on a shock tube technique. And, further development of the investigation is considered necessary.

An analytical method is proposed in this paper which enables easy and precise estimation of the flow losses with orifices and nozzles. In addition, a practically convenient experimental formula is proposed for the estimation of the compressibility effect by expressing the amount of loss as a ratio of loss coefficient of compressible flow case to that of incompressible flow case.

For the purpose of elucidating the flow loss mechanism, the characteristics of the flow through the orifice are observed by means of pitot tube traverse measurements. As a loss estimation method of compressible flows, a shock tube technique is demonstrated in Appendix.

2. Nomenclature

\[ A = \text{Cross sectional area} \]
\[ A_o = \text{Orifice opening area} \]
\[ a = \text{Velocity of sound} \]
\[ C_o = \text{Contraction coefficient} \]
\[ C_v = \text{Velocity coefficient} \]
\[ C_f = \text{Coefficient of skin friction} = \frac{1}{2} \rho \frac{a^2}{p} \]
\[ C_p = \text{Specific heat at constant pressure} \]
\[ D = \text{Inner pipe diameter} = 25 = 50 \text{mm} \]
\[ F = \text{Dimensionless force} = [\int_{x_i}^{x_f} p_i dA] / p_i A_i \]
\[ K_f = \text{Loss coefficient} \]
\[ M = \text{Mach number} \]
\[ \dot{m} = \text{Mass flow rate} = \int n dA \]
\[ p = \text{Static pressure} \]
\[ q = \text{Volumetric flow rate} = \int u dA \]
\[ R_e = \text{Reynolds number} \]
\[ a = \text{Axial distance measured from orifice} \]
\[ r = \text{Radial height measured from pipe wall} \]
\[ f = \text{Flow coefficient} \]
\[ S = \text{Diameter ratio} = \sqrt{A_i / A_o} \]
\[ E = \text{Expansion factor} \]
\[ \zeta = \text{Ratio of compressible to incompressible loss coefficient} = \kappa / \kappa_i \]
\[ \xi = \text{Momentum correction factor} \]
\[ \rho = \text{Density} \]
\[ \sigma = \text{Mass flow correction factor} \]

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3. Theory

3.1 Analytical flow model

A flow configuration of the present analytical model is shown in Fig.1. The flow relatively far downstream from the orifice can be assumed to be a one-dimensional flow. However, the radial distributions of velocities, densities and temperatures are recomendably not to be neglected for the accurate treatment of this type of flow analysis. In the present analysis, the mean flow conception\(^8\) is introduced, taking these radial distributions into consideration indirectly.

Assuming an adiabatic flow of perfect gas in the region from section 1 to section 2, a control surface, the continuity, momentum and energy equations are derived as follows.

\[
\int p\omega dA = \int p\omega dA = m \tag{1}
\]

\[
\int \left( p_1 + \frac{1}{2} \rho u_1^2 \right) dA = \int \left( p_2 + \frac{1}{2} \rho u_2^2 \right) dA \tag{2}
\]

\[
\int \left( C_s T_1 + \frac{1}{2} \rho C_{\omega} \right) p_1 dA = \int \left( C_s T_2 + \frac{1}{2} \rho C_{\omega} \right) p_2 dA \tag{3}
\]

By using the following averaged quantities\(^9\),

\[
\bar{p} = \frac{1}{m} \int p \omega dA, \quad \bar{u} = \frac{1}{m} \int u \omega dA, \quad \bar{T} = \frac{1}{m} \int T \omega dA \tag{4}
\]

the energy flow at any cross section is expressed as follow.

\[
C_s \int T \omega dA = \frac{1}{\varepsilon - 1} \int p \omega dA - \frac{1}{\varepsilon - 1} \int \rho \omega T dA \tag{5}
\]

The relation \( \bar{u}^2 = \kappa R \bar{T} \) is obtained by equations (4) and (5). If the section 1 and 2 are located relatively far from the orifice, the pressure gradient \( \partial p / \partial y \) can be neglected. In this case, it is confirmed that the series of equations, the equation of sound velocity and energy per unit mass, expressed by averaged quantities hold just as uniform flow without any correction factor.

\[
p = \bar{p} \bar{T}, \quad \bar{u} = \bar{u}, \quad \bar{T} = \frac{\varepsilon - 1}{\varepsilon} \bar{T}, \quad M = \bar{M} \tag{6}
\]

By use of those relations, and presuming \( p, \bar{M} \) and \( C_p \) (which will be discussed later) to be constant, equations (1) to (3) are developed in to the following equations \( [ \) the indication of the averaged quantity, \( (\cdot) \) is omitted from equations (7) to (15) \( ] \).

\[
\frac{\rho u_1}{A_1} = \frac{\rho u_1}{A_1} = m \tag{7}
\]

\[
\left[ 1 + \frac{2\bar{u}^2}{\sigma} \left( M_1^2 - 1 - \bar{M}_1^2 \right) \right] \frac{p_1}{p_1} + \frac{F_1}{F_1} = \frac{1 - \frac{2\bar{u}^2}{\sigma}}{2 \bar{M}_1} \left( M_1^2 - \bar{M}_1^2 \right) \frac{p_1}{p_1} = F_1 \tag{8}
\]

\[
\bar{T}_1 = \frac{\bar{T}_1(\bar{u}_1)^2}{\bar{T}_1(\bar{u}_1)^2} + 2 + \frac{2\bar{u}^2}{\sigma} \left( M_1^2 - 1 - \bar{M}_1^2 \right) \frac{p_1}{p_1} \tag{9}
\]

where, \( \sigma = \xi = 1 \) implies a uniform flow condition.

\[
\sigma = \frac{\varepsilon}{\sqrt{(1 - (\varepsilon - 1))}} \tag{10}
\]

\[
\xi = \left[ \frac{\int p \omega dA}{(\bar{M})} \right] \tag{11}
\]

On the assumption of adiabatic, nonviscous and uniform flow in the region from the inlet of the orifice to the vanes of the orifice, a relation between the pressure ratio \( p_b/p_1 \) and the inlet static-to-total pressure ratio ( or inlet Mach number \( M_1 \) ) in obtained by the following equation.

\[
\left[ \frac{p_1}{p_1} \right]^{1 - \gamma} \left[ \frac{1 - \gamma}{\gamma - 1} \right]^{1 - \gamma} \left[ \frac{1 - \gamma}{\gamma - 1} \right]^{1 - \gamma} \left[ \frac{1 - \gamma}{\gamma - 1} \right] \frac{p_1}{p_1} \tag{12}
\]

since \( p_1/p_1 = 1 \) in subsonic flow case and \( p_1/p_1 = (2/(\varepsilon + 1)) \) in supersonic flow case, the value \( \left[ \frac{p_1}{p_1} \right]^{1 - \gamma} \left[ \frac{1 - \gamma}{\gamma - 1} \right]^{1 - \gamma} \left[ \frac{1 - \gamma}{\gamma - 1} \right] \) at given \( p_1/p_1 \) can be calculated if the value of compressible contraction coefficient \( C_{\omega} \) is known.

In a real fluid flow, the velocity coefficient \( C_v \) \( ( = M_1^2 / M_1 \omega ) \) becomes slightly less than unity due to the presence of a flow loss in the region from the inlet to the vena contracta. According to the authors' results\(^5\), this viscous loss effect is approximately estimated by the constant velocity coefficient \( C_{\omega} = 0.97 \). The details of this procedure including the treatment of \( C_v \) are already discussed in reference (5), and will not be repeated here.

By taking the region from section 1 to section \( \phi \) (also \( \delta \) ) as a control surface, the theoretical dimensionless loss acting on the inlet face of the orifice \( P_{th} \) is expressed by.

\[
P_{th} = \left[ 1 - C_{\omega} \left( \frac{p_1}{p_1} \right)^{1 - \gamma} \left( \frac{1 - \gamma}{\gamma - 1} \right)^{1 - \gamma} \left( \frac{1 - \gamma}{\gamma - 1} \right)^{1 - \gamma} \right] \frac{2\bar{u}^2}{\sigma} \left( \frac{p_1}{p_1} \right)^{1 - \gamma} \tag{13}
\]
For the calculation of this flow model, the following steps are required:

1. Specify the value $\beta$ and $\rho_b/\rho_1$
2. Determine $M$, $p_b/p_1$ and $C_0$ from equation (1) by using the authors' procedure
3. Calculate $F_{ch}$, $p_{ch}$, $F$, $p_b$, $C_0$, and set the values $\sigma$, $\xi$ and $\mu_0$ (which will be discussed later)
4. Evaluate $M$ from equation (8) and (9).

The overall total pressure ratio and overall static pressure ratios are given by equations (14) and (15) respectively. Then the quantities required for the calculations of the loss and resistance of orifices and nozzles in flows are determined.

\[
\frac{M_2}{M_1} = \frac{T_2}{T_1} \left( \frac{p_1}{p_2} \right)^{1/\gamma} \quad \text{(14)}
\]

\[
F = M \left( \frac{T_1}{M_1} \right)^{1/\gamma} \quad \text{(15)}
\]

3.2 Loss coefficient

The amounts of the overall total pressure loss of fluid meters are commonly expressed in non-dimensional form as the loss coefficients, the total pressure loss being divided by a dynamic pressure or a differential pressure.

\[
K_i = \frac{(p_1 - p_{in})}{(p_1 - p_{out})} \quad \text{(16)}
\]

\[
L = \frac{(p_1 - p_{in})}{(p_1 - p_{out})} \quad \text{(17)}
\]

The inlet dynamic pressure is normally used in the expression (16) for the convenience of comparison with the conventional pipe wall resistances. The relation between $K_i$ and $L$ is derived by using the conventional mass flow equation.

\[
L = \frac{\pi d^3}{4 \rho_1 C_0} \quad \text{(18)}
\]

Equation (19) being derived from equation (18), the ratio of compressible-to-incompressible loss coefficient $\xi$, is introduced in this paper to express and discuss the effect of compressibility affecting the flow loss.

\[
\xi = \frac{K_1}{K_i} = \frac{L}{C_0} \quad \text{(18)}
\]

In spite of a considerable number of proposed experimental formulae for $K_i$, and $L$, no discussion on their reliability and/or effectual usage has been made. Therefore, those questions will also be resolved in this paper. On the other hand, a theoretical expression of incompressible loss coefficient has been proposed by Benedict, and equation (20) is derived from his theory.

\[
L = \frac{1 + 2 \alpha \beta ^{1/2} (\beta - 1)}{C_0} \quad \text{(20)}
\]

4. Experimental apparatus and procedure

Since the arrangement of the present experimental apparatus is almost the same as used in reference (5), only the details of the orifice face pressure measurements ($p_{ch}$ and $p_{in}$, see Fig. 1) and energy flow measurements, are described in this section.

For the pressure measurements at the inlet and the outlet faces of orifice, suitably placed three or four pressure holes (0.8 mm diam.) in the radial direction of the duct are drilled perpendicularly to the orifice plane. And the pressures at those points are led through 2 mm diameter holes which are drilled from the outside face of the orifice into the points in order to avoid any influence on flow patterns, to each of the manometers separately. The pressures are averaged by equation (21).

\[
\rho = \frac{\int p_{in} dA}{A_2 - A_1} \quad \text{(21)}
\]

where, the pressure at the inlet edge of the orifice ($p_{ch}$) is evaluated by the one-dimensional theory, and $p_b$ is substituted for the pressure at the outlet edge ($p_{out}$). It has been confirmed that these values are almost the same as the extrapolation value determined from the face pressure curve.

Total and static pressures are measured by use of a pitot tube (1.2 mm o.d.). And $u_s$, $p$, and $M$ are calculated using these total and static pressure data by assuming a constant total temperature in the radial direction of the duct. Mean total-to-static pressure ratios are determined from equation (22).

\[
\frac{p}{p_b} = \frac{\int (p_{in}/\rho u dA)}{u_s} \quad \text{(22)}
\]

The results shown in the following sections are arranged mainly about the orifice cases of interest.

5. Results and discussions

5.1 Analytical results and correction factors

The averaged pressures of inlet and outlet faces of orifices and their empirical relations are shown in Fig. 2. It is seen in both cases that the deviation from ideal conditions increases with an increasing $\beta$. The value of $F$ is less than that of $p_{ch}$ calculated from equation (13), while the value of $p_{ch}$ is larger than that of $p_b$. The reason for $F/F_{ch}<1$, is probably due to
the existence of a corner eddy at the bottom face of the orifice and of steeper expansion at the edge of the orifice, which is impossible to predict by the one-dimensional theory. And the reason for $P_b/P_0 > 1$, is likely to be that a reverse flow in the dead air region, being generated by the entrainment effect of the jet, pushes the back face of the orifice.

As for the value of wetted length $\bar{z}_0$, by present experiments described later $\bar{z}_{0f}$ is the length measured from the upstream side of the control surface to $z/D = 0.3$, and $\bar{z}_{0d}$ is the length measured from $z/D = 4$ to the downstream side of the control surface in orifice cases and from $z/D = 5$ in nozzle cases. As will be seen in the latter section, these values correspond to limited locations where the variation of the radial distribution of velocities with axial direction due to the existence of the orifice is hardly recognized, and also correspond to the limiting points where hardly any variation of $f_{L}$ (based on eq.(22), see Fig. 3) as well as the wall pressure (see Fig. 10) with axial distance appears. Those results provide a verification of equation (8) which is derived by assuming $f_{0}$, $M_{f}$, $C_{f}$ = constant within the wetted region $\bar{z}_{0f}$.

For the value of the pipe friction factor $4\bar{f}$, it has been confirmed that incompressible data are available for subsonic flow conditions at least. In this paper, Kármán–Nikuradse relation for the hydrodynamically smooth pipes is used for the $C_{f}$ calculation.

The values of mass flow and momentum correction factors obtained by the pitot tube traverse experiments are $\alpha = 1.008$, $\beta = 1.002$, $\xi = 0.997$, $\zeta = 0.999$, and are nearly constant irrespective of the orifice pressure ratio, $\beta$ and the shape of constrictions.

The comparison between present analytical results and "simple" results (which implies no correction factors and $\tau_0$ in the calculation, and corresponds to the previous analytical result) is made in Fig.4 for larger $\beta$ cases, in which a remarkable difference in the results between them appears. It is recognized from Fig.4 that the present analysis interprets and predicts the experimental results rather better than the "simple" theory.

5.2 Incompressible loss coefficient

Conventionally used incompressible loss formulae and their value demonstrated by the $K_{L}$ value using equation (18), are shown in Fig.5. Where, $\alpha$ for vena contracta tap system is selected for the conversion of $L_{g}$ to $K_{L}$ for orifice cases, and $\alpha$ for long radius flow nozzle (whose pressure tap system corresponds to the vena contracta tap system for flow nozzle case) is for the conversion, because of the $\alpha$ meanings.

For reference, the typical results, based on $\alpha$ of the corner tap and that of ISA1932 nozzle (which corresponds to the corner
tap system for flow nozzle case), are also plotted in Fig.5.

Significant discrepancy due to the α difference for the conversion is hardly recognized in the orifice case, and the values calculated from these loss formulae are in good agreement with ASME and present experimental data. For flow nozzle case, however, a considerable discrepancy caused by the α difference is considerably recognized particularly for larger β. In Fig.5, the numerical factors of the JSME Handbook's equation are "modified" so as to fit the present and ASME experimental data. As to this "modified" equation and Benedict's equation (20) cases, good agreement with the experimental values is seen in the case of the value α for long radius flow nozzle being adopted for the conversion. On the other hand, it is recommended to use the α of ISA1932 nozzle for JIS,z8762 equation and JSME Handbook's equation.

5.3 Estimation of the compressible flow loss

For the purpose of providing design data for various plants and pipe systems, the values of the loss ratios ζ (=Eلم/ست) obtained by the present theory and present experiment are shown in Fig.6 by broken lines and plots respectively. It is recognized that the compressible flow loss of orifices and nozzles can be predicted by the present theory with a high accuracy in a wide pressure and β range. For the convenience of practical use, an empirical equation for ζ obtained by present investigation is also proposed in this figure (which is applicable to subsonic flow regime).

By the way, the pressure ratio $p_b/p_1$ is selected for the abscissa of Fig.6 and the independent variable of the empirical equation. The value of the inlet flow Mach number $M_1$ can be calculated in accordance with equation (12). And, a practically convenient relation (23) which is based on the conventional mass flow equation, can be used to evaluate $M_1$ for the flow conditions of $p_b/p_1 \geq 0.5$.

$$M_1 = \alpha^{\beta}(1 - p_b/p_1)^{1/\gamma}$$

(23)

5.4 Flow characteristics and the loss mechanism

The axial variation of the pressure loss coefficient expressed by the term $L$ is shown in Fig.7(a), where the vertical width of the plot implies the $L$ variation with the pressure ratio $p_b/p_1$, and the arrow indicates the direction of $L$ variation with a decreasing $p_b/p_1$. From this figure, the general characteristics of the flow can be summarized as follows:

[1] The flow loss increases with a decreasing $\beta$ (the loss of enlargement).
[2] The loss increases with a decreasing $p_b/p_1$ (due to the presence of compressibility) *.
[3] Major part of the total dissipation energy arises in the outlet region in the vicinity of the orifice.
[4] The loss at the inlet region of the orifice is very small compared with that at the downstream region.

* The amount of the loss being indicated by the term $L$, looks as if only weak influence of compressibility arose, and the reason can be understood from equation (18).
the symbols shown in the parenthesis indicate the positions indicated by the same symbols in Fig.7(b).

Fig.8 Variation of velocity profile with axial distance \( P_b/P_I = 0.7 \)

Fig.7(b) shows a detail of the terms [3] and [4] by expressing the amount of loss as a local-to-overall loss ratio, which is divided by \( z/D = 10.6 \). It is recognized from Fig.7(b) that the major part (60–70%) of the overall dissipation energy arises within one-pipe diameter downstream of the orifice. The reason that the loss curve at larger \( \beta \) continues considerably far downstream is probably due to the effect of the pipe friction.

The radial distribution of velocities at several positions in the axial direction has been observed, and the results at each location are shown in Fig.8. The variation of the flow patterns from the jet structure near the vena contracts to the flow reattachment, and to a re-developed flow region, is well characterized in Fig.8. And it is confirmed that the flow region from the orifice to one-pipe diameter downstream, at which the major part of the total dissipation of the flow energy arises, is a dead air region with a reverse flow. In this region, as is generally known, a severe exchange of the kinetic energy in the radial direction occurs with the aid of the shear layer originated at the orifice edge. And the reason that the larger dissipation of the energy arises in this region, could be interpreted such that an effective kinetic energy will be largely reduced to non-axial energy and/or energy of turbulence.

The reattachment points (or regions) are observed to be from about 1.0 \( (\beta = 0.8) \) to 2.5 \( D \) \( (\beta = 0.3) \) downstream of the orifice. And a variation of the location of the reattachment points with the pressure ratio is hardly recognized. On the other hand, a variation of the velocity profile with the pressure ratio is confirmed to some extent (see Fig.9). Furthermore, Fig.9 shows that the contraction coefficient increases (i.e., decrease in enlargement loss) with a decreasing pressure ratio (the phenomenon is the counter effect of compressibility).

Fig.9 Variation of the velocity profile with \( P_b/P_I \) and \( z/D \)

Though an exchange of the kinetic energies continue further downstream of the reattachment point \( (\text{see Fig.8(b) to (e)}) \), a considerable amount of the axial kinetic (or effective) energy is conserved in such a region because of the flow being restricted by the duct. In such manner, a velocity profile flattening and static pressure recovery occur (see Fig.10) as the result of this exchange. Therefore, the loss rate gradually decreases with the axial distance (see Fig.7), and the velocity profile gradually approaches a fully developed profile, where the pipe wall friction dominates the flow loss.

Fig.10 Relation between wall pressure and the velocity profile

Fig.10 shows a relation between the axial distribution of wall pressures and the radial distribution of velocities. It is seen from Fig.10 that the pressure recovery in the region from the orifice to the reattachment point (the width of the plot, illustrated by a shaded portion, indicates the experimental uncertainty), is considerably small, whereas a large recovery of the pressure does occur in the region from the reattachment point to the point at
which the irregularity of the velocity profile due to the existence of the orifice almost disappear (i.e., maximum pressure recovery point).

The velocity distribution in the inlet flow region of the orifice are shown in Fig.11. It is recognized that a fully developed turbulent velocity profile is established, and maintained up to 0.7D ahead of the orifice. The shape of the velocity profile in such flow region appears to be identical irrespective of the pressure ratio and β. However, closer to the orifice, a flow configuration with a large acceleration and/or deceleration is observed. And the presence of an inlet loss, as shown in Fig.7(b), seems due to this effect. By the way, a re-developed velocity profile far downstream (see Fig.8(f)) is also plotted in Fig.11 as a reference. A difference in the fully developed velocity profile between upstream and downstream of the orifice is recognized. The reason could be interpreted such that the influence of high intensity of turbulence generated by the orifice continues considerably downstream.

6. Conclusions

The results of the present investigation are summarized as follows:

1] An analytical treatment of a compressible flow through orifices and nozzles, considering the effects of pipe wall friction, the variation of orifice face pressures and the non-uniformity of flow properties, is proposed in this paper. And the compressible flow loss is precisely estimated by this theory.

2] By expressing the amount of the loss as a ratio of loss coefficient of compressible flow ζ to that of incompressible flow ζ, ζ = ζ/K, practically convenient experimental formulae and figures are proposed. Furthermore, the reliability of the conventional incompressible loss formulae being used for the reference value is also discussed.

3] The results of energy flow and pressure measurements show that the major part (60-70%) of the total dissipation energy arises within one-pipe diameter downstream of the orifice, and a large pressure recovery does occur in the flow region from a reattachment point to the point where a significant irregularity of the axial velocity profile caused by the existence of the orifice almost disappears (this location corresponds to a maximum pressure recovery point).

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Appendix

A method of the loss estimation by means of a shock tube method

[1] An interpretation of the shock tube flow with an orifice and the usefulness of the shock tube technique. For a steady flow near an orifice established in a shock tube, the flow condition is physically similar to that in the conventional pipe orifice case because of the same principal equations, continuity, momentum, energy equation and equation of state, which govern each case within the flow region surrounded by the control surface. However, for the shock tube flows, the shock wave relation has to be considered in addition to these four equations. Therefore, as will be described below, it is necessary for the appearance of a contact surface that those equations be hold simultaneously.

The shock tube flow with an orifice may be characterized as follows (see Fig.12): After an incident shock wave interacts with the orifice, the flow corresponding to the initial condition will be attained through the physical iterations, being accompanied with unsteady complex wave systems. And the waves gradually attenuate with the lapse of time due to the presence of fluid viscosity. Since there exists a relationship between the pressure ratio across the
orifice and the upstream flow Mach number, a reflected shock wave will be generated which holds this relationship, and will travel in upstream direction. In the same manner, the downstream flow properties will also be set by the condition related with the pressure ratio. And a transmitted shock wave will appear and travel downstream to hold the relation between pressure and velocity of this flow region 3. By the way, the mass flow rate and the flow Mach number behind the transmitted shock wave do not coincide with those at the region 2, which is seemingly a physical inconsistency. But this inconsistency is solved by the appearance of a contact surface which divides the flow regions 2 and 3.

A shock tube is capable of producing any flow condition with considerable easiness and at lower experimental costs, unlike a conventional wind tunnel which involves difficulties, with respect to capacity, pressure ratio and an experimental safety. For this reason, a shock tube technique seems to be one of the useful methods for predicting the loss and resistance, especially under flow conditions with the orifice pressure ratio below 0.3.

[2] Procedure of the experimental analysis
The following procedure is used to determine the loss and resistance.
(1) Provide $\beta, \kappa$ (driven gas) and the strength of an incident shock wave ($P_0/P_4 \equiv P_{04}$) as the initial condition.
(2) Calculate the strength of reflected shock and transmitted shock waves ($P_{01}$, $P_{24}$) from the experimental data.

Fig.13 Flow Mach number obtained from present analysis using shock tube experimental data\textsuperscript{4,6}

![Flow Mach number graph]

Fig.14 Loss coefficient obtained from present analysis using shock tube experimental data\textsuperscript{4,6}

![Loss coefficient graph]

(3) Determine $M_{24}$, $M_{3}$ and $P_{01}$ from the following equations:

\[ M_{24} = \frac{(\tau-1)}{(1+\tau)(\gamma+\tau)P_{04}^{\gamma/2}} \left( \frac{P_{04}^{\gamma/2}}{P_{04}} - 1 \right)^{\gamma/2} \]
\[ M_{3} = \left( \frac{M_{24} P_{04}^{\gamma/2}}{\gamma+\tau} \right)^{2/\gamma} \]
\[ P_{01} = P_{w}^{\gamma/\gamma+\tau} P_{02} \]

Then, all the quantities needed for the calculation of loss and resistance are obtained. Using the method above and shock tube experimental data of Dadone and Pandolfi\textsuperscript{8}, typical calculated results are shown in Figs.13 and 14.

References