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A Study on a Turning Flow into Annulus from an Inner Pipe in a Concentric Double Pipe

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It is important to know a head loss of the flow in a concentric double pipe such as a double funnel. The present study gives its expermental data using various types of concentric double pipes, in particular, giving the attention to the loss of total head through a turning-flow region from the outlet of the inner pipe to the inlet of the annulus between the inner and the outer pipes. They are given in terms of a non-dimensional clearance length ξ between the outlet of the inner pipe and the bottom of the outer pipe for various values of area ratio (m=0.576~1.05) and Reynolds number ($R_e=4.2\times 10^3\sim 3.5\times 10^4$). It should be noted that all curves of coefficient of head loss against ξ have peaks at about ξ=0.875 and valleys about ξ=0.50.

1. Introduction

The present paper deals with a fluid flow issuing from an inner pipe and turning back into annulus of a concentric double pipe. This kind of flow can be very often found in engineering designs such as a chemical reaction tube and a concentric double funnel etc. Hence, it is very important to give the experimental data of loss of total head through the turning-flow region.

In 1934 Frey[1] reported the experimental data of pressure loss through a bend in a single channel for various forms of the 90° and 180° bends. However, the present study fundamentally differs from his study in the geometrical shape of the flow path.

In order to characterize the geometrical feature of the flow path three non-dimensional parameters should clearly be defined. The first is the ratio of the clearance length, which is the distance between the outlet of the inner pipe and the bottom of the outer one, to the inner pipe diameter, the second the areal ratio of the cross-sectional area of the annulus to that of the inner pipe, and the third the ratio of the cross-sectional area of the thickness of the inner pipe to the differene of the cross-sectional area between the outer and the inner pipes. Then, the experimental data of the coefficient of the loss of total head through the turning-flow region are given in terms of the non-dimensional clearance length for various values of the area ratio and of the Reynolds number. The effect of the thickness ratio does not appear in these results, since all double pipes used have almost no much differences in their values of thickness ratio as shown in Table 1. Further, a certain kind of instability is observed over a short range of the non-dimensional clearance length in the pressure measurement.

2. Experimental Apparatus and Procedures

Figure 1 shows a schematic diagram of a complete setup of equipment. The

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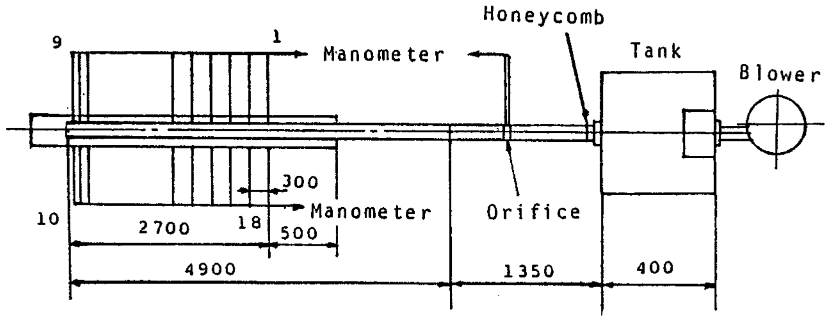

Fig. 1 Schematic diagram of complete setup of equipment.

air stored in the tunk enters the inner tube through a honeycomb and an orifice, issures from its outlet, turns back into the annulus between the inner and outer pipes and is finally discharged in the atmosphere.

A constant flow rate was maintained by controlling an electric current supplied to the blower using a variable transformer and was measured by the orifice which is calibrated beforehand. Two kinds of orifices were used, one ($\phi=19.5$ mm) for a larger amount of air flow and the other ($\phi=12$ mm) for a smaller amount, respectively. The inlet and outlet temperatures were measured by a thermocouple and a thermometer, respectively. Atmospheric pressure in the laboratory room was measured by an aneroid barometer. Then, the weight per unit volume of the air was calculated from the equation of state of the ideal gas, using the measured atmospheric pressure and the arithmetic mean value of the inlet and outlet temperatures. The averaged velocity of the flow in the inner pipe was calculated from the measured flow rate and the calculated value of the weight of unit volume of air. In order to give the value of Reynolds number the kinematic viscosity of the air was estimated for the mean temperature using a table of physical properties.

The outer and inner pipes used were commercial smooth ones made of drown-brass or vinyl chloride, their straightness being carefully examined. The concentic double-pipe was set up supporting the inner pipe by adjusting bolts screwed into the outer pipe. Various combinations of the outer and inner pipes are listed in Table 1 in terms of their dimensions, the area ratio m and the thickness ratio t, in which m and t are defined as follows:

Table 1 Various combinations of inner and outer pipes

No.	Inner pipe (mm)	Outer pipe (mm)	Area ratio (m)	Thickness ratio (t)
1	33.37×41.95	51.06×60.30	0.576	0.253
2	35.00×38.05	46.87×50.00	0.611	0.229
3	31.86×34.99	44.16×47.90	0.715	0.224
4	27.58×30.14	38.91×42.19	0.790	0.196
5	30.18×33.76	44.15×47.90	0.889	0.220
6	40.50×44.67	59.35×63.33	0.931	0.188
7	25.43×28.50	37.90×40.00	0.976	0.209
8	19.67×22.03	29.85×31.85	1.05	0.211

$$m=\frac{D_i^2-d_0^2}{d_i^2}\,, \tag{2.1}$$

and

$$t=\frac{d_0^2-d_i^2}{D_i^2-d_i^2}\,, \tag{2.2}$$

where D_i is the inner diameter of the outer pipe, d_0 and d_i the outer and inner diameters of the inner pipe, respectively.

The dimensions of the pipes were measured by a vernier caliper and an inside micrometer caliper. However, these measurements can be applied only to the inlet or outlet of a pipe for the measurements of the inner diameter. To obtain the mean value of inner diameter through a pipe, it was also calculated by weighing water filled in the pipe. The calculated values for various pipes are at most 1.2% greater than those measured directly at the inlet or outlets, which can be considered as mean inner diameters without much failure and are listed in Table 1.

Eighteen pressure taps were laid on the walls of the inner and outer pipes, nine for each, as shown in Fig. 1. The diameter of the holes of the taps was made to be smaller than the thickness of the pipes for accuracy of pressure measurment. A calming length was taken 50~70 times as long as the inner pipe diameter between the outlet of an orifice pipe and the first pressure tap on the wall of the inner pipe. The pressure at a tap was read by a Betz micromanometer whose reading error is within 0.02 mmAq. The pressures at 18 taps were read successively in a run of experiment fixing the values of the clearance length and the flow rate. Fairly steady values of pressure were read at the first five taps on each of the inner and outer pipes, since the distance between the fifth pressure tap and the outlet of the inner pipe was taken 40~60 times as long as the inner pipe diameter. Those values of pressure give the pressure gradient in the inner pipe as well as in the annulus. The remaining six taps gave violently fluctuating pressures, which shows that the flow is very turbulent in the turning-flow region.

The pressure was measured at each tap for various values of Reynolds number R_e in the range from 4.2×10^3 to 3.5×10^4 and for various values of the non-dimensional clearance lengths ξ in the range from 0.125 to 50. R_e and ξ being defined as follows;

$$R_e=\frac{V_i d_i}{\nu} \tag{2.3}$$

and
$$\xi=l/d_i\,, \tag{2.4}$$

where V_i is the velocity averaged over the cross-sectional area of the inner pipe, ν the kinematic viscosity estimated as above mentioned and l is the distance between the outlet of the inner pipe and the bottom of the outer pipe.

The total head loss H_t through the turning-flow region from the outlet of the inner pipe to the inlet of the annulus is given by Bernoulli's equation taking into account the head loss.

$$H_t=\frac{1}{\gamma}\,(P_i-P_c)+\frac{1}{2g}\,(V_i^2-V_c^2) \tag{2.5}$$

or
$$H_t=H_p+H_v\,. \tag{2.6}$$

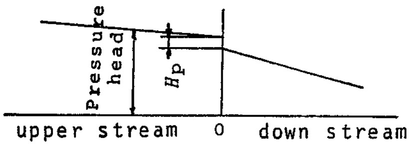

Fig. 2 Hydraulic gradient line along an inner and outer pipe.

where P_i and P_c are the pressures at the outlet of the inner pipe and at the inlet of the annular space, respectively, γ and g the weight per unit volume of air and gravitational constant, V_i and V_c the averaged fluid velocities over the cross-sectional area of an inner pipe and over an annular space, respectively. H_p and H_v are the loss of the pressure head and the loss of the velocitiy head, respectively. The first and second terms in the right hand side of Eq. (2.6) can be calculated as follows. The loss of the pressure head H_p through the turning-flow region is obtained by extending the measured hydraulic grade lines along the inner and outer pipes up to the outlet of the inner pipe and the inlet of the annulus, which are denoted by 0 in Fig. 2, and by reading the difference of the limitting values of pressure at 0. The hydraulic grade lines are calculated by applying the method of least squares to the measured pressure distributions along an inner pipe and outer one.

On the other hand, the loss of the velocity head H_v can be calculated from the averaged velocities in an inner pipe and an annulus. The non-dimensional loss coefficient is generally defined by

$$\begin{aligned}\zeta &= H_t/\left(\frac{V_i^2}{2g}\right)\\ &= (P_i-P_c)/\left(\frac{\gamma V_i^2}{2g}\right)+1-\left(\frac{V_c}{V_i}\right)^2 \end{aligned} \qquad (2.7)$$

Now, all the experimental data will be rearranged in non-dimensional forms by making use of (2.1), (2.3), (2.4) and (2.7), the effect of thickness ratio expressed by (2.2) being neglected, since its variation is small among the used combinations of the inner and outer pipes. Thus, the non-dimensional loss coefficient ζ will be expressed in terms of the non-dimensional clearance length ξ, areal ratio m and Reynolds number R_e.

3. Accuracy of Loss Coefficient

The accuracy of loss coefficient is estimated by comparing the friction factor λ_s' given in the present experiment with that given by the well-established formula, λ_s. First, Fig. 3 shows a typical relation between the friction factor for a smooth single pipe and Reynolds number. The full line is calculated from the Blasius formula ($\lambda_s=0.3164R_e^{-1/4}$), and the marks in the figure show λ_s' obtained in the present experiment. It is easily found that λ_s' has a maximum deviation of 13% in the range of Reynolds number from 4.2×10^3 to 5.0×10^3.

Next, Fig. 4 shows the friction factor λ_c' calculated by the pressure gradient obtained from the measured pressure distribution along the outer pipe as well as the known friction factor λ_c. The marks in the figure show λ_c' obtained in the present

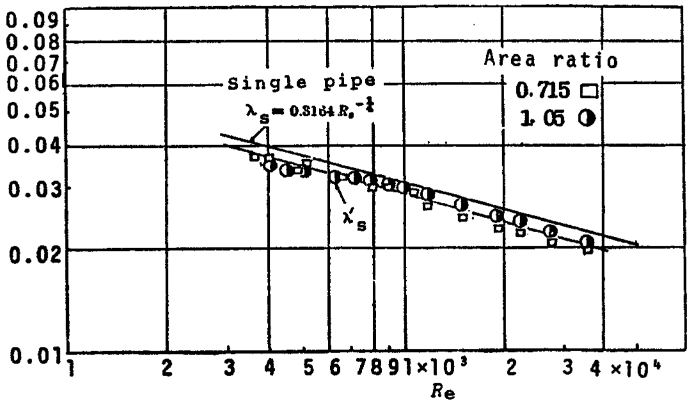

Fig. 3 Relation between friction factor λ and Reynolds number R_e.

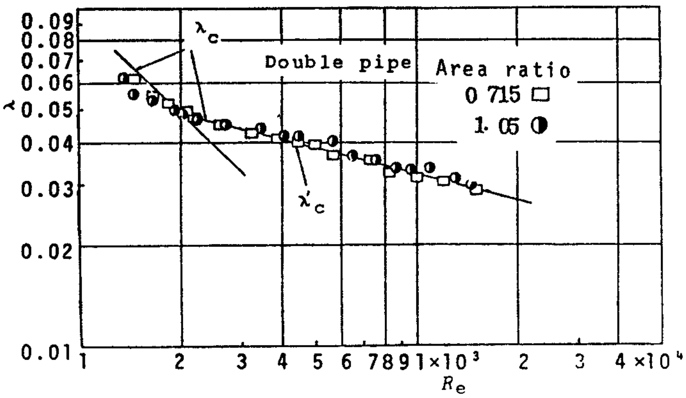

Fig. 4 Relation between friction factor λ and Reynolds number R_e.

experiment and the full line is calculated from the Asanuma formula.[2] Then, it can be found in Fig. 4 that these values of λ_c' are in satisfactory agreement with those calculated from Asanuma's formula in the almost whole range of Reynolds number except $R_e < 1.5 \times 10^3$. The error in the measurement of the inner diameter is at most 1.2%, as mentioned in the preceding sectiom, and the velocity head can be measured with a well-calibrated orifice without a serious error.

Thus, the considerable contribution to the error in the final results for the loss of the total head is due to the error in the measurement of the friction factor in the inner pipe.

4. Experimental Results and Discussions

4.1 Loss coefficients

Figures 5–8 show the experimental corelations between the loss coefficient ζ and the non-dimensional clearance length ξ for the various values of the non-dimensional area ratio ($m=0.576\sim1.05$) and Reynolds number ($R_e=4.6\times10^3\sim3.5\times10^4$). In these figures, the loss coefficient ζ decreases rapidly when ξ increases from 0.125 to

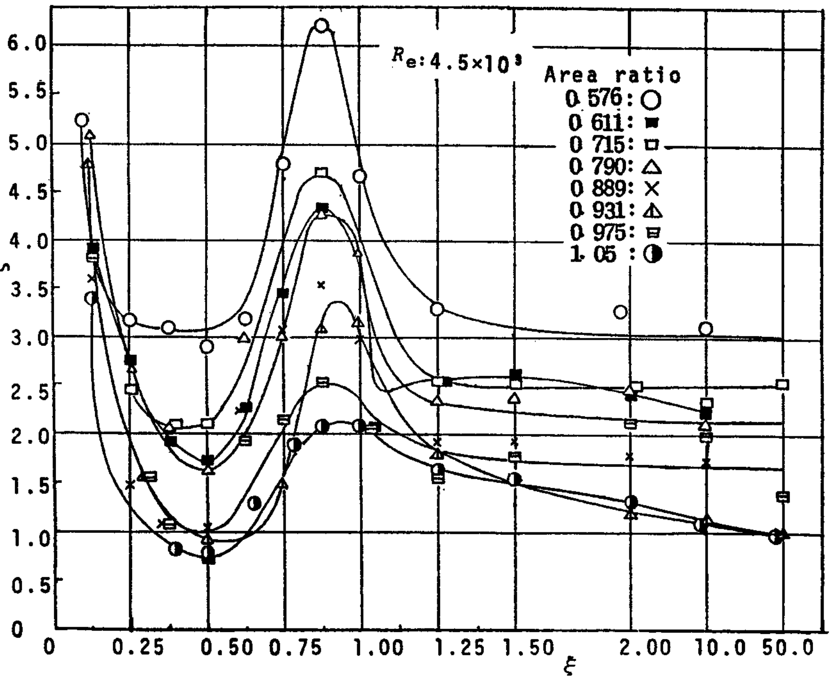

Fig. 5 Relation between loss coefficient ζ and non-dimensional clearance length ξ.

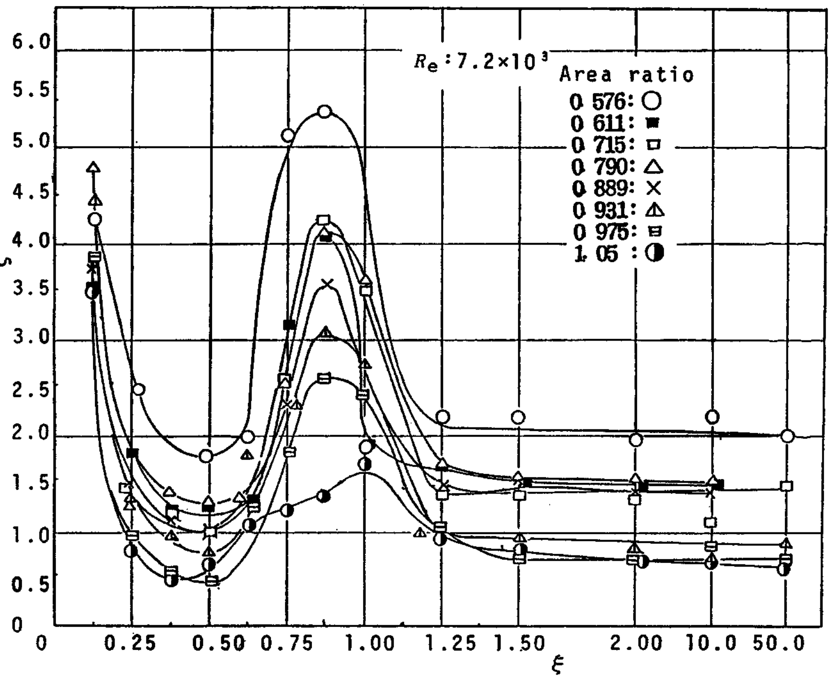

Fig. 6 Relation between loss coefficient ζ and non-dimensional clearance length ξ.

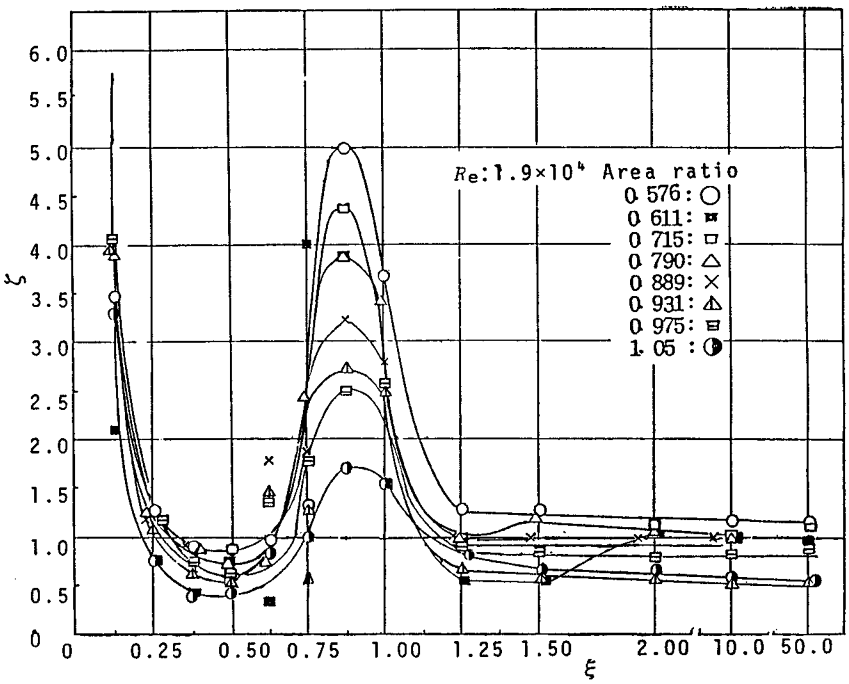

Fig. 7 Relation between loss coefficient ζ and non-dimensional clearance length ξ.

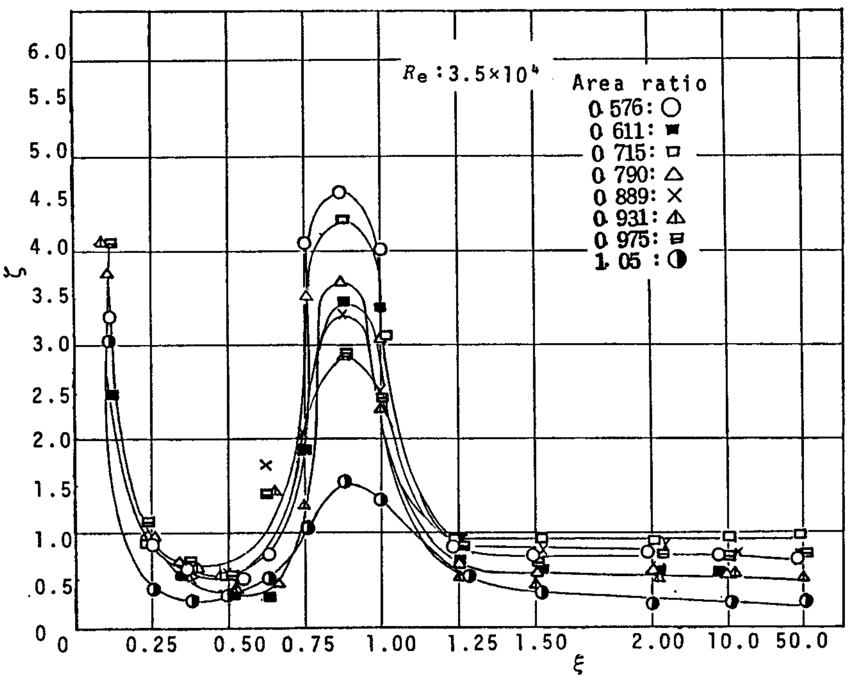

Fig. 8 Relation between loss coefficient ζ and non-dimensional clearance length ξ.

0.50 and takes a minimum value at about $\xi=0.50$ irrespective of the values of m and R_e. Then, ζ increases rapidly when ξ increases from 0.50 to 0.875 and takes a maximum value at about $\xi=0.875$ for any set of the values of m and R_e.

ζ depends on both ξ and m up to $\xi=1.25$ but does not so much on ξ in the range of $\xi>1.25$. It seems to reach a respective constant asymptotically as $\xi\rightarrow\infty$. ζ has an increasing tendency with decrease of m, which is clearly shown at the maximum point $\xi=0.875$, except the case of $m=0.611$. The two values of ζ at $\xi=0.75$ for $m=0.931$, which are shown in Fig. 7, imply a typical example of the fact that there is a certain kind of instability in the inner pipe flow.

4.2 Flow with pressure fluctuation

The pressure fluctuation was observed only in the inner pipe flow but little in the annular flow. Figs. 9–12 show the typical examples of pressure signal on an oscilloscope through an amplifier and a pressure transducer from the pressure tap, which is located on 1800 mm apart from the outlet of the inner pipe. Each smaller scale on the ordinate in the Figures expresses 12.7 P_a/div. and that on the abscissa 2.0 ms/div. The origin of the pressure head in these Figures corresponds to about 333.4 P_a in gauge pressure. Fig. 9 shows the pressure signal with very small pressure fluctuation for $\xi=0.5$, and Fig. 10 the signal with fairly small pressure fluctuation for $\xi=0.875$, where the loss coefficient ζ takes the maximum value.

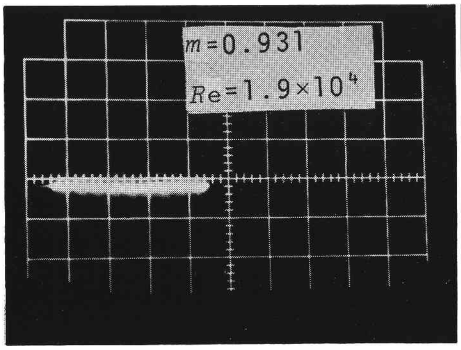

Fig. 9 Pressure signal for $\xi=0.5$.

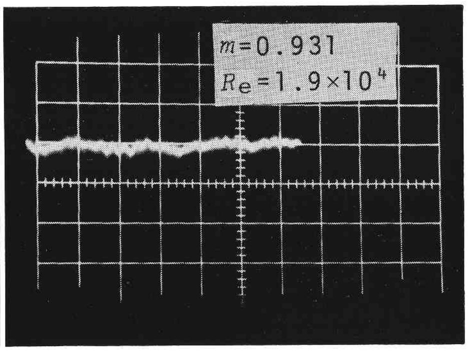

Fig. 10 Pressure signal for $\xi=0.875$.

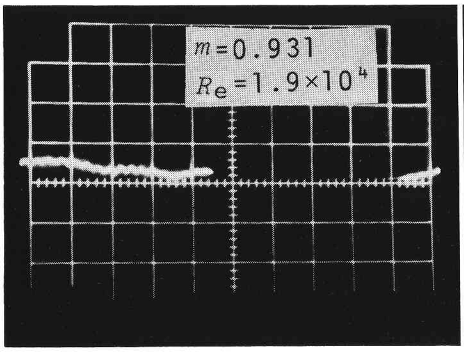

Fig. 11 Pressure signal for $\xi=0.75$.

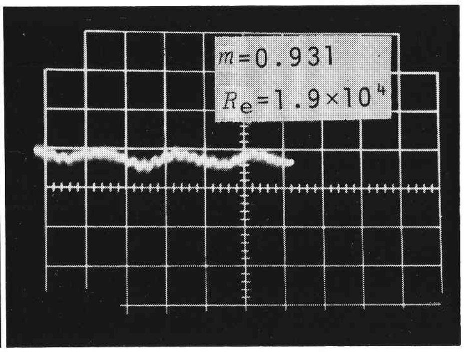

Fig. 12 Pressure signal for $\xi=1.0$.

Figs. 11 and 12 show the signal with an extremely large pressure fluctuation for $\xi=0.75$ and $\xi=1.0$, respectively. Fig. 11 expresses the typical example of the two values of loss coefficient ζ for $\xi=0.75$ and $m=0.931$, as mentioned in the preceding subsection. The larger value of ζ is derived from the maximum pressure signal in this Figure, while the smaller one from the minimum pressure signal.

5. Conclusion

The following conclusions can be derived from present experimental results.
1) The effect of the non-dimensional clearance length ξ upon the total loss coefficient ζ is clarified for various values of area ratio m ($0.576 \sim 1.05$) and Reynolds number R_e ($4.2\times10^3 \sim 3.5\times10^4$).
2) The minimum and maximum values of loss coefficient ζ occur at about $\xi=0.50$ and 0.875, respectively, irrespective of the value of m and R_e.
3) There is a certain kind of instability of fluid flow in the inner pipe over a short range of $0.75\leqq\xi\leqq1.0$.

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2) T. Asanuma and T. Kurahashi, JSME, Vol. **18**-69 (1952), 28.