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An Examination of a High Speed Liquid Jet Apparatus*

(About the Effect of Convergent Nozzle on Jet)

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An apparatus, in which a liquid column is accelerated in a tube and a high-speed liquid jet is obtained through a convergent nozzle, is investigated. The high-speed liquid jet is obtained in a certain duration of time (velocity : 380 m/s, duration : about 1 ms), and the flow in the tube and the nozzle is calculated by the characteristic method of one dimensional flow. The main results are as follows.

(1) The pressure rise at the nozzle inlet and the velocity can be determined by the area ratio of the nozzle and the velocity of liquid close to the nozzle inlet.

(2) It is obtained that pressure rise at the nozzle inlet shows an overshoot due to a transient contract flow in the nozzle, and that the rate of overshoot becomes large as the area ratio increase.

(3) Corresponding to the pressure changes, there occur sprays in the jet. The effect of the nozzle shape on those is shown by jet photographs.

1. Introduction

Various types of apparatus have been developed⁽¹⁾⁻⁽⁴⁾ to obtain high speed liquid jet for use in coal mining, rock cutting or metal machining. But very few attempts were made to apply such liquid jets to studies of high speed liquid flow itself (for example, high speed liquid flow around a wedge). In this paper, a high speed water jet apparatus was mainly constructed and examined from the latter standpoint. The apparatus was designed to get the jet by accelerating a water column with a finite length in a horizontal tube by a compressed air and afterward discharging it from a convergent nozzle. This apparatus has the following advantages: High velocity jet can be under of a comparatively low pressure air and the velocity can be kept constant during a certain interval of time, though short. In the constructed apparatus, the water jet velocity of 380 m/s and its duration of 1 ms have been achieved.

This paper deals with a pressure change in time at the nozzle inlet, the velocity of the jet, its duration, and the shape of the jet, thereby clarifying the effect of the convergent nozzle on the jet. Experimental results were obtained from the measurements of the nozzle entrance pressure and taking photographs of the jet by means of various nozzles having different area ratios of the nozzle outlet to inlet and different nozzle shapes. The jet was discharged into an air

of vacuum pressure (20~30 mm Hg) with little interference of air so that the effect of the nozzle could be more clearly known. Also the flow in the tube and the nozzle was analyzed as a one-dimensional compressible fluid flow, using the characteristic method and the numerical integration method for comparison with experimental values. From the comparison of them, it has been confirmed that the calculation is effective within a time duration in which the flow in the nozzle is steady. But some questions were pointed out about the jet duration which is important in the actual application. Leach⁽⁴⁾ and O'Keefe⁽⁵⁾ studied the effect of various nozzles on the high speed water jet. However, the former dealt with a steady jet into atmosphere and the latter with an extremely high speed jet obtained in case of the water column being shorter than the nozzle. Thus, their experimental conditions differ greatly from those discussed in this paper.

2. Apparatus and the method of measurement

2.1 Outline of the apparatus and experimental conditions

The experimental apparatus consists of three components which are an equipment to produce a jet, a test section, and a receiver. The equipment to produce a jet is made up of an air receiver (inside diameter of 150 mm and length of 1,000 mm), a horizontal tube (diameter of 20 mm and length of 6,000 mm), and a convergent nozzle fixed to the end of the tube. The test section (a tube of 250 mm diameter and length of 480 mm) is connected to above mentioned equipment. The test section has square windows (150 mm(L) × 480 mm(W)) on both the sides to observe the jet visually. The receiver is a cylindrical tank of 1,000 mm

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inside diameter and 1,500 mm height; high speed water jet is finally dispressed by atmospheric air in it. A water column of a finite length is accelerated through the horizontal tube by compressed air from the air reservoir. Then the high speed column is discharged into the test section from the convergent nozzle. The water column is set by pouring water into a part of the horizontal tube partitioned with thin diaphragms. A ball valve is used to release the compressed air into the tube. In this experiment, the column length was limited to one meter and the driving pressure of air was varied from 10 to 30 kg/cm². In this experimental range, the initial velocity u_0 m/s of the column at the nozzle inlet was approximately given by the relation $u_0 = 14.2\sqrt{p_0}$, where the deviation in this case from experimental values is within 3%. The less the interference between jet and air is, the more clear becomes only the effect of the nozzle. The pressure inside the test section was set at 20 to 30 mm Hg, which is almost equal to the vapor pressure of water in experiment. A compressed air acts upon the column through a brass piston (outside diameter of 19.9 mm, length of 20 mm and weight of 250 g). This piston can keep the column unchanged in shape while being accelerated and strengthens the force pushing the column into the nozzle. The piston also serves to prevent a high compressed air from discharging into the test section after the column passes through the nozzle.

2.2 Nozzles for testing

The convergent nozzle in the above-mentioned equipment has two functions: one is for converting kinetic energy of high speed column into pressure head (water hammer pressure), and the other for converting the pressure head into kinetic energy of the high speed jet, then the velocity in the head section of the column is increased, as a result. Here, the area ratio of the nozzle outlet to inlet and the nozzle shape, which are especially important factors, are discussed mainly. The final straight section of nozzle is also dealt with.

Fig.1 Convergent nozzles for testing

Figure 1 shows the inside shape and dimensions of the convergent nozzles used in this experiment, where β and k are the area ratio of nozzle outlet to inlet and the exponents indicating the nozzle shape, respectively. Figure 1 (a) shows conical tapered nozzles to examine the effect of the area ratio; the contraction angle was made small so that the flow in the nozzle might be a one-dimensional flow. The outlet of each conical tapered nozzle was provided with a straight tube section of three times the length of outlet diameter. The nozzles shown in Fig. 1 (b) are used to examine the effect of the nozzle shape. With the area ratio and the nozzle length kept constant, the diameter change of the convergent section along the nozzle axis is given by the following exponential function.

$$D(x) = D_0 \left\{ 1 - \left(1 - \frac{D_n}{D_0} \right) \frac{1 - e^{kx/L_n}}{1 - e^{k}} \right\} \dots \dots (1)$$

where x is the distance from the nozzle inlet, D_0 the nozzle inlet diameter, D_n the nozzle outlet diameter, L_n the convergent section length, and k the exponent.

As known from the above equation and Fig. 1 (b) each nozzle shape is distinguished by the exponent k . In this paper these nozzles are named "the exponential nozzle". The final straight section in the exponential nozzles was made comparatively long, i.e. 10 times the length of the outlet diameter.

2.3 The method of measurement

In order to make clear the effect of each nozzle shown in Fig. 1 on the jet, the initial velocity of the water column and the pressure change in time at the nozzle inlet were measured, and also the shape of jet was photographed. A schematic diagram of this measurement is shown in Fig. 2. The initial velocity of the water column at the nozzle inlet was obtained from the signals of the column passing through two points near the nozzle inlet (see Fig. 2-A1 and A2). The column velocity has been confirmed to be nearly constant in that section. Temporal pressure change at the nozzle inlet was detected by means of a quartz pressure transducer (Fig. 2-B1). Three xenon tubes C5 C6 and C7 in Fig.2 were flashed to take triple pictures of a jet on a film so that

Fig. 2 Schematic diagram of measurement

the jet shape and its temporal change could be clarified. To distinguish the triple pictures each time, blue, red, and green filters were applied to light sources. Pressure rises at the nozzle inlet was used as a trigger for the xenon tubes.

3. Experimental results

3.1 Pressure change at nozzle inlet

Pressure changes in time measured at the nozzle inlet are shown in Fig. 3. Fig. 3 (a) and (b) show the measurements of the conical tapered nozzle and the exponential nozzles, respectively. These pressure changes were obtained by a driving air pressure of 20 kg/cm² with the initial velocity of the water column at the nozzle inlet being $u_0=64$ m/s. It can be seen from the pressure forms in Fig. 3 that, when the head section of the column enters the convergent nozzle,

(a) Conical tapered nozzles ($k=0$)

(b) Exponential nozzles ($\beta=1/16$)

Fig. 3 Pressure changes at nozzle inlet ($u_0=64$ m/s)

pressure at the nozzle inlet first rises to a constant value, p_1 , related to the initial velocity of the column and the area ratio. Then the initial pressure rise is reflected by the front surface of the piston at the rear of the column and the reflection wave causes a pressure rise at the nozzle inlet. During this period, a reaction force decelerates the piston and its effect gradually appears in the form of pressure decay.

The observed pressure changes show that there is an overshoot in first pressure rise. This overshoot is especially remarkable in case of the conical tapered nozzle with a large area ratio (i.e. T-1 nozzle). The rate of overshoot, which is defined by the ratio of the peak value, p_{s1} , in the first pressure ratio to p_1 , is shown Fig. 4 with respect to the area ratio β and the exponent k , respectively. It shows that

Fig. 4 Rate of overshoot in initial pressure rise

the larger the area ratio becomes, the greater the overshoot rate. On the other hand, no remarkable feature of the overshoot due to k can be recognized in the exponential nozzle with a small area ratio. From these facts and the analysis described later it will be seen that, when the head section of the column pushes into the convergent nozzle, there will occur a transient contract flow which makes the effective area ratio smaller than the actual one. Therefore, the absolute value of such overshoot can not exceed the pressure rise in the area ratio $\beta=0$.

Pressure changes in Fig. 3 show that the reflection wave from the front surface of the piston has an almost discontinuous change compared with the initial pressure rise. This is probably due to the fact that a pressure rise causes the sound velocity in water to increase, distorting the reflection wave. Since the exponential nozzles are shorter than the conical tapered nozzles, the initial pressure rises in the exponential nozzles are steeper and the reflection waves from the front surface of the piston are also steeper and higher.

3.2 Photograph of jet shape

Photographs corresponding to each pressure change given in Fig. 5. As mentioned in the section 2.3, the triple pictures of the jet obtained each time can be distinguished by respective colors of the jet.

In the jet of the conical tapered nozzle shown in Fig. 5 (a), there are

hemispherical sprays to a great extent at the jet head and at other points of the jet. This is explained by the fact that, if the velocity of the jet at the nozzle outlet increases suddenly, then the following high speed jet impinges on the leading low speed jet. Dunne et al.⁽⁶⁾ analyzed such spray forming of the jet adopting the conservation law of momentum and found the moving velocity of the junction of two jets. As the jet velocity in the front and rear of the spray in this experiment is not measured, there is no way of confirming that

(a) Conical tapered nozzle

(b) Exponential nozzle

G: Green R: Red B: Blue

Fig. 5 Jet shape and its change in time ($U_0=64$ m/s)

calculation. From the jet photographs of the conical tapered nozzles in Fig. 5 (a), we can obtain a position-time diagram of the hemispherical sprays as shown in Fig. 6. The origin of the abscissa in this figure is taken at the nozzle outlet. Comparison of this diagram and the pressure changes at the nozzle inlet shown in Fig. 3 (a) tells us that the sprays caused by the initial pressure rise and by the secondary pressure rise due to the reflected pressure wave from the piston. Therefore the jet flow between the first and second sprays corresponds to the flow caused by the pressure which is kept nearly constant at the nozzle inlet.

Fig. 6 Position-time diagram of jet for conical tapered nozzle ($U_0=64$ m/s)

In case of a conical tapered nozzle, the jet flow described above has a comparatively good uniformity. Conversely, the jet of the exponential nozzle shown in Fig. 5 (b) has a wide radial spread. Since the exponential nozzle has a long final straight section in the nozzle outlet, a boundary layer will be developed there. It seems that the development of the boundary layer deprives the flow of its uniformity, and causes a radial spread of the jet. Leach et al.⁽⁴⁾ have pointed out that a decay of stagnation pressure in the steady jet becomes smallest if the final straight section of the conical tapered nozzle has about two or three times the length of its outlet diameter.

3.3 Pressure rise at nozzle inlet and flow velocity at nozzle outlet

Pressure changes in Fig. 3 show that pressure at the nozzle inlet becomes a constant value for a certain interval of time. This indicates that a flow in the nozzle attains temporarily a steady state. The pressure rise at the nozzle inlet and the jet velocity during the time interval were analyzed and compared with experimental values. Main symbols used here are as follows:

- ρ_0 : density of water
- C_0 : sound velocity of water
- U_0 : initial velocity of water column
- U_1 : flow velocity at nozzle inlet
- U_2 : velocity of jet at nozzle outlet
- p_1 : pressure rise at nozzle inlet
- β : area ratio of nozzle outlet to inlet
- ζ : loss coefficient of nozzle
- t_1 : jet duration
- l_0 : water column length

Pressure rise in the nozzle inlet is

caused by reduction of water column velocity from u_0 to u_1 . Therefore the pressure p_1 is given from the conservation of momentum as follows:

$$p_1 = \rho_0 c_0 (u_0 - u_1) \quad (2)$$

On the other hand, assuming the flow in the nozzle is steady and compressibility of water is negligible, the equations of energy and continuity are

$$(p_1/\rho_0) + (u_1^2/2) = (1+\zeta)(u_0^2/2) \quad (3)$$

$$u_1 = \beta u_0 \quad (4)$$

Eliminating u_2 from the above equations, we have

$$p_1 = \{\beta^2(1+\zeta)-1\}(\rho_0 u_0^2/2) \quad (5)$$

From the equations (2), (4) and (5), p_1 , u_1 , and u_2 can be determined for given values of u_0 , β and ζ . Using the dimensionless parameter,

$$M^* \equiv (\beta^2 - 1)(u_0/c_0), \quad \zeta^* \equiv \zeta/(1-\beta^2)$$

the results are given by the following formulae,

$$p_1/\rho_0 c_0 u_0 = 1 - r(M^*, \zeta^*) \quad (6)$$

$$u_1/u_0 = \beta u_0/u_0 = r(M^*, \zeta^*) \quad (7)$$

where

$$r \equiv 2 / \{1 + \sqrt{1 + 2(1 + \zeta^*)M^*}\} \quad (8)$$

If the compressibility of water is estimated from the equation of state (cf. ref. 7)

$$p = B \{(\rho/\rho_0)^n - 1\} \quad (9)$$

where, $B = 3,000 \text{ kg/cm}^2$; $n = 7$, the fundamental equations (2), (3), (4) are modified respectively as follows:

$$2\{c_1/(n-1)\} + u_1 = 2\{c_0/(n-1)\} + u_0 \quad (10)$$

$$\{c_1^2/(n-1)\} + (u_1^2/2) = \{c_0^2/(n-1)\} + (1+\zeta)(u_0^2/2) \quad (11)$$

$$\rho_1 u_1 = \beta \rho_0 u_0 \quad (12)$$

where c_1 and ρ_1 represent the sound velocity and the density of water corresponding to the pressure p_1 , respectively. The sound velocity of water is a function of pressure as follows;

$$c \equiv \sqrt{dp/d\rho} = c_0(1+p/B)^{(n-1)/2n} \quad (13)$$

The pressure rise p_1 and the jet velocity u_2 against the initial velocity u_0 are given in Fig. 7 and Fig. 8 respectively. Solid lines and dotted lines in both figures

Fig. 7 Pressure rise p_1 at nozzle inlet

Fig. 8 Jet velocity u_2

show the calculated results in case of constant sound velocity of water (i.e. $C_0 = 1,435 \text{ m/s}$) and those taking account of the pressure dependence of the sound velocity by means of the equation (13), respectively. In either case, the loss coefficient of nozzle was neglected. It is seen from calculated results that if the change in sound velocity of water is considered, the pressure rise at the nozzle inlet becomes high and the jet velocity increases.

Figure 7 shows that fairly good agreement between observed and calculated values is obtained in the conical tapered nozzle and that experimental values become comparatively higher than calculated ones in the exponential nozzle. This is due to the fact that, in this experiment, every exponential nozzle has a shorter convergent section in the nozzle outlet than the conical nozzles, thereby increasing pressure loss due to flow contraction and to wall friction. Figure 9 shows the difference between calculated values of pressure p_1 and experimental ones in each nozzle. Effects of β and k on the above differences are not so remarkable.

Fig. 9 Effect of nozzle parameter on pressure p_1

Figure 8 shows that measured values of the jet velocity are greatly lower than calculated ones. The ratio of measured value u_2 to calculated value u_{2cal} is indicated against the nozzle parameters, and k in Fig. 10. As mentioned before, in the conical tapered nozzle, there is a large hemispherical spray in the jet head, therefore the scattering deviation in the measured values of velocity is large and the measured values are lower than calculated values on whole. This is due to leading low speed jets. On the other hand, in the exponential nozzle, the smaller k is, the measured values u_2 are so much the lower than calculated values. As it is seen from the nozzle shape (Fig. 2),

Fig. 10 Effect of nozzle parameter on jet velocity U_2

this is probably due to the fact that the smaller k is, the larger becomes the effect of pipe wall friction.

The measured results of p_1 and U_2 shown in Fig. 7 and 8 can be rearranged with use of the above-mentioned dimensionless parameters, as shown in Fig. 11 (a) and (b). It is seen from the figures that when M^* is large (namely, either the area ratio is small, or the initial velocity of water column is large), the effect of the change in sound velocity due to pressure cannot be disregarded. In case of the exponential nozzle, the calculated curve can almost coincide with the measured values when introducing an appropriate loss coefficient of nozzle.

In order to know the jet duration, we measured t_1 , which is the time interval from the peak of the initial pressure rise to the start of the second pressure rise measured at the nozzle inlet, and which is shown in Fig. 12. It is noted that the time interval t_1 thus defined cannot exactly express the steady jet duration since there is an overshoot decaying in the initial pressure rise as shown in Fig. 3. Two curves in Fig. 12 were obtained from the calculation of one round-trip wave-travel in the column represented by the equation,

(a) Pressure rise at nozzle inlet

Fig. 11 Representation of exponential result by dimensionless parameters

Fig. 12 Time duration t_1

$$t_1 = \frac{L_0(1+(c'/c''))}{c' + u_0} \quad (14)$$

where c' ($=c_0 - u_0$) is the absolute velocity of compression wave travelling from nozzle inlet to the piston, and c'' ($=c_1 + u_1$) the absolute velocity of compression wave reflected by the piston. In this calculation, the nozzle length was neglected as compared with the water column length and results are given for $\beta=1$ (solid line) and for $\beta=0$ (dotted line). Figure 12 shows that the time interval t_1 becomes short as the interval velocity of water column at nozzle inlet increase. This figure also indicates that the measured value t_1 becomes much lower than the above calculated one. Because there is a disturbance in the head section of the water column and also there is the effect of the nozzle length. The effects of β and k on the duration t_1 are not so remarkable as seen in Fig. 13 which represents the ratio of the measured value t_1 to calculated value $t_{1,cal}$ against each parameter.

Fig. 13 Effect of nozzle parameters on duration t_1

4. Discussion of pressure change by characteristic method

4.1 The method of calculation

In the above calculation p_1, u_1, u_2 and t_1 were obtained by assuming that the flow in the nozzle is completely steady. One dimensional compressible fluid flow in the nozzle and the tube has been calculated by the characteristic method to examine in more detail the flow. The symbols used here are as follows:

- t : time
- x : distance from nozzle inlet
- ρ : density
- u : velocity
- p : pressure
- X : position
- X_f^D : position of water column head or of nozzle outlet
- D : inside diameter of tube and nozzle
- L_n : nozzle length
- f : pipe friction loss coefficient

W_p : piston weight

Symbols p_0 , c_0 , u_0 , D_0 and l_0 have been defined before.

Suppose that the diameter changes gradually along the nozzle axis and that the flow in the tube and the nozzle is one-dimensional, then the equations of continuity and motion will be expressed as follows;

$$\rho_t + u\rho_x + \rho u_x + 2\rho(D_z/D)u = 0 \quad (15)$$

$$u_t + uu_x + (1/\rho)p_x + (f/2D)u|u| = 0 \quad (16)$$

Initial conditions are

$$\left. \begin{aligned} t=0 : u=u_0, \quad p=-p_0(x/l_0) \\ x=X_f : X_p=-l_0, \quad X_f=0 \end{aligned} \right\} \quad (17)$$

Boundary conditions for $t>0$ are as follows:

$$x=X_p(t) : p=p_0 - [W_p / \{g(\pi D_0^2/4)\}](du_p/dt) \quad (18)$$

$$x=X_f(t) : p=0 \quad (19)$$

where X_f represents the position of the head of water column while the head is inside the nozzle and is equal to the nozzle length l_0 after the head has reached the nozzle outlet. Assuming that the sound velocity of water is constant, the equations (15) and (16) can be transformed into the following well-known characteristic forms

$$\begin{aligned} \text{along } dx/dt=u+c_0 \\ dp/dt + \rho_0 c_0 (du/dt) + \rho_0 c_0 \{2c_0(D_z/D) \\ + (f/2D)|u|\} u = 0 \quad (20) \\ \text{along } dx/dt=u-c_0 \\ dp/dt - \rho_0 c_0 (du/dt) + \rho_0 c_0 \{2c_0(D_z/D) \\ - (f/2D)|u|\} u = 0 \quad (21) \end{aligned}$$

If the initial condition is represented by Eq.(17), the boundary conditions by Eq.(18) and (19), and the nozzle shape $D(x)$ by Eq. (1), the above equations can be numerically integrated for variables $p(x,t)$, $X_p(t)$ and $X_f(t)$ on the x,t -plane.

4.2 Calculated results

Calculated examples by the above method are shown in Fig. 14. In this example, E-3 nozzle without the final straight section has been calculated under the following conditions: $\rho_0 = 102 \text{ kg s}^{-2} \text{ m}^{-4}$, $c_0 = 1435 \text{ m/s}$, $u_0 = 64 \text{ m/s}$, $p_0 = 20 \times 10^4 \text{ kg/m}^2$

$$l_0 = 1.00 \text{ m}, \quad W_p = 0.250 \text{ kg}, \quad f = 0.01$$

In this calculation, the water column is equally divided into 50 sections at initial state.

(b) Relation of state variables in water column

(c) Pressure changes in time

(d) Velocity changes in time

Figure 14 (a) shows the motion of the column and the typical pressure wave travelling in the x,t -plane (physical plane). Figure 14 (b) shows the relations between the variables p and u at the nozzle inlet and at the surface of the piston. In this figure two one-point chained lines represent the relation given by equations (2) and (5), with $\beta = 1/16$, and $\zeta = 0$. Fig. 14 (c) shows pressure changes at the nozzle inlet and at the surface of the piston. Figure 14 (d) shows the flow velocities at the nozzle inlet and outlet and the piston velocity. Each point corresponding to each of Fig. 14 is expressed by the symbols $A_1, A_2, \dots, C_5$ and C'_5 . In this result, the effect of pipe friction loss is very small, because the final straight section of the nozzle in the nozzle outlet was omitted.

In order to compare the above calculated results with experimental ones, both pressure changes at the nozzle inlet are shown in Fig. 15, which shows generally good agreement between them. However, the following differences can be pointed out from this figure. In the calculated results there is no overshoot in the initial pressure rise, the shape of the second pressure rise becomes relatively smooth and the time duration t_1 becomes long. These differences arise from the assumption that a transient contract flow in the nozzle. pressure dependence of sound velocity in water and a disturbance in the head section of the column can be neglected in the

(a) Motion of column and typical wave method

Fig.14 Numerical result obtained by the characteristic method (E-3 nozzle, $u_0 = 64 \text{ m/s}$)

calculations.

In case of the conical tapered nozzles, the calculated pressure changes at the nozzle inlet are shown in Fig. 16. Each curve in this figure corresponds to each one shown in Fig. 3 (a). From the comparison of both curves, it can be seen that the differences above mentioned between the calculations and measurements become more

Fig. 15 Comparison of pressure changes at nozzle inlet between experiment and calculation (E-3 nozzle, $U_0 = 64$ m/s)

remarkable.

O'Keefe et al. have treated the same calculation model as one discussed in this paper (5) taking account of the pressure dependence of sound velocity. However, their calculated results are fairly different from those in this paper, since the flow velocity is extremely high (the velocity range of U_0 is from 660 to 1770 m/s) and moreover the column is shorter than the nozzle.

5. Conclusions

For obtaining a high speed liquid jet, one obtaining a high speed liquid jet, one method is presented, and effects of some convergent nozzle on jet have been analyzed experimentally and theoretically. The main results obtained in this paper are follows.

(1) If a water column is long enough compared with nozzle length, an almost steady flow in the nozzle can be obtained for a certain time duration. The pressure rise at the nozzle inlet and the jet velocity at the nozzle outlet in that duration are determined by the area ratio of the nozzle outlet to inlet and the initial velocity of the water column at the nozzle inlet. These results have been rearranged by using dimensionless parameters

(2) The observed pressure change at the nozzle inlet are examined by the calculation for a one dimensional unsteady flow, then the following are pointed out.

(i) The actual pressure rise starts later than the calculated one, since there is a disturbance in the head section of the water column. Then the observed time duration mentioned above becomes shorter than the calculated one.

(ii) There is an overshoot appearing in the first pressure rise. The one dimensional flow analysis of compressible fluid can not explain this phenomenon. It seems to be for the reason that, when the column head section enters a convergent nozzle, there

occurs a transient contract flow within the nozzle, and then the effective area ratio of the nozzle becomes smaller than the actual one.

(iii) The reflected wave from the piston behind the water column becomes steeper and higher than the calculated one for the reasons given in (i) and (ii). This is also concerned with the wave distortion due to pressure dependence of the sound velocity.

The above differences become more remarkable as the area ratio of the nozzle becomes large and its convergent section long. In the exponential nozzle used here, the measured values are in good agreement with the calculated ones because of its small area ratio and short convergent section.

(3) From the photographs of the jet shapes, it is seen that a long straight section in the nozzle outlet may cause the jet to spread in radial direction under the effect of pipe wall friction in that section.

(4) There occur hemispherical sprays in the jet. The spray takes place because of a sudden increase in the jet velocity in the nozzle outlet. This velocity increase arises from the overshoot described above in (ii) of (2) and from the reflection of pressure wave from the piston. The jet flow between the first two sprays may have a nearly constant velocity, since the pressure rise at the nozzle inlet is kept almost constant in that duration.

(5) From the facts described in (2), the duration of the jet with nearly constant velocity seems to be much shorter than one round-trip wave-travel time in the water column. This becomes more important as the area ratio of nozzle becomes larger.

Fig. 16 Calculated pressure changes at inlet of conical tapered nozzle ($U_0 = 64$ m/s)

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