The Effect of Pumping in a Vertically Vibrating Pipe*

(Experimental Study on Flow Mechanism in a Pump)

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Research was carried out on the pumping effect of a vibrating pipe system. Measurements were made on the relative displacement of the vibrating pipe and the valve, and the pressure and flow rate in the pumping system. Relevant time historical data are compared and details on the hydrodynamic mechanism of the pumping effect are discussed in this report. Analysis on the fundamental pumping energy for investigating the energy conversion between the vibrating pipe and fluid was also carried out and these results are introduced as well.

Key Words: Fluid Vibration, Unsteady Flow, Pump, Pumping Effect, Energy, Vibrating Pipe

1. Introduction

Under certain vibratory conditions, it is possible to pump liquid out from a vertically vibrating cylindrical straight pipe equipped with a spring-controlled valve at its upper top edge. The authors have so far conducted theoreticcal analysis and experiments on such pumping effects and have previously reported the principle of liquidrise⁽¹⁾, the effect of the vibrating pipe length on the liquid rising characteristic^{(2)~(4)}, and pumping performance⁽⁵⁾.

These previous research results suggested that vibration conditions and the configuration of the suction pipe are important factors for achieving a favorable liquid rising performance.

It was also found that this liquid rising effect could be used effectively in small-scale pumps with low heads. A pump which utilizes this effect can be

used for pumping liquid with strong corrosive properties safely without any danger of leakage.

In designing such a pump, however, it is necessary not only to clarify the flow mechanism of the said pumping effect as actual machinery, but also to make further practical studies on factors which influence the pump head and efficiency. Many points have yet to be clarified as this type of pump involves a complicated flow mechanism during pump operation; moreover, the structural difference from conventional pumps must be worked out.

In this study, which is based on results of previous reports by the same authors, measurements were carried out on time historical data of displacement in several sections of the pump system to clarify the pumping mechanism in detail. The pressure and flow rate were also measured.

Details on the hydrodynamic vibration system were analyzed, based on experimental results, and the pumping effect in terms of hydrodynamics was clarified.

Furthermore, the basic factor which effects the pump performance was clarified by evaluating both the time dependent work rate of the vibrating pipe

* Received 21st December, 1990. Paper No. 89-1080A

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and the available work rate as the pump. The fundamental design data necessary for developing an actual pump were thus obtained.

2. Nomenclature

- A : Cross-sectional area of pipe $= \pi D^2/4$
 D : Internal diameter of pipe
 D_3 : Outer diameters of pipe and valve
 E_1, E_2 , and E_3 : Work rate of vibrating pipe
 E_4, E_5 , and E_6 : Work rate of pump
 f : Frequency
 g : Acceleration of gravity
 H : Pump head
 H : Total head
 P : Pressure
 Q : Flow rate
 Q_2 : Flow rate through valve clearance
 t_1 : Time valve opened
 t_2 : Time valve closed
 v : Mean velocity in cross-sectional area of pipe
 X_L : Displacement of liquid column in vibrating pipe
 X : Displacement
 X_P : Displacement of vibrating pipe
 X_V : Displacement of valve
 X_0 : Peak to peak displacement of vibration
 $\dot{X}$: dX/dt
 η : Pump efficiency
 ρ : Liquid density

Subscripts

- 1 : Static pipe
 2 : Vibrating pipe
 3 : Outlet of valve chamber
 t : Connecting part of pipe

Fig. 1 Schematic diagram of test apparatus

3. Experimental Apparatus and Method

The schematic diagram of the experimental apparatus is shown in Fig. 1. Details are as follows. The suction pipe ① is vibrated in a direction parallel to the pipe axis by the electrodynamic shaker ③; the amplitude of vibration is controlled by a frequency and amplitude controller to which the signal output from the accelerometer, attached to the vibrating pipe, is fed back; the stationary part of the suction pipe ② is fixed so that the bottom end of the pipe is submerged in the liquid while its top end is connected to the vibrating pipe; the dampener ⑤ is installed downstream from the pump discharge pipe ④ to enable as uniform a discharge pressure as possible; adjustment of the pump head is made by changing the height of the discharge tank ⑥; the submerged stationary suction pipe ② is devised to remain at almost the same depth, before and after the pumping operation, by the size of the tank ⑦ which is sufficiently large compared with the total inner volume of the pump and the pipe.

Figure 2 shows a detailed view of the pump. The circular valve ③, with the same outer radius as the vibrating pipe ①, is attached to the upper top end of the vibrating pipe by a coil spring ④. The connections between the vibrating pipe vs the valve chest and static suction pipe are sealed off by O-rings. Linear variable differential transformer-type displacement transducers are attached directly onto the vibrating pipe and the valve whose displacement was measured.

The suction flow rate was measured by an electromagnetic flowmeter installed at the bottom end of the static suction pipe. Calibration of the electromagnetic flowmeter was done according to a known vibration piston flow, whereby the gain and frequency characteristics of the phase were calibrated.

Fig. 2 Detailed view of test pump

The waveform deformation in flowmeter output signals was removed by a vessel type low-pass filter circuit (cut-off frequency : 100 Hz). These output signals were sampled at 0.1 ms time intervals using a recorder. They were then sent to a personal computer to be processed. Their gain and phase were corrected.

4. Experimental Results & Considerations

4.1 Pump performance

The objective of this research was to clarify the fundamental flow mechanism which causes the said pumping effect. This was done by analyzing the output waveforms of displacement, the pressure and the flow rate at certain sections of the pumping system. Since pump performance features had already been clarified and made available in a previous report ⁽⁵⁾, only the pump performance of a case where $X_0=2.0$ mm, 2.5 mm and 3.0 mm and whereby the reproducibility of output waveforms is favorable, is provided as an example (see Fig. 3). As can be gathered from Fig. 3, the flow rate Q increases along with an increase in the frequency and in X_0 under the same pump head H . The H decreased almost linearly along with the increase of Q within the range shown in the figure.

Selection of driving equipment becomes easy as relatively great pressure and flow rate can be obtained at the vibratory acceleration of $(2\pi f)^2 X_0/2$ whereby the acceleration due to gravity is merely 15g at most.

4.2 Fluctuation characteristics in displacement, pressure, and flow rate

Figure 4 shows an example of displacement, pressure, and flow rate waveforms measured within one vibration cycle at several sections of the pump system. These waveforms were measured at the operation point ($H=1.2$ m and $Q=2.2$ l/min) indicated by an arrow in Fig. 3. It can be gathered from Fig. 4 that the valve opens ($X_v > X_p$) at $t=t_1$, the time when P_2 (internal pressure in the vibrating pipe) reaches a maximum value, and that pressure pulsation occurs when the valve is opened. The initial pulsation cycle (indicated as 4.3 ms in Fig. 4) coincides with the time it takes the pressure waves (spreading in the liquid inside the pipe, whereby the elasticity of the suction pipe wall is considered) to travel back and forth along the length of the suction pipe a total of two times. That is, pressure waves due to the water hammer effect spread within the suction pipe when the valve is open.

The pressure, P_2 , temporarily undergoes a rapid decrease at $t=t_2$ after the valve closes ($X_v = X_p$). The pressure increases sharply, however, along with the increase in the flow rate Q_1 of the inflow from the bottom end of the stationary suction pipe, and the valve opens.

Fig. 3 Pump performance

Fig. 4 Displacement, pressure and flow rate in pumping system

It is assumed that the opening/closing of the valve, due to fluctuations in P_2 , are strongly affected by conditions of liquid flow just before the opening/closing of the valve. Considering the flow rate of each section in the proximity of the valve and the dynamics in the valve and vibrating pipe, in the case where there is corresponding movement between the valve and pipe, the following equation of continuity with regard to Q_1 , Q_2 , Q'_2 , and Q_3 (see Fig. 2) can be applied (liquid compression is disregarded in this equation) :

$$Q_1 = Q_2 + A_3 \dot{X}_p = Q'_2 + A_3 \dot{X}_v = Q_3 \quad (1)$$

wherein $Q_2 = A_2(\dot{X}_L - \dot{X}_P)$ and whereby the following equation can be obtained through Eq. (1):

$$Q'_2 = Q_2 - A_3(\dot{X}_V - \dot{X}_P) \quad (2)$$

Accordingly, it is evident from Eq. (2) that $Q'_2 = Q_2$ when $\dot{X}_V = \dot{X}_P$. Figure 4 also shows the results of the above equation whereby the values of Q'_2 and Q_2 were obtained using the substituted values of Q_1 , $\dot{X}_P$, and $\dot{X}_V$. It can be gathered from Fig. 4 that although $Q'_2 > 0$ when the valve is open, until it closes, Q_2 attains a negative value in the proximity of valve closure and a backflow occurs in the vibrating pipe. It is assumed that P_2 undergoes a rapid decrease influenced by the inertia of movement of the backflow occurring in the vibrating pipe and caused by the closing of the valve.

Figure 5 shows the fluctuation in P_2 , in a case where $f = 35 \text{ Hz}$ and $X_0 = 2.5 \text{ mm}$, while the valve closes following an increase in the pump head H .

It can be gathered from Fig. 5 that P_2 rapidly decreases along with the increase in H and that the minimal value of P_2 decreases as well. This is assumed to be due to an increase in both the valve closing velocity and the flow rate of the backflow, consequent to an increase in the H .

The minimal value of P_2 decreases along with an increase in the H , and fluctuation in P_2 versus time shows a flat relationship. The reason for this is assumed to be due to the rapid increase of pressure just before the opening of the valve. This pressure surge occurs when water columns in the vibrating pipe, which become separated due to sufficient reduction in P_2 , abruptly merge again.

Referring back to Fig. 4 and studying the outline

Fig. 5 Liquid pressure fluctuation in vibrating pipe for various pump heads

of the pump operation, it can be understood that the valve opens when the vibrating pipe makes a downward motion, whereby the fluid flows into the valve chamber. This fluid is pushed out of the valve chamber when the valve closes along with an upward motion of the vibrating pipe.

Comparing this pumping operation with that of a conventional piston pump (with one suction valve and one discharge valve), we obtain the following results.

Only one valve is necessary for the reciprocating movement which enables pumping by the vibrating pipe system, whereas two valves (suction and discharge) are necessary for that by the conventional piston pump. However, when the compressibility of the liquid is disregarded, the flow rate Q_P in the conventional 2-valve piston pump does not depend on H and becomes determined by only the piston stroke volume Q_P ($Q_P = 60fX_0A$, wherein A is the cross-section of the piston and whereby frequency f and amplitude X_0 are constant).

In case the of our pump, which uses the vibrating pipe system, flow rate Q increases with the decrease in H .

In example of a case where $f = 35 \text{ Hz}$ and $X_0 = 2.0 \text{ mm}$, Q_P of the piston pump was found to be approximately 3 liters/min while the actually measured Q of the present pump exceeded 4 liters/min (see Fig. 3), indicating a greater Q_P . This is because the inertia of movement of the liquid which enters the valve chamber, when the valve is open works to discharge the liquid through the discharge pipe, provided that the pressure buildup in the valve chamber is not great. However, increase in H results in the discharge pressure working as back pressure on the valve, and the time that the valve is open is shortened. It is assumed that this factor and the consequent shortening of the

Fig. 6 Relationship between valve-opening time and pump head

clearance between the valve and pipe cause a reduction in the discharge flow rate when the valve is open. Figure 6 shows the relationship between the H and the time that the valve is open. It is evident from this figure that the time the valve is open is reduced with the increase in H .

4.3 Work rate during pump operation

As discussed so far, the pumping effect of the vibrating pipe is different from that of the piston pump with 2 valves. Accordingly, the work rate per vibratory cycle was investigated to study the energy conversion and the pumping effect which follow the opening/closing of the valve.

The work rate per cycle of vibration was studied assuming that the liquid medium was not compressed and using the waveforms of pressure, flow rate, and variation of each pump section indicated in Fig. 4. This included the work rate (input) of the liquid done by the vibrating pipe and the work rate (output) of the pump.

(a) Work rate done by the vibrating pipe

While the valve is closed, the vibrating pipe works to increase the energy of the liquid column in the suction pipe. Assuming that an ideal liquid medium with no liquid loss is used and the total head is indicated as H (i. e., using the symbols and stationary control surfaces shown in Fig. 7), the liquid energy e_1 which passes through control surface A_1 during Δt s can be expressed as $e_1 = \rho g A_1 v_1 H_1 \Delta t$. The liquid energy e_2 that passes through control surface A_2 during Δt s can be expressed in the same way, i. e., $e_2 = \rho g A_2 v_2 H_2 \Delta t$.

Accordingly, as energy E_1 (accumulating in the liquid column inside the suction pipe while the valve is closed), is $A_1 = A_2$, the following equation can be

applied :

$$E_1 = \rho g A_1 \int_{t_2}^{t_1+T} (v_1 H_1 - v_2 H_2) dt \quad (3)$$

wherein T is the vibratory cycle.

The cross-sectional mean velocity of the flow v and the total head at control surfaces A_1 and A_2 can be expressed by the following equations :

$$\left. \begin{aligned} v_1 &= Q_1/A_1 \\ v_2 &= \{Q_1 - (A_2 - A_1)\dot{X}_p\}/A_1 \end{aligned} \right\} \quad (4)$$

$$\left. \begin{aligned} H_1 &= v_1^2/(2g) + P_1/(\rho g) \\ H_2 &= v_2^2/(2g) + P_2/(\rho g) \end{aligned} \right\} \quad (5)$$

The value of energy E_1 which accumulates in the liquid column can be obtained from Eqs. (3) ~ (5) above.

Furthermore, the vibrating pipe (i. e., the valve at the upper end of the pipe) provides a piston effect on the liquid in the valve chamber, located on the external side of the pipe, while the valve is closed.

This work rate E_2 can be expressed in the following equation (see Fig. 2 for symbols) :

$$E_2 = A_3 \int_{t_2}^{t_1+T} P_3 \dot{X}_p dt \quad (6)$$

As the thick section of the vibrating pipe wall provides the piston effect while the valve is open, assuming that there is no friction loss on the pipe wall and that the work rate is E_3 , the following equation can be applied :

$$E_3 = (A_3 - A_2) \int_{t_1}^{t_2} P_3 \dot{X}_p dt \quad (7)$$

(b) Work rate for effective pumping performance

Assuming that E_4 is the work rate for pumping performance by the piston effect of the vibrating pipe while the valve is closed, the following equation can be applied :

$$E_4 = \int_{t_2}^{t_1+T} P_3 Q_3 dt \quad (8)$$

Assuming that E_5 is the work rate that is used for the discharge of liquid from the vibrating pipe through the valve while the valve is open, the following equation can be applied :

$$E_5 = A_2 \int_{t_1}^{t_2} P_3 \dot{X}_L dt \quad (9)$$

Moreover, assuming that E_6 is the effective work rate which can be obtained in correspondence to the piston effect of the vibrating pipe wall while the valve is open and that Q_r is the flow rate resulting from the piston effect, the following equation can be applied :

$$E_6 = \int_{t_1}^{t_2} P_3 Q_r dt \quad (10)$$

whereby $Q_3 = A_2 \dot{X}_L + Q_r$, so the work rate which can be obtained for effective pumping performance while the valve is open can be expressed in the following equation :

Fig. 7 Control surface

$$E_5 = E_6 = \int_{t_1}^{t_2} P_3 Q_3 dt \quad (11)$$

Matching the energy accumulated in the liquid by the pipe vibration with the work rate for effective pump performance and putting the relevant data in order, E_2 and E_4 correspond to the piston effect while the valve is closed, while E_3 and E_6 correspond to the that while the valve is open. There was also correspondence between E_1 , the energy increase amount in the liquid column while the valve is closed, and work rate E_5 , obtained as the value of pressure and flow rate while the valve is open. This means that there is an energy recovery effect, in addition to the piston effect, by which the energy in the liquid column, accumulated inside the suction pipe while the valve is closed, is recovered as pressure and flow rate while the valve is open.

Figure 8 shows work rate values obtained in concrete terms through data sampled at 0.1 ms intervals and whereby Eqs. (3)~(11) are made discrete. The values of E_2 and E_4 , work rates resulting from the piston effect of the vibrating pipe while the valve was closed, showed an increase proportional to H and, as expected, matched well with each other. This indicates that the precision of the actual measurement on waveforms, whose results are shown in Fig. 3, is relatively high.

Figure 9 shows E_1 , the amount of energy increase in the liquid column while the valve was closed, and E_5 , the work rate which was effective while the valve was open. Note that the value of E_5 was obtained here through Eq. (11) and under the assumption that $E_6 = E_3$. In contrast to the values shown in Fig. 8, it is evident that the energy increase amount of the liquid column is significantly great when compared to the work rate due to the piston effect. It is also evident

Fig. 8 Work rate in piston effect

that the ratio of the energy, accumulated in the liquid column, to be converted to E_5 (the energy effective for pump performance), is relatively small.

However, it should be noted that these values of E include liquid loss as they are results of actual measurements.

Assuming that η_m is the mechanical efficiency which includes the efficiency of the vibratory mechanism, that η_v is the volumetric efficiency based on the leakage flow rate, and that η_h is the fluid efficiency, the overall pumping efficiency η can be obtained through the following equation:

$$\eta = \eta_m \eta_v \eta_h \quad (12)$$

However, the following Eq. (13) defines η_h as a value obtained from a comparison between the work rate for effective pump performance and the overall energy in the liquid obtained by the vibration of the

Fig. 9 Relationship between work rate increase of liquid column while the valve is closed and available work rate for pump performance while the valve is open

Fig. 10 Hydraulic pump efficiency

pipe :

$$\eta_h = E_4 + E_5 + E_6 / (E_1 + E_2 + E_3) \times 100 \quad (13)$$

Fig. 10 shows an example of the relationship between η_h obtained from Eq. (13) and H . The pump performance became higher with the increase in frequency and a maximum value of up to 50% or so was measured within the sphere of our research. As this value is almost identical to the pump efficiency of conventional pumps, the applicability is assumed to be sufficient.

The numerator of Eq. (13) can be expressed as follows (considering $Q_3 = A_3 \dot{X}_P + Q_2$; provided that the vibrating pipe carries out sinusoidal vibration and P_3 is at a constant versus t) :

$$\begin{aligned} E_4 + E_5 + E_6 &= A_3 \int_{t_1}^{t_1+\tau} P_3 \dot{X}_P dt + \int_{t_1}^{t_1+\tau} P_3 Q_2 dt \\ &= \int_{t_1}^{t_2} P_3 Q_2 dt \end{aligned} \quad (14)$$

As H and Q are inversely proportioned in approximated terms, as indicated in Fig. 3, η_h attains its maximum value either versus H or Q . Also, as is made evident in Fig. 8, the input/output work rate values of the piston effect are almost identical, suggesting that η_h depends strongly on E_1 and E_5 .

5. Conclusions

The mechanism of fluid dynamics in a pump which applies the pumping effect of a vibrating pipe was studied and clarified. Vital data for developing a practical pump which uses this pumping effect of the vibrating pipe was accumulated. The following summarizes the results of this research.

(1) Although the valve equipped on the vibrating pipe closes immediately after the pressure in the vibrating pipe drops to below that in the valve chamber, the pressure inside the vibrating pipe just after closing the valve rapidly decreases due to the occur-

rence of backflow which begins just before the closing of the valve. The pressure in the vibrating pipe, however, rapidly increases with the increase in the in-flow rate from the bottom end of the stationary pipe, causing the valve to open.

(2) The vibrating pipe works to push up the valve, and after the valve becomes closed, causes a piston effect on the fluid inside the valve chamber, and at the same time works to accumulate energy in the fluid column inside the suction pipe.

(3) The energy accumulating in the fluid column inside the suction pipe while the valve is closed is employed as effective energy for the fluid from the pump while the valve is open. The ratio is approximately 50% within the sphere of the research carried out.

(4) The pumping mechanism within one cycle was clarified in detail, i. e., in terms of fluid dynamics, and the basic design data necessary for improving the pumping performance of the said type of pump were accumulated from the digital signal analysis for measured waveforms.

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