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Performance of Air-Lift Pump*

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A basic equation of air-lift pump performance is derived from a simple momentum equation. In the analysis, frictional pressure drop in two-phase flow is expressed by means of liquid volume fraction of the mixture, and slug flow is assumed in order to obtain the relationship between liquid volume fraction and flow rates of both phases. The results predicted by the analysis are compared with experimental data of other investigators as well as of the authors. Good agreement with experimental data is shown in the ranges of 25 mm$\leqq D$ (inner diameter)$\leqq$100 mm, 4 m$\leqq L$ (length of pipe)$\leqq$42 m and $0.4<\sigma$ (submergence ratio)<0.8. To give a clear explanation for pump characteristics, the efficiency of air-lift pump is subdivided into two components, the pipe efficiency and the ideal discharge efficiency. A procedure of a pump design is proposed.

1. Introduction

As advantageous utilizations of an air-lift pump, it is well known to pump fluids which are corrosive for metals, explosive or toxic, volatile or evaporative (for example, higher temperature), and underground water. The special boiling air-lift pump which uses the change in phase from liquid to gas has been developed recently(1). Also in the future, it will play an important role in these areas.

Numerous investigations of an air-lift pump have been reported in the literature(3)~(8). However, there are little information on its performance, especially because of the lack of basic knowledge of two-phase flow in the past twenty years. Thus, it seems that any reliable method of the air-lift pump design has not been established. A publication in which the erroneous data are put is found even among the accepted ones(2). Within the knowledge of the authors, Stenning et al.(9) suggested the reasonable analytical treatment. Their approach was based on utilization of the slip ratio of gas to liquid. It seems, however, that the use of gas volume fraction or liquid holdup is comprehensive in analysis of an air-lift pump rather than slip ratio. Detailed and rational description of the procedure of pump design was not made in their papers, because the efficiency of pump was not discussed.

The study to be described here is an analytical investigation which makes use of the accepted and suitable correlations for liquid holdup and frictional pressure drop in two-phase flow. The aim of the research is to provide detailed basic information on both the characteristics of an air-lift pump and its design method. The experiments were performed with two model pumps, and made to serve for checking the accuracy of the analysis.

2. Nomenclature

c_1, c_2, c_3: coefficients
D: pipe diameter
F: cross-sectional area of pipe
F_f: frictional force
f_l: liquid holdup
$\bar{f}_l$: average liquid holdup
g: acceleration of gravity
G_l: weight flow rate of liquid
H_d: height of discharge
H_s: depth of submergence
L: length of upriser, H_d+H_s
L': length of suction pipe
Q_g: volume flow rate of air (at atmospheric pressure)
$\bar{Q}_g$: volume flow rate of air (at mean pressure in upriser)
$\bar{Q}_{ga}$: volume flow rate of air (just at origin of discharge)
Q_l: volume flow rate of discharged water
R_{el0}: superficial liquid Reynolds number, $W_{l0}D/\nu_l$
$\bar{w}_{g0}$: superficial air velocity in pipe, $\bar{Q}_g/F$
w_l: linear velocity of liquid
w_{l0}: superficial liquid velocity

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z: constant
ζ_e: coefficient of entrance loss in suction pipe
γ_l: specific weight of liquid
γ_{TP}: specific weight of two-phase mixture
η_1: pipe efficiency defined as Eq. (17)
η_2: ideal discharge efficiency defined as Eq. (18)
η_p: efficiency of discharge, $\eta_1\eta_2$
$\eta_{c,is}$: overall isothermal efficiency of an air compressor
σ: submergence ratio, H_s/L
λ_{l0}: friction factor based on liquid single phase flow
τ_{TP}: wall shear stress of two-phase flow

3. Derivation of equation

3·1 Basic equation

Let us consider an air-lift pump system shown in Fig. 1 which has a vertical pipe with constant cross-sectional area. The part $H_s+H_d=L$ is an upriser or discharge pipe in which the two-phase mixtures flow upwards, while the residual part L' is a simple suction pipe. When the air is injected into the upriser through the injector located at H_s under the liquid level, the liquid is then pumped up to the discharge height H_d at the rate of Q_l. Applying the momentum theory to the control surface under steadily operative condition, we obtain

$$\frac{G_l}{g}(w_{l1}-w_{l2})-F\int_0^L\gamma_{TP}dz-\int_0^L(\pi D\tau_{TP})dz-F_f-F\zeta_e\frac{w_{l1}^2}{2g}\gamma_l+F\gamma_lH_s=0 \quad\cdots(1)$$

in which only the momentum term of gas is neglected.

The first term of Eq. (1) represents the change in momentum of the liquid. This term becomes

$$\frac{G_l}{g}(w_{l1}-w_{l2})=\frac{\gamma_l}{g}Fw_{l0}^2\left(1-\frac{1}{f_{l2}}\right)\quad\cdots(2)$$

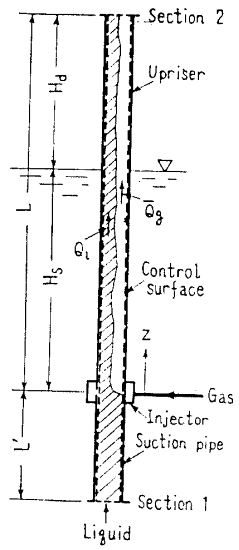


Fig. 1 Analytical model of air-lift pump

with the following equations of continuity

$$w_{l1}=w_{l0}\quad\cdots(3)$$

$$w_{l2}=\frac{w_{l0}}{f_{l2}}\quad\cdots(4)$$

$$G_l=Fw_{l0}\gamma_l\quad\cdots(5)$$

where F is the cross sectional area of pipe, f_{l2} the liquid holdup at the exit section, and w_{l0} the superficial liquid velocity assuming that liquid alone flows in the pipe.

The second term is the weight of the fluid in the upriser. In general, the specific weight of gas in an ordinary air-lift pump can be negligible comparing with that of liquid, so the specific weight of the mixture γ_{TP} is approximately equal to γ_lf_l. The second term then becomes

$$F\int_0^L\gamma_{TP}dz=FL\gamma_l\bar{f}_l\quad\cdots(6)$$

in which $\bar{f}_l$ is the average liquid holdup,

$$\bar{f}_l=\frac{1}{L}\int_0^Lf_ldz$$

The third term represents the frictional force exerting on the wall of the upriser. Up to date, some equations have been proposed to estimate the two-phase frictional pressure drop in a vertical pipe, but most of them are inconvenient for this case. However, Akagawa(10) and Katsuhara(11) suggest the following equation for it

$$\frac{\Delta P_{TP}}{\Delta P_l}=f_l^{-z}\quad\cdots(7)$$

which is simple in expression, and agrees fairly well with experimental data. In this equation, ΔP_{TP} is the frictional pressure drop of two-phase flow, ΔP_l that of single-phase liquid flow calculated on the assumption that the liquid is flowing alone at the same mass flow rate as in the two-phase case, and z a constant to be dependent on the angle of pipe inclination and the roughness of pipe wall. Applying Eq. (7) to the upriser of an air-lift pump and using $\bar{f}_l$ as the liquid holdup, the third term becomes

$$\int_0^L(\pi D\tau_{TP})dz=F\Delta P_{TP}=F\bar{f}_l^{-z}\lambda_{l0}\frac{L}{D}\frac{w_{l0}^2}{2g}\gamma_l\quad\cdots(8)$$

The fourth and fifth terms are the frictional force exerting on the wall of the suction pipe and the pressure force due to the entrance loss, respectively. They can be rewritten, therefore, as

$$F_f+F\zeta_e\frac{w_{l1}^2}{2g}\gamma_l=F\left(\lambda_{l0}\frac{L'}{D}+\zeta_e\right)\frac{w_{l0}^2}{2g}\gamma_l\quad\cdots(9)$$

Substituting Eqs. (2), (6), (8) and (9) in Eq. (1), after some arrangement the following equation is obtained

$$(H_s-L\bar{f}_l)\gamma_l=c_1\bar{f}_l^{-z}\frac{w_{l0}^2}{2g}\gamma_l+c_2\frac{w_{l0}^2}{2g}\gamma_l+c_3\frac{w_{l0}^2}{2g}\gamma_l\quad\cdots(10)$$

where c_1, c_2, c_3 are coefficients defined as

$$c_1 \equiv \lambda_{l0}\frac{L}{D}, \quad c_3 \equiv \lambda_{l0}\frac{L'}{D}+\zeta_e, \quad c_3 \equiv 2\left(\frac{1}{f_{l2}}-1\right)$$

The left and right hands of Eq. (10) represent the driving force for pumping action and the hydraulic losses in a pump, respectively. Rearranging Eq. (10) again, we obtain

$$\frac{w_{l0}^2}{2gL}=\frac{\sigma-\bar{f}_l}{c_1\bar{f}_l^{-z}+c_2+c_3} \quad \cdots\cdots(11)$$

Let us consider a suitable equation for estimating the average liquid holdup in the upriser. The following equation (12) seems to be recommendable from two points of view. First, the slug flow frequently occurs in an air-lift pump. Second, an explicit relationship between liquid holdup and the flow rates of both phases in the upriser is required for predicting the performance of the pump.

$$\bar{f}_l=1-\frac{\bar{w}_{g0}}{1.2(\bar{w}_{g0}+w_{l0})+0.35\sqrt{gD}} \quad \cdots\cdots(12)$$

This equation is derived from the equation of velocity of gas slug in a vertical pipe proposed by Nicklin et al.[12], which is a well-known simple form of $w_g=1.2(w_{g0}+w_{l0})+0.35\sqrt{gD}$. In Eq. (12), however, w_{g0} is replaced by $\bar{w}_{g0}$ for simplicity, the superficial gas velocity at the mean pressure section of the upriser.

3·2 Efficiency

If we assume that the overall efficiency of an air-lift pump system including an air compressor consists of the efficiency of discharge and that of a compressor

$$\eta_t=\eta_p\eta_c \quad \cdots\cdots(13)$$

The efficiency η_c can be considered as

$$\eta_c=\eta_{c,is}, \quad \cdots\cdots(14)$$

the overall isothermal efficiency of the compressor, defined as

$$\eta_{c,is}=\frac{\left[\begin{array}{l}\text{power requirement for compressing}\\ \text{gas in isothermal condition}\end{array}\right]}{[\text{brake horsepower of compressor}]} \quad \cdots(15)$$

The efficiency of discharge can be subdivided furthermore into two components

$$\eta_p=\eta_1\eta_2 \quad \cdots\cdots(16)$$

in which η_1 and η_2 are designated respectively as the pipe efficiency and the ideal discharge efficiency, and are defined as follows

$$\eta_1=\frac{[\text{actual liquid power gained}]}{\left[\begin{array}{l}\text{ideal liquid power gained, in the}\\ \text{case of neglecting relative velocity}\\ \text{of both phases and hydraulic losses}\\ \text{in a pump}\end{array}\right]} \quad \cdots(17)$$

and

$$\eta_2=\frac{\left[\begin{array}{l}\text{ideal liquid power gained, in the}\\ \text{case of neglecting relative velocity}\\ \text{of both phases and hydraulic losses}\\ \text{in a pump}\end{array}\right]}{\left[\begin{array}{l}\text{power requirement for compressing}\\ \text{gas in isothermal condition}\end{array}\right]} \quad \cdots(18)$$

It seems that the pipe efficiency η_1 presents the performance of an individual air-lift pump, and the ideal discharge efficiency η_2 does the inherent quality of the air-lift pump.

As described in the foregoing, the efficiency of discharge η_p depends only on an air-lift pump with exception of a compressor, while η_t is the over-all efficiency including also an air-compressor. It should, therefore, be recommended that η_p can be utilized in the case of comparing the performance of this kind of pumps themselves, and η_t, on the other hand, has to be employed in comparing it with the other types of pump.

The efficiencies of η_1 and η_2 can be formulated as below. Considering f_l' as the liquid holdup in the case of neglecting the relative velocity of both phases and hydraulic losses in a pump, we obtain from Eq. (10)

$$H_s-Lf_l'=0 \qquad \therefore\ f_l'=\frac{H_s}{L}=\sigma \quad \cdots\cdots(19)$$

If we assume Q_l' as the ideal discharge flow rate when no relative velocity exists in the upriser, another expression for f_l' is obtained

$$f_l'=\frac{Q_l'}{\bar{Q}_g+Q_l'} \quad \cdots\cdots(20)$$

Hence, combining Eqs. (19) and (20), we find the following ideal discharge flow rate

$$Q_l'=\frac{\sigma}{1-\sigma}\bar{Q}_g \quad \cdots\cdots(21)$$

From Eq. (11), the actual discharge rate Q_l is rewritten as

$$Q_l=F\sqrt{\frac{2gL(\sigma-\bar{f}_l)}{c_1\bar{f}_l^{-z}+c_2+c_3}} \quad \cdots\cdots(22)$$

Finally, the pipe efficiency η_1 defined as Eq. (17) can be formulated as Eq. (23) using Eqs. (21) and (22).

$$\eta_1=\frac{\gamma_l Q_l H_d}{\gamma_l Q_l' H_d}=\frac{1-\sigma}{\sigma}\frac{Q_l}{\bar{Q}_g}=\frac{1-\sigma}{\sigma}\frac{1}{\bar{w}_{g0}}\sqrt{\frac{2gL(\sigma-\bar{f}_l)}{c_1\bar{f}_l^{-z}+c_2+c_3}} \quad \cdots\cdots(23)$$

If we assume that the gas should be compressed from the atmospheric pressure p_a to $\gamma_l H_s$, the ideal discharge efficiency η_2 yields

$$\eta_2=\frac{\gamma_l Q_l' H_d}{p_a Q_g \ln\left(\dfrac{\gamma_l H_s+p_a}{p_a}\right)} \quad \cdots\cdots(24)$$

in which Q_g is the gas flow rate reduced to the atmospheric pressure. Substituting Eq. (21) into (24), we obtain

$$\eta_2=\frac{\sigma L}{\left(\dfrac{1}{2}\sigma L+H_a\right)\ln\left(\dfrac{\sigma L+H_a}{H_a}\right)} \quad \cdots\cdots(25)$$

It is seen from this equation that η_2 is the function of the submergence of a pump, $\sigma L=H_s$, alone.

Although the efficiency of discharge η_p is the ratio of the actual liquid power gained to the power requirement for compressing gas and has the same expression accepted as in the past, it can be definitely indicated in the present paper to consist of two components, equations (23) and (25).

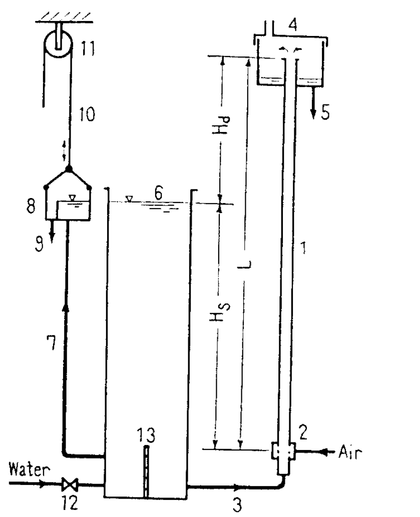
4. Experiments

4·1 Experimental apparatus

A schematic diagram of the experimental apparatus is shown in Fig. 2. When the metered stream of air was conducted to 2 and injected continuously into the upriser, the corresponding rate of water flowed into it through the suction pipe 3. Then, the two-phase mixtures flowed upwards in the upriser and were discharged continuously from its exit opening into the atmosphere. The drainage from the discharge tank 4 was measured as the volume flow rate of discharged water Q_l.

The source of water for the air-lift pump was the reservoir 6 with constant water level, for which the tap water was supplied corresponding to the discharge. In order to keep the water level at given position, i.e. the depth of submergence of a pump H_s, the following methods were employed: The over-flow tank 8 was connected to the reservoir through flexible hose 7, a small quantity of over-flow could always take place adjusting the valve 12. Since, in addition, the over-flow tank was hung up by a wire 10 using a standing pulley 11, the position of the tank, that is the submergence H_s, could be held at its given level.

Two acrylic resin pipes were employed as the upriser during the course of the experiments, 28.3 mm ID(7.5 m long) and 50.6 mm ID(6.8 m long). By means of the preliminary single-phase water flow experiments, inner surface of the former was known to be smooth and that of the latter to be slightly coarse with roughness $\varepsilon/D \cong 0.002$. In addition, the hydraulic losses in the suction pipe 3 were estimated also by preliminary tests.



1 : Upriser
2 : Air injector
3 : Suction pipe
4 : Separator
5 : Discharged water
6 : Reservoir
7 : Flexible hose
8 : Over-flow tank
9 : Drainage
10 : Wire
11 : Pulley
12 : Control valve for water supply
13 : Buffer plate

Fig. 2 Experimental apparatus

4·2 Experimental results

Results for the 28.3 mm ID pipe are shown as the relationship between the water rate discharged and the air rate supplied in Fig. 3. In the figure, $\bar{Q}_g$ represents the air flow rate at the mean pressure in the upriser. Each curve shown in Fig. 3 depicts the characteristics of an air-lift pump; For a fixed submergence ratio, no discharge takes place until a certain air rate is reached, but once the air rate exceeds the critical value, the discharge increases rapidly with an increasing air rate up to a maximum. After reaching the maximum, on the contrary, the discharge decreases slowly with a further increase of air rate. At a fixed air rate the discharge increases, as will be expected, with an increasing submergence ratio. It seems that the trends described above are common, independent of the scale of an air-lift pump.

5. Discussion

5·1 Comparison between analysis and experiment

Consideration may be given first to the pump installed for a well of medium depth. Comparisons between analysis and experiments are shown in Figs. 4(a) and (b). These figures show the relationship between dimensionless discharge rate, $w_{l0}^2/2gL$, and mean air volume fraction, $(1-\bar{f}_l)$. In calculating these analytical values from Eq. (11), z was assumed to be 1.75 for a smooth pipe on the basis of Katsuhara's recommendation[11]. As the coefficients c_1 and c_2 changed within narrower ranges in this experiment, they were assumed to be fixed values at $w_{l0}=0.6$ m/sec, the superficial water velocity at the maximum efficiency that will be described in the section 5·3. The coeffi-

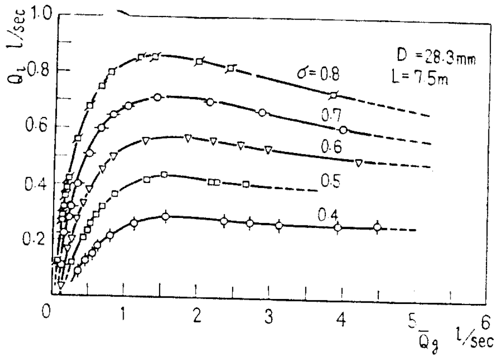
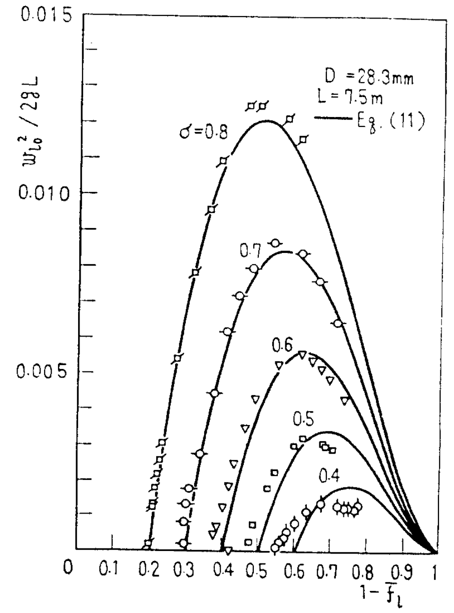


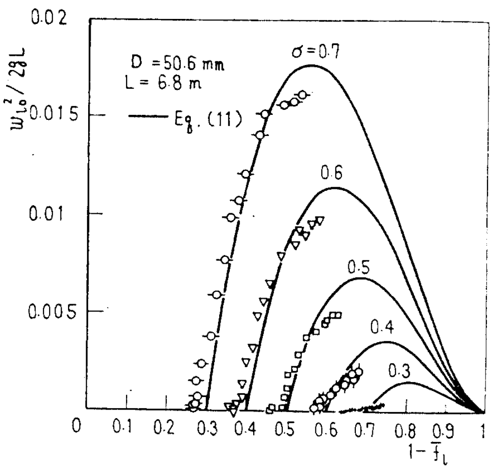
Fig. 3 Relationship between volume flow rate of water discharged and air rate supplied

cient c_8 was calculated using f_{l2} which was estimated by trial and error method. Further comparison between analytical value and experimental value obtained by Stenning et al.[9] from their model air-lift pump, $D=1$ in and $L=168$ in, is made in Fig. 5.

Now, attention will be directed to the pump installed for a deeper well. Figure 6 is comparison between analytical value and experimental one taken from Pickert's experiment[4] of a larger scale pump, $D=100$ mm and $L=42$ m. Although there are three more series of data for submergence ratio, $\sigma=0.646$, 0.574, and 0.268, in his original paper other than in this figure, they have been omitted because of complexity. In calculating the analytical value from Eq. (11), since the roughness of the pipe is not described in his paper, it was assumed to be the same order as that of a new cast iron pipe.



(a)



(b)

Fig. 4 Comparison between analysis and experiments

In Figs. 4, 5 and 6, the agreement between the data points and the analytical curve is remarkably good. But, it is also indicated in theses figures that the curves do not necessarily agree with the data for whole ranges of σ and $(1-\bar{f}_l)$. It seems that there are mainly two reasons for this less good agreement, especially in particular operative conditions such as smaller value of σ. One is that the slug flow has been postulated as flow pattern in the upriser regardless of flow condition. And the other is that the liquid holdup in each run has been assumed to be its average value at the mean pressure section of the upriser.

However, in an air-lift pump, the change in the

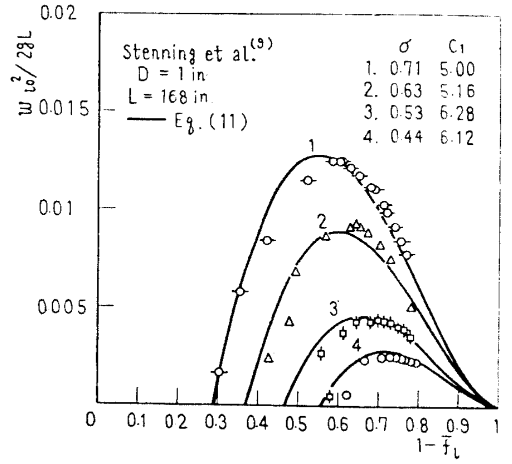


Fig. 5 Comparison between analysis and experiment[9]

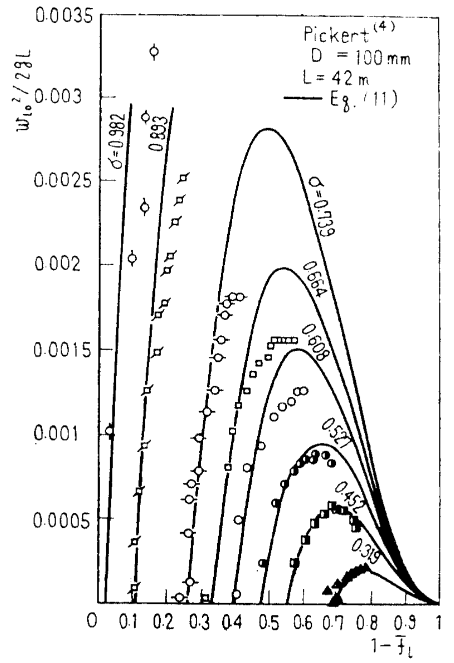


Fig. 6 Comparison between analysis and experiment[4]

submergence ratio is limited to a somewhat narrower range, i.e. $\sigma=0.5\sim0.8$, and the slug flow is the possible flow pattern in its ordinary operations. In addition, it seems that it is extremely difficult to analize Eq. (1) with regard to the change in frictional loss and liquid holdup along pipe axis owing to the expansion of air. Correspondingly, it is impossible to expect to obtain a simple relationship between the volume flow rate of discharged water and the air rate supplied. Equations (11) and (12) can, therefore, be recommended as the basic equations for an air-lift pump.

5·2 Characteristics of pumping action

A characteristic curve of water flow rate discharged versus air rate supplied can be illustrated as in Fig. 7. As explained in section 4·2, no discharge takes place until the air rate reaches to the value corresponding to point a, $\bar{Q}_{ga}$, because the water level in the upriser does not exceed the delivery height H_d. When air is supplied more than $\bar{Q}_{ga}$, the pump starts to discharge water. Water rate increases first with an increasing air rate up to a maximum c, and then falls away gradually from c to d.

In general, with an increasing air rate, the driving force of an air-lift pump increases owing to the reduction of the liquid holdup in the upriser, and, on the contrary, hydraulic losses in pump increase because of being higher fluid velocity. The rate of increase in the former predominates over that in the latter from a to c, and vice versa from c to d.

The efficiency of discharge η_p attains its maximum when the pipe efficiency η_l is just maximum. Thus, as easily known from Eq. (23), drawing the tangent line from an origin to the characteristic curve, the condition of the maximum efficiency is obtained as the point of contact b as shown in Fig. 7.

The range of $0<\bar{Q}_g<Q_{ga}$ in which no discharge occurs can be described as the following imaginary condition: If the whole upriser were filled with the mixture even in this case, the driving force would turn to the opposite direction from the ordinary operation because $\gamma_l H_s<\gamma_l \bar{f}_l L$ in Eq. (10). The dashed curve ae represents the above description.

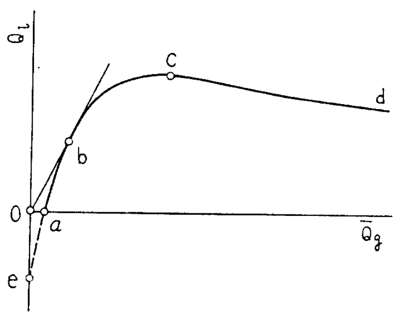


Fig. 7 Characteristic curve for air-lift pump

If we assume that the frictional pressure drop is proportional to $\bar{f}_l^{-z}$ even in the downward two-phase flow, the characteristic curve can be formulated as a whole.

$$Q_l=F\sqrt{\frac{2gL|\sigma-\bar{f}_l|}{c_1\bar{f}_l^{-z}+c_2+c_3}}\,\mathrm{sgn}(\sigma-\bar{f}_l) \quad\cdots\cdots(26)$$

Attention may next be directed to the lower critical air rate that causes the discharge. From Eq. (11), when the water level in the upriser attains just its exit opening, the following equation is obtained

$$\bar{f}_l=\sigma \quad\cdots\cdots(27)$$

According to the experiment of Nicklin et al.[12], the rising velocity of an air slug in a vertical pipe filled with still water is

$$w_g=1.2w_{g0}+0.35\sqrt{gD} \quad\cdots\cdots(28)$$

Thus, corresponding liquid holdup can be written as

$$\bar{f}_l=\frac{0.2\bar{w}_{g0}+0.35\sqrt{gD}}{1.2\bar{w}_{g0}+0.35\sqrt{gD}} \quad\cdots\cdots(29)$$

Substituting Eq. (29) into (27), the lower critical air rate $\bar{Q}_{ga}$ then becomes

$$\bar{Q}_{ga}=\frac{0.35(1-\sigma)F\sqrt{gD}}{1.2\sigma-0.2} \quad\cdots\cdots(30)$$

Therefore, it has been revealed that the air has to be supplied at least at a rate more than that of Eq. (30) in order to cause the discharge of water.

5·3 Maximum efficiency

In the design of an air-lift pump, it is necessary to know the information on the condition of its maximum efficiency. Figure 8 shows an example of how the maximum efficiency of discharge $\eta_{p\,max}$ is changed according to the submergence ratio, for a pump of $D=28.3$ mm and $L=7.5$ m. In the figure, the curve is a calculated value, and the open circles the experimental values. Although the agreement between them is unsatisfactory, the figure depicts the existence of the maximum $\eta_{p\,max}$ in the range of $\sigma=0.7\sim0.8$, and the pronounced tendency of its reduction in $\sigma<0.5$.

Figure 9 represents experimental results for the

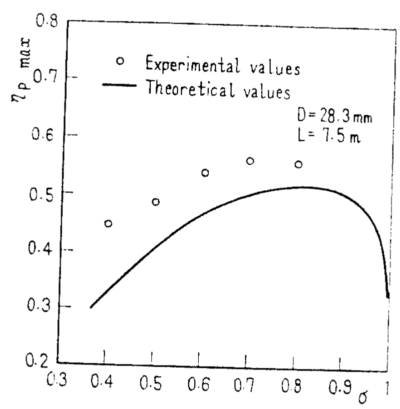


Fig. 8 Variation of maximum efficiency $\eta_{p\,max}$ with submergence ratio σ ($D=28.3$ mm, $L=7.5$ m)

superficial water velocity w_{l0b} at the maximum efficiency for each submergence ratio, including the data obtained by other investigators as well as by the authors. It appears from the figure that though the pumps from which these data were obtained are different in size and roughness from each other, w_{l0b} data fall almost within the range of 0.4~0.7 m/sec against the ordinary submergence ratio $\sigma=0.5\sim0.8$. Data for the superficial water velocity w_{l0c} at the maximum discharge condition are also plotted in the figure. Neglecting c_2 and c_3 comparing with $c_1\bar{f}_l^{-z}$ in Eq. (11), we obtain

$$\frac{w_{l0}^2}{2gL}=\frac{\sigma-\bar{f}_l}{c_1\bar{f}_l^{-z}}$$

Performing the differentiation of this equation and solving $d(w_{l0}^2/2gL)/d\bar{f}_l=0$, the following equation is obtained

$$\bar{f}_l=\frac{\sigma z}{z+1}$$

Substituting this into the above equation and recalling $c_1=\lambda_{l0}L/D$ the superficial velocity at the maximum discharge condition has the following trend,

$$w_{l0c}\propto(D\sigma^{z+1})^{1/2} \qquad (31)$$

which is similar to the tendency of the experimental data shown in Fig. 9.

6. Procedure of an air-lift pump design

Since an air-lift pump is, in general, designed to obtain the discharge Q_l required against a given discharge height H_d, a recommendable procedure of design for this case will be as follows.

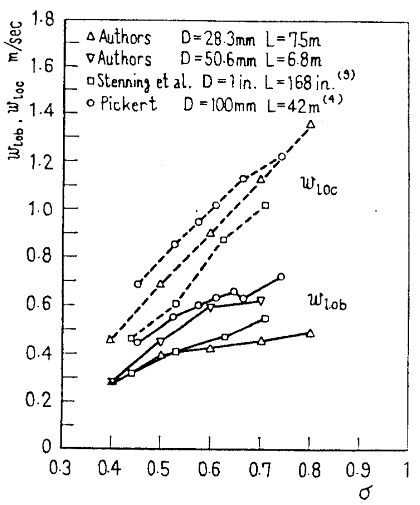
(1) Determine the total length of pump L: Judging from Fig. 8, there is an adequate submergence ratio in $\sigma>0.5$ in ordinary operation. If σ is too small, the efficiency is reduced. Thus, the air consumption is increased, and the running cost becomes larger. In contrast to this fact, when the plant is installed for pumping up the underground water, the excavation cost is increased with an increasing submergence ratio, that is, the depth of well. Therefore, the optimum ratio must exist.

(2) Determine the diameter of an upriser: The superficial water velocity corresponding to the optimum operation, the maximum efficiency, exists within $w_{l0b}=0.4\sim0.7$ m/sec for $\sigma=0.5\sim0.8$ from Fig. 9. The diameter of an upriser can thus be determined from this fact.

(3) Find out the relationship between the water rate discharged and the air rate supplied: The relationship between them at a given submergence ratio is obtained from Eqs. (11) and (12) by iterative calculations. Process of this phase is shown in Fig. (10) as the flow chart; Calculate first the superficial water velocity w_{l0} from Eq. (11) correspondingly to a given submergence ratio σ and liquid holdup $\bar{f}_l$. Next, assume $\bar{w}_{g0}$ and calculate the corresponding liquid holdup $\bar{f}_l$. Repeat this process until $\bar{f}_l=\bar{f}_l'$ is satisfied. As a result of the iteration, $\bar{w}_{g0}$ and thus Q_g can be found out.

(4) Prepare the performance curves: Once the air rate supplied is obtained, the efficiency η_p and the power consumption L_{is} are also found out. Consequently, the performance curves of the air-lift pump designed can be drawn as a similar expression to that for a centrifugal pump, such as the curves for air rate, power consumption and efficiency versus the rate of discharged water as shown in Fig. 11.

According to the above procedure, a design has been made for example, with $Q_l=1.2$ l/s, $H_d=6\sim10$ m, and $\varepsilon/D\cong0.005$ as for designing conditions. The resulting pump becomes $D=50$ mm and $L=20$ m in size, and its performance is estimated as Fig. 11. In



Note: There are no data points of W_{loc} in the figure for the case of D=50.6 mm, L=6.8 m, because sufficient air rate could not be supplied

Fig. 9 Results for the superficial water velocity at the point of maximum efficiency and of maximum discharge

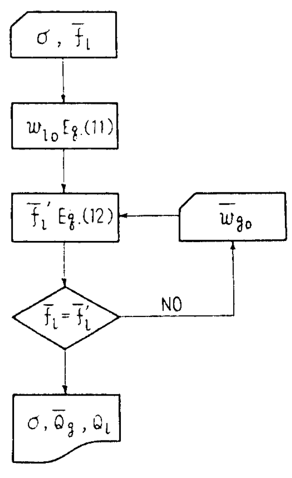


Fig. 10 Flow chart for evaluation of air rate supplied

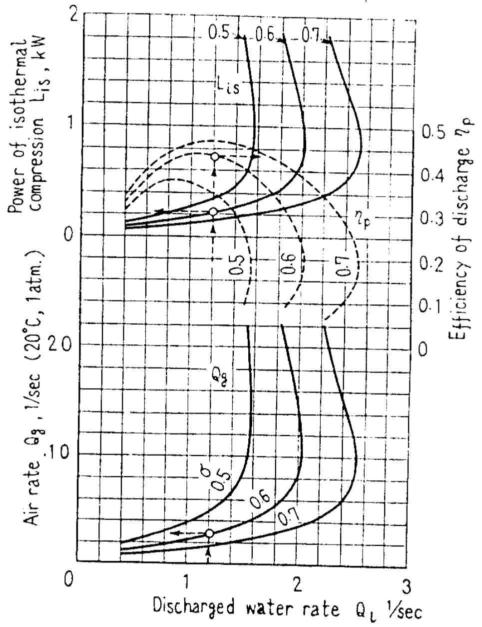


Fig. 11 Predicted performance curves for an air-lift pump designed
Design conditions:
$Q_l=1.2\,l/s$, $H_d=6$ 10 m, $\varepsilon/D=0.005$
Size of pump:
$D=50$ mm, $L=20$ m, without suction pipe

this figure, the air rate Q_g is calculated as the value at 20°C and 1 atm, and the power L_{is} is in kilowatts (kW).

7. Conclusions

The performance of an air-lift pump was investigated both theoretically and experimentally. The results obtained are summarized as follows.

(1) A basic equation of an air-lift pump performance is derived from the momentum equation.

(2) The results predicted by the analysis are compared with experimental data of other investigators as well as of the authors. Good agreement with experimental data is shown in the ranges of 25 mm$\leqq D\leqq$100 mm, 4 m$\leqq L\leqq$42 m, and $0.4<\sigma<0.8$.

(3) In order to get a clear comprehension for pump performance, the efficiency of discharge is divided into two components, the pipe efficiency and the ideal discharge efficiency.

(4) A procedure of pump design is proposed on the basis of the present theory and the data obtained by experiments.

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Discussion

M. Arie (Hokkaido University):

(1) Since it could apparently be judged that the two-phase flow mechanism in upriser would considerably depend on the type of an air-injector, the injector will have an influence not only on the efficiency of the air-lift pump itself, but also on the overall efficiency of air-lift pump because of the change in the load of the air compressor. However, there is no detailed description about it in the paper.

Please indicate the details of the air injector employed in this experiment, if you agree to the discusser's comment. If the authors have other special reasons for not describing its details, please indicate them.

(2) The height of discharge H_d is defined as the vertical length of the upriser measured from the water level to the exit opening in this paper. In practice, however, the mixture of air and water is discharged to a height greater than the level of the exit opening corresponding to the velocity head $w_{l2}^2/2g$. Therefore, it may be proper to define H_d as $H_d+w_{l2}^2/2g$.

It is stated in this paper that the coefficient c_3 was calculated by using f_{l2}, which has been estimated by trial and error method. What were the values of f_{l2} attained in the present calculation?

(3) An air-lift pump design is given as example, and its estimated performance is shown on the basis of the present theory and data. It is regrettable, however, that the predicted performance has not been verified by an experiment. Has the performance been verified experimentally later?

Authors' closure

(1) The air-injectors employed in this experiment were radial injection type which had many holes in rows on the pipe periphery. Their details are as follows;

D=28.3 mm upriser: 32 holes of 1 mm diameter (16 holes in a row×2 rows, 4 mm distance between rows)

D=50.6 mm upriser: 64 holes of 1 mm diameter (16 holes in a row×4 rows, 4 mm distance between rows)

If it is necessary to evaluate the difference of characteristics between two certain air-injectors, the differences of additional loss at injection and of void fraction in upriser between them should be considered. Within the knowledge of the authors, however, it seems difficult to evaluate these differences from the two-phase flow data obtained up to date. Thus, the authors infer that these differences may be negligible in comparison with the accuracy of estimation of liquid holdup and frictional pressure drop in upriser except for a special air-lift pump with peculiar injector or very short upriser.

(2) It is difficult to derive the energy equation especially for two-phase gas-liquid flow system comparing with single-phase flow. In this analysis, the momentum equation is applied. Therefore, the efflux of water from upriser is taken into account as the momentum flux. The control surface defined in this paper may be relevant. Although the liquid holdup in the upper part of the upriser was slightly decreased due to the expansion of air, the calculated liquid holdup f_{l2} at the pipe exit was almost equal to mean holdup $\bar{f}_l$ in present cases.

(3) A test on the designed pump has not been done particularly.