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Analysis of Capsule Pipeline System by the Method of Characteristics

(Numerical Analysis and Comparison with Transport Experiment)*

By Yuji TOMITA**, Kimihiro ABE*** and Tomosada JOTAKI****

Based on the calculation of our previous paper, the transport characteristics of capsule in pipelines are examined by varying the pipe inlet pressure, pipe length and pipe diameter, and further it is shown that by the use of an exhaust valve placed at the middle of the pipeline an efficient transport of capsule can be achieved. To confirm the numerical analysis some experimental works are carried out and it is found that the numerical analysis is valid. The measurements according to Cudlin and Harman and those obtained with practical capsule pipelines are qualitatively predicted by the present calculation as well.

1. Introduction

In a previous paper⁽¹⁾, we numerically analyzed an air capsule pipeline system to examine the main transport characteristics of the capsule and obtained various interesting results. Many works have been done so far on the air capsule transport from different points of view, but the experiments on the transport characteristics in a conveying pipe are relatively few except that of Cudlin et al.⁽²⁾ In this paper, after describing some results of numerical analysis following the previous paper, we report on our experiment on capsule transport and discuss our numerical results with the measurements in details. Furthermore, we compare our computation with the measurements by Cudlin et al. and those obtained with practical lines.

2. Basic equations

The basic equations used in the numerical analysis are briefly described. The computation is based on the following assumptions:⁽³⁾

- (1) The air is a perfect gas and the static temperature is constant.
- (2) The capsule is regarded as a particle with the mass of the body and the friction coefficient of the pipe is constant.
- (3) There is neither a gap between the capsule and the pipe wall nor an air leakage through the capsule.

Then, the equations of continuity and momentum for the air in one-dimensional approximation and the equation of motion for the capsule are

$$\frac{\partial P^*}{\partial t^*} + u^* \frac{\partial P^*}{\partial x^*} + P^* \frac{\partial u^*}{\partial x^*} = 0 \quad (1)$$

$$\frac{\partial P^*}{\partial x^*} + P^* \frac{\partial u^*}{\partial t^*} + P^* u^* \frac{\partial u^*}{\partial x^*} = \varphi \quad (2)$$

$$W^* \frac{d^2 Z^*}{dt^{*2}} = P_R^* - P_F^* - F^* \quad (3)$$

where the asterisk denotes a dimensionless quantity and the symbols are defined as follows;

$$F^* = fW/P_0A$$

$$P^* = P/P_0$$

$$t^* = t\sqrt{gRT}/L$$

$$u^* = u/\sqrt{gRT}$$

$$W^* = WRT/P_0AL$$

$$x^* = x/L$$

$$Z^* = Z/L$$

$$\varphi = -(\lambda L/2d)P^*u^{*2}$$

and

A : pipe cross sectional area

d : pipe inside diameter

F : pipe friction for the capsule

f : kinematic friction coefficient between the capsule and the pipe wall

g : gravity acceleration

L : pipeline length

P : pressure

P_0 : atmospheric pressure

R : gas constant

T : absolute temperature of air in a pipe

t : time

u : air velocity

W : capsule weight

x : axial coordinates

Z : capsule coordinates

λ : pipe friction coefficient for air and the subscripts are;

F : front side of the capsule

R : rear side of the capsule.

From assumption (3) the following equation is obtained,

$$\dot{Z}^* = u_F^* = u_R^* \quad (4)$$

When air flows into the pipe, it causes a pressure loss. Assuming the pipe inlet as Borda mouthpiece, the inlet pressure P_1^* is given by

$$P_1^* = P_i / [1 + (u_i^{*m-1})^2] \quad (5)$$

where P_i is the pressure outside of the pipe and $( )^m$ is a dimensionless quantity $( )$ at the k 'th grid point of x^* and at the m 'th grid point of t^* .

Pressure losses such as at the pipe bend or at the capsule loading section, which are not included in the pipe friction, can be treated as if the pressure discontinuously

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** Associate Professor, Kyushu Institute of Technology, Tobata, Kitakyushu.

*** Engineer, Mitsubishi Chemical Industry, Yahatanishi, Kitakyushu.

**** Professor, Kyushu Kyoritsu University, Yahatanishi, Kitakyushu.

changes at the place by assuming such a section as a point.

In the following, we call the numerical analysis which neglects the third term of the left-hand side of Eqn.(2) Analysis I and that which does not neglect it Analysis II.

Fig. 1 Capsule mean velocity and final velocity vs. pipe length.

Fig. 2 Capsule mean velocity and final velocity vs. inlet pressure.

Fig. 3 Capsule mean velocity and final velocity vs. pipe diameter.

3. Mean velocity and final velocity of the capsule

As described in the previous paper, the present systems are dynamically similar if P_i^* , W^* , f^* and L/d are unchanged. However, it is comprehensible to present some illustrative results which are obtained by assigning concrete values to W , f and T to investigate the effects of pipe length, pipe diameter and inlet pressure on the mean velocity $\dot{Z}_{mean}$ and the final velocity $\dot{Z}_a$ of the capsule.

Figure 1 shows the effect of L when $P_i^* = 1.5$ and $d = 0.2 m$. Under this condition $\dot{Z}_{mean}$ takes a maximum near $L = 200 m$ and then decreases with an increasing L . In the figure, Δx^* denotes the grid length in the computation.

Figure 2 shows the effect of P_i when $L = 500 m$ and $d = 0.2 m$. $\dot{Z}_{mean}$ increases with about 0.5 power of $P_i^* - 1$. The limit of transport in the figure indicates a situation which is obtained by assuming $d^2 Z^* / dt^{*2} = 0$ in Eqn.(3).

Figure 3 shows the effect of d when $L = 500 m$ and $P_i^* = 1.5$. The reason why $\dot{Z}_{mean}$ increases with an increasing d is that the weight per unit cross section of capsule reduces with an increasing d for a given W .

4. Effect of exhaust valve

In a practical line of great length, an exhaust valve or a booster is often installed in the line to increase the capsule velocity. In this computation, we place an exhaust valve to investigate its effect on the capsule motion. Following the practical use, we assume that the exhaust valve is open to the atmosphere before the capsule reaches this point and is shut after its passing. Figure 4 shows the result of computation as well as the numerical conditions, where the solid line is the case of using an exhaust valve and the dashed line is the case of using none. Both cases follow a similar path until an expansion wave from the exhaust valve returns in the former. On arrival of the wave, $\dot{Z}$ of the former case increases rapidly compared to the latter case. However, since the stagnant air between the exhaust valve and the pipe exit is moved when the valve is shut, $\dot{Z}$ decreases rapidly, and varies almost equally to that of the latter case after its passing through the valve. In the former $\dot{Z}_{mean}$ is 46.8 m/s while in the latter case it is 42 m/s, and moreover the final velocity in the former is less than that in the latter, which helps stop the capsule.

5. Comparison with experimental results

5.1 Capsule transport in pressure system

5.1.1 Experimental apparatus and procedure

Figure 5 shows a schematic diagram of the apparatus. The pipeline consists of a straight circular pipe of 33.1 m length having 0.0416 m I.D. and placed horizontally. Pressure transducers are set at points A to F as shown in the figure to measure the pressure. By the use of air reservoirs, a reducing valve

Fig. 4 Effect of exhaust valve on the capsule motion.

Fig. 5 Schematic diagram of experimental apparatus.

Fig. 7 Time history of pressure at each measuring point (Experimental).

Fig. 8 Time history of pressure at each measuring point (Computation).

Fig. 6 Capsule used.

and a blow tank, the inlet air pressure is kept constant throughout the capsule transport. Figure 6 shows the capsule used. In order to avoid the air leakage through the gap between the capsule and the pipe wall, the front and rear sides of the capsule are wrapped with felt of 2mm thickness in natural condition. The weight of capsule is varied by using different capsule materials and by filling it with iron piece. The friction coefficient of each capsule are measured with a spring balance. In Table 1, which appears later, the values of W and f are given.

The capsule was placed at the starting point at first, and the sluce valve was shut. After compressed air was supplied by adjusting the reducing valve to a fixed pressure, the sluce valve was instantaneously opened. The pressure at each point was recorded by an oscillograph. The sluce valve could be fully opened instantaneously by lowering the lever by hand.

5.1.2 Experimental results and discussion

Figure 7 shows the time histories of pressure at each measuring point for $W=0.110$ kgf and $p_i=1.0$ kgf/cm², where p_i is the gauge pressure. Figure 8 shows the computed result under the same conditions with the experimental ones in Fig. 7, where the solid line indicates Analysis II and the dashed line indicates Analysis I. When the compression wave generated at the capsule front arrives at the measuring point, the pressure increases from zero and, during the capsule passing at the point, it discontinuously rises. After the capsule leaves the point, the pressure is nearly constant or slowly decreases. When the capsule goes out of the pipeline, the expansion wave reflected from the pipe exit returns to the point and the pressure rapidly

Fig. 9 Capsule velocity and pressure at the capsule front and rear.

Fig. 10 Pathline of the capsule and pressure waves.

Table 1 Comparison of experimental capsule travelling time, mean velocity and pressure difference across the capsule with numerical results.

Capsule	p_i (kgf/cm ²)	Experiment				Analysis II		
		t_a (s)	Z_{mean} (m/s)	Reversal of $p_R - p_F$		t_a (s)	Z_{mean} (m/s)	Reversal of $p_R - p_F$
No. 1 $W = 0.067 \text{ kgf}$ $f = 0.53$	1.0	0.482	65.4	x*		0.377	83.6	o**
	0.8	0.555	56.8	x		0.419	75.3	o
	0.6	0.642	49.1	x		0.483	65.3	o
	0.4	0.738	42.7	x		0.595	53.0	o
No. 2 $W = 0.110 \text{ kgf}$ $f = 0.46$	1.0	0.500	63.1	x		0.402	78.4	o
	0.8	0.565	55.8	x		0.448	70.4	x
	0.6	0.660	47.8	x		0.516	61.1	x
	0.4	0.780	40.4	x		0.634	49.7	x
No. 3 $W = 0.234 \text{ kgf}$ $f = 0.61$	1.0	0.580	54.4	x		0.469	67.2	x
	0.8	0.625	50.5	x		0.524	60.2	x
	0.6	0.725	43.5	x		0.605	53.1	x
	0.4	0.890	35.4	x		0.745	42.3	x

* Not observed: ** Observed.

Fig. 11 Comparison of experimental mean velocity with computation.

decreases. From these figures we can obtain the arrival time of the capsule at each point, the pressure difference across the capsule and the arrival times of compression and expansion waves. Small fluctuation of the pressure at point A in the initial period seems to be due to the small oscillation of pipe at the rapid opening of sluice valve. The computed result, as a whole, corresponds to the measurement qualitatively, but a reversal of the pressure difference across the capsule which occurs in the computation at point D is not observed in the experiment. Furthermore, the arrival of capsule is earlier in the computation. Figure 9 shows the pressure at the capsule front and rear and the capsule velocity vs. the capsule displacement obtained from the measurement and computation. The velocity from measurement is the average velocity in each interval. The agreement with the computation is very

good, but a reversal of pressure difference across the capsule is not observed in the experiment. It is noted from Fig. 9 that, in Analysis I, the frontal pressure oscillates near the pipe exit. Figure 10 shows the pathline of capsule and the arrival of both first compression wave generated at the capsule front and expansion wave generated by the capsule discharge out of the pipeline. It is

difficult to determine the time the capsule takes to travel the pipeline, that is, the capsule travelling time, from Fig. 7. This time is the ordinate of the cross point of the capsule pathline and the arrival of expansion wave at the pipe exit in Fig. 10. The mean velocity $\bar{z}_{\text{mean}}$ over the pipeline is determined by using this value in the experiment. Figure 11 shows $\bar{z}_{\text{mean}}$ vs. p_i . The experimental values are lower than the computed ones, although both are qualitatively similar. It can be seen that $\bar{z}_{\text{mean}}$ increases with $p_i^{0.5}$, which is the same as the result in Fig. 2, and that, for a given p_i , $\bar{z}_{\text{mean}}$ is smaller for larger W . The difference from the computed result is larger for smaller W . Table 1 shows the main results of experiment and computation. In the experiment a reversal of the pressure difference across the capsule is not observed nor it is seen in the computation when W is larger.

The discrepancy between the experimental and numerical results, we think, is caused as follows;

- (1) The short travelling time of the capsule of about 10^{-1} s due to the short pipe length has a large effect of static friction at the starting time, which is not considered in the computation, resulting in lower acceleration of the capsule and smaller $\bar{z}_{\text{mean}}$.
- (2) The lever lowering time of sluice valve is not small enough compared with the capsule travelling time.
- (3) There may be air leakage through the gap between the capsule and the pipe wall.
- (4) The assumption of isothermal flow is not adequate.

5.2 A comparison with the measurement by Cudlin et al.

Cudlin et al. carried out an experiment in vacuum system using a horizontal pipeline of 0.305 m I.D. and of 541 m length.⁽²⁾ From the figure in their paper the length from the pipe inlet to the discharge point is considered to be 513 m and the remaining 28 m to be the length for stoppage. Then, we assume in the computation that the pipe length is 513 m. The paper tells that the capsule has solid tires and that the gap between the capsule and the pipe wall is sealed with gaskets. The pipe inlet is open to the atmosphere and the compressor has a capacity of $2.36 \text{ m}^3/\text{s}$. The capsule is set at 161 m from the pipe inlet or at the pipe inlet and is released after the pressure in the pipeline is lowered. They measured the capsule velocity and the pressure at the capsule front and rear.

It is possible to apply the present method of computation to this vacuum system only by changing both of boundary and initial conditions. We give the pressure at the discharge point by approximating the measurement. The initial condition for air flow rate at each point is given, for convenience sake, in proportion to the distance from the starting point of capsule by assuming the discharge as $2.36 \text{ m}^3/\text{s}$.

Fig. 12 Capsule velocity and pressure at the capsule front and rear.

Fig. 13 Effect of capsule weight and friction coefficient.

Figure 12 shows a comparison with the computed result for the case when the initial coordinate of the capsule is 161m from the pipe inlet, where the reversal of the pressure difference across the capsule is significant. We assume $W=50$ kgf and $f=0.5$ for this case. Under these conditions, a fairly good agreement is obtained. The pressure at capsule front oscillates near the pipe exit in the measurement as well as in computation. It may be considered that, when the capsule comes near the exit, the pressure at the discharge point rapidly changes. Figure 13 shows Z for various values of W and f , but little effect is seen in this range.

5.3 Comparison with measurements in practical lines

5.3.1 Flow of air alone The measured values shown here are obtained in the pressure system of a factory. Figure 14 shows the pipeline configuration from which air flow data were obtained. The I.D. of line is 0.0693m. There are bends of 3m. radius at various points. The transport pressure is adjusted by the use of bourdon pressure gauge at the place before the capsule loading section and it takes a few seconds to open the valve fully. The experimental conditions were not sufficient to measure with enough accuracy and to compare the results with the computation because of field measurement. The pressure was measured by a mercury manometer at various points of pipeline and the air flow rate Q by the calibrated quadrant flow nozzle before the capsule loading section.

Figure 15 shows the time histories of the air flow rate Q , and the pressure farthest upstream between point B and point C. The solid line in the figure represents the result of Analysis I, where the loss at bends and at capsule loading section and that of pipe friction are taken into account but the effects of joint pipe, inlet loss and the exhaust valve near the pipe exit are neglected. The computation for p and Q fairly well trace the measurements. The discrepancy is likely due to slow response of measuring instruments.

Figure 16 shows the steady state air velocity near the pipe inlet u_s and the air flow rate Q_s in the case when the inlet pressure p_i is varied for the same flow as in Fig. 15. Since Q_s is in proportion to the

Fig. 14 Pipeline configuration for air flow alone.

Fig. 15 Time histories of pressure and volumetric air flow rate.

Fig. 16 Steady state air velocity and volumetric air flow rate vs. inlet pressure.

Fig. 17 Time histories of pressure at measuring points.

Fig. 18 Pipeline configuration for capsule transport.

Fig. 19 Capsule used.

Fig. 20 Capsule mean velocity vs. inlet pressure.

air velocity at the pipe exit, it is found from this figure that the air velocities at the pipe inlet and the exit are predictable from p_i . This result was already obtained in the previous paper and Fig. 16 gives an experimental evidence that the inlet air velocity tends to saturate for larger value of p_i . Figure 17 shows the time histories of pressure at various points of pipeline for different p_i when the air flows from point A to point C. A similar result was also obtained in the previous paper.

5.3.2 Capsule transport Figure 18 shows the pipeline configuration from which the capsule transport data were obtained. The pipe diameter is 0.0693 m which is equal to that of the line in Fig. 14. The pipeline length is 246 m. There were about 20 bends of 3 m radius and a joint pipe in the

Table 2 Comparison of experimental capsule travelling time with numerical results.

W (kgf)	P_i (kgf/cm ²)	t_a (s) *		
		Experiment	Analysis I	Analysis II
1.29	0.6	9.2	6.30	6.31
	1.0	6.3	4.79	4.81
	1.5	5.1	3.91	3.93
1.84	0.6	9.6	6.43	6.44
	1.0	6.3	4.90	4.90

*Travelling time

line. Furthermore, there was an exhaust valve near the pipe exit which is installed for deceleration of the capsule. The capsule loading section was vertically 12 m high relative to the pipe end.

Figure 19 shows the capsule used in the measurement which was made of steel and the front and rear sides were wrapped with cloth-containing gum. It was capable of charging 0.55 kgf load. To be able to pass the bend, the O.D. of the capsule was made smaller by 3.8 mm than I.D. of the pipe. Then, the air leakage through the gap would not be small. The friction coefficient with the pipe wall was about 0.4.

In the computation, the pipeline is assumed as horizontal, and the inlet loss and pipe friction are considered. We neglect the air leakage through the gap.

Table 2 compares the main results of measurement and computation. The influence of capsule weight is not observed in both analyses. Figure 20 shows Z_{mean} vs. p_i for $W=1.29$ kgf. Also in this case, Z_{mean} is in proportion to $p_i^{0.5}$. The discrepancy of the computed values with the measurements may result from neglect of the air leakage through the gap.

6. Conclusions

The present study has an emphasis on the experimental confirmation of the numerical analysis of the air capsule pipeline system by the method of characteristics according to the previous paper. We can say that the various characteristics obtained by the numerical analysis are supported qualitatively by the experiment. In the case of practical line containing many bends, it will be essential to take into account the air leakage through the gap between the capsule and the pipe wall. On the other hand in the case of short pipeline, the friction characteristics between the capsule and the pipe wall is important. The assumption of isothermal flow will also be examined in the future.

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