A General Design Equation for Air Lift Pumps Operating in Slug Flow

Air lift pumps are finding increasing use where pump reliability and low maintenance are required, and where corrosive, abrasive, or radioactive fluids must be handled. Although air lifts are used in nuclear fuel reprocessing plants, no general, theoretically sound equation has been proposed in the literature for tall air lift design. Such an equation is developed from two-phase flow theory to predict the height to which an air lift pump operating in the slug flow regime can lift a given volumetric flow rate of liquid, given the air flow rate and pressure at the point of gas introduction. The widely used drift-flux model for the prediction of holdup is combined with an approximate relationship to predict pressure loss, and is substituted into the total pressure differential. Integration of the resulting equation provides an explicit formula for the calculation of lift. Experimental work using a variety of liquids in a 38 mm dia. air lift test installation supports the new design equation and demonstrates its flexibility.

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Air lift pumps provide a reliable means of raising corrosive or abrasive liquids and slurries because they incorporate no moving parts to erode and wear. Such pumps are ideal also for handling highly radioactive liquids because they require virtually no maintenance and can be treated as remote units. Although air lift pumps have a wide variety of possible applications, most studies have been concerned with dewatering mines or raising oil from dead wells. More recently, the importance of air lifts in moving liquids at nuclear fuel reprocessing plants has been realized, so that more accurate design equations are required. To date, liquid flow rate in the air lift has been predicted using either an energy balance or empirical correlations. The energy balance method, although valid, cannot take into account losses in the pump except in terms of an overall efficiency, which is generally not known and cannot be predicted accurately. Empirical correlations, although valid for the test data used, are not necessarily applicable over the very wide range of operating conditions and lift heights (from 2 to 2,000 m) typical in air lift applications. Nicklin (1963) provided a theoretically sound analysis of air lift pumps using a momentum balance. However, this analysis was accurate only in the design of short pumps, since there was no provision for variation in gas volumetric flow rate over the tube length. For taller pumps, the method has to be applied incrementally. In the analysis below, a differential momentum balance is integrated over the whole pump length to provide a closed-form equation that is valid for air lifts of any height operating in bubble or slug two-phase flow. This new equation compares favorably with data in the literature, and with experimental data from a 38 mm dia. test installation.

CONCLUSIONS AND SIGNIFICANCE

Results from a 38 mm air lift test installation support a new design equation, based on a two-phase flow momentum balance. The test apparatus was constructed to provide for the operation of the air lift in either vacuum or overpressure conditions, and four different liquids were used in the tests. Operating curves plotted from the results over a range of air flow rates agree with curves given by the new design equation. In addition, the new design equation is able to predict the performance of very tall air lifts used by Shaw (1920) in mine dewatering, and so appears valid over a wide range of operating conditions and configurations. In using this equation, one need know only the air flow rate, essential measurements of the pump, and friction factor for the flow in order to predict the liquid flow rate. Since all of these terms are readily available to the design engineer, this new equation will prove easier to use than energy balances, which require a knowledge of the pump efficiency, or empirical correlations, which are often cumbersome and not universally applicable.

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INTRODUCTION

Air lift pumps are used to raise liquids or slurries from wells or vessels, particularly where submerged mechanical pumps are undesirable or where simplicity of construction is required. In the simplest case, a vertical tube is partially submerged in liquid in a vessel, and air is introduced at or near the bottom of the tube. Being less dense than the liquid, the buoyant air-liquid mixture formed in the air lift tube rises and is expelled at the top of the pump, Figure 1a. An alternative U-tube airlift arrangement, in which the submergence is provided by a downcomer leg, is illustrated in Figure 1b. The two configurations are equivalent, except that some significant energy losses may occur in the U-tube arrangement due to the flow of fluid in the downcomer.

In theory, air lift pumps could operate with any interdispersion of the air and liquid phases, but in practice most air lift pumps operate in the slug flow regime, which persists over a wide range of air and liquid velocities (Taitel et al., 1980; Clark, 1984a). In slug flow large bubbles of air are surrounded by an annular film of liquid in contact with the pipe wall. These air bubbles are separated by slugs of liquid (spanning the whole tube diameter) which may contain a few smaller air bubbles. Although sufficient literature exists to describe the slug flow regime (Govier and Aziz, 1972; Griffith and Wallis, 1961), current design techniques are still based on semiempirical equations such as those presented by Shaw (1920) and O'Neill (1975). These equations become inaccurate when applied to any nonstandard air lift designs. In this paper, a new design equation is derived from two-phase flow theory and is shown to be applicable over a wide range of operating conditions and pump heights.

Literature Survey

Although air lifts have been used in mine dewatering since the eighteenth century, until the last three decades two-phase flow theory was inchoate and could not be applied to pump design. The earliest theoretical approaches relied on an energy balance, equating the energy possessed by the compressed air introduced at the base of the pump to the useful work done in raising the liquid from the level in the well to the top of the pump. Shaw (1920) noted and derived several such analyses. Assuming isothermal operation, for a pump with 100% efficiency the volume of free air (at atmospheric pressure), V_g, required to raise some volume of water, V_w, through L feet, is given by

$$V_g/V_w = \varrho_w g L/[P_2 \ln(P_o/P_2)]$$

where P_o is the pressure at the bottom of the tube, where gas was introduced, and P_2 is the pressure at the discharge, typically atmospheric pressure. However, in the air lift pump there are energy losses that can be ascribed to losses in the bulk flow and to the slippage of the gas bubbles through the liquid. Due to these losses, most early air lifts operated at 35 to 55% energy efficiency, so that the theoretical energy balance could not be applied except with an energy efficiency term, typically 50% (Shaw, 1920). Modified energy balance equations in various forms have appeared in the literature in the intervening time, and are still recommended for air lift design today (O'Neill, 1975). Nevertheless, the energy balance is insufficient for accurate design, since the true pump efficiency is seldom known until the pump has been constructed, or the design subjected to more rigorous two-phase flow analysis. In cases where low submergences or narrow bore pipes are used, these energy balances overpredict flow rate to a significant degree.

In the petroleum engineering field, design of gas lifts for recovering oil from slow or dead oil wells has followed a different path, and most designs rely on graphical correlations, or empirical equations based on extensive full-scale well data (Lawson and Brill, 1975; Brown, 1967; Ros, 1961). Whereas most chemical engineering air lift designs are concerned with raising water or aqueous solutions, the petroleum engineer must deal with a range of viscosities and densities in considering the oil to be lifted. In addition, the oil under pressure at the base of the well generally contains varying amounts of light fractions, which become gas at higher elevations (lower pressures) in the tube. Therefore the mass flow of gas in the string is not constant and further complicates the prediction of gas lift performance. Much effort in the presentation of petroleum engineering correlations has been devoted to dealing with these problems, but the work is unsuited to the design of air lifts in the chemical and nuclear engineering fields.

Nicklin (1963) provided the first useful analysis of the air lift pump based on a momentum balance. The pressure gradient in the air lift tube was separated into the hydrostatic head term and frictional losses. The hydrostatic head was predicted from the holdup of liquid and gas, which were evaluated using a two-

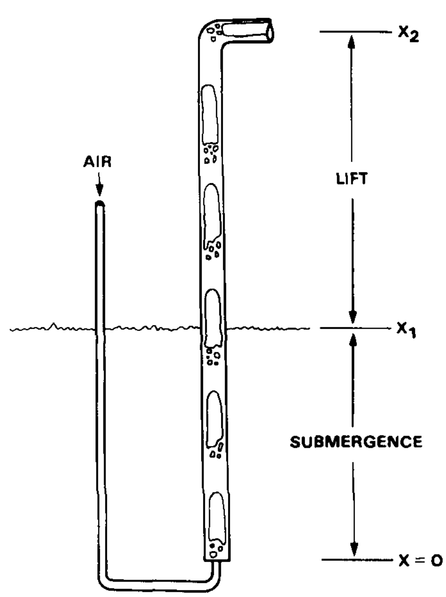

Figure 1a. Air lift pump: air lift tube immersed in a well or vessel.

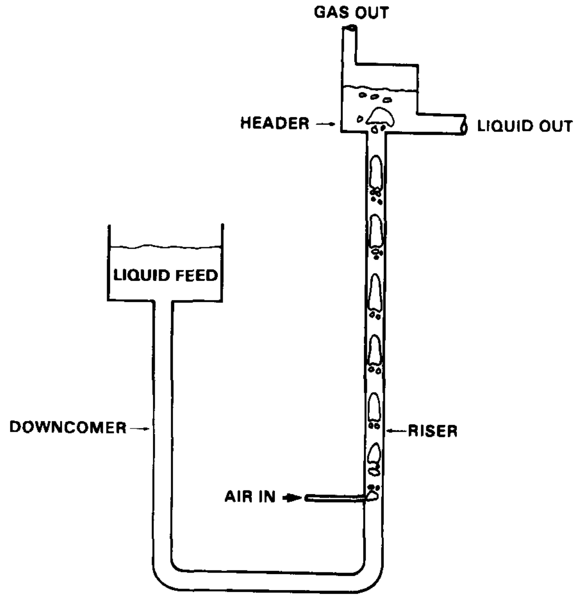

Figure 1b. Air lift pump: U-tube arrangement.

phase flow drift-flux model, which is discussed in more detail in the analysis below. From an energy balance point of view, losses due to the bubble slippage are accounted for in the prediction of holdup, and frictional losses are accounted directly, using a modified single-phase flow (D'Arcy) equation. However, Nicklin's differential analysis was really valid only at a single height in the pump, since the equation assumed a constant value for the gas superficial velocity, which increases with increasing pump height. Stenning and Martin (1968) successfully applied with Nicklin analysis to a short test pump, using an average air superficial velocity, but this approach would overpredict the required air rate in taller pumps (Dabolt and Clark, 1985).

More recently, Husain (1975) and Husain and Spedding (1976) proposed an original energy balance, assuming a spectrum of energies for the bubbles in the air lift tube. The air sparger was viewed as an energy-emitting orifice, and the analysis assumed inviscid liquid and gas. However, this analysis still required a term for "efficiency of gas usage." Moreover, the definition of this efficiency differed from that used in the early energy balances, being inversely proportional to the gas fraction present. Jeelani et al. (1979) have demonstrated that the "constants" derived in the Husain and Spedding analysis may vary when applied to narrow bore lifts, so that the analysis would not appear to be suited to general air lift design. Although this analysis supplies the correct shape for most of the air lift operating curve, the constants for the equation must be found by regression.

Dabolt and Plummer (1980) examined a wide range of data for air lifts in the 12 to 50 mm dia. range, for application in nuclear fuel reprocessing. The submergence ratio, or ratio of submergence to total pump height, was found to be the dominant variable governing lift performance, and the following design equation was proposed

$$V_a = \ln(1 - V_w/A)/B$$

where V_a is the volume of air required to lift a volume V_w of water, and where A and B are quadratic functions of the submergence ratio, with coefficients dependent on pipe diameter. Frictional losses before the sparger, and in the air lift tube, and pressures in the head pot and feed tank, were accounted for by adjusting the geometric submergence ratio to an effective submergence ratio. The equation found better agreement with data than previous design approaches used in fuel reprocessing, but cannot be used in large bore air lift design outside of the nuclear fuel reprocessing field, since correlations for A and B have been proposed only in the 12 to 50 mm pipe diameter range.

Although many methods are available for predicting the liquid flow rate in air lift pumps, none of the equations is both theoretically sound and accurate in prediction of pump performance. What is required is a general analysis that can be applied with confidence to any air lift pump configuration.

ANALYSIS

An air lift design equation is derived below, using a momentum balance based on established two-phase flow theory. This equation is derived making assumptions that are valid for the practical operating range of air lift pumps, and is not intended to be applied without consideration to all vertical two-phase flow situations.

Consider a vertical pipe partly submerged in a reservoir of water, and let the base of the tube have a reference height of zero (Figure 1a). If the tube is submerged a depth of x_1 below the surface of the supply vessel, then the static pressure, P_o, at the base of the pipe, is given by

$$P_o = \varrho_1 g x_1 \tag{1}$$

where ϱ_1 is the liquid density. Neglecting entrance effects, P_o is therefore the pressure at the base of the air lift tube, at the point of air introduction. In the case where the submergence is provided by a downcomer, as in a U-tube air lift (Figure 1b), P_o must be reduced by the losses occurring in the downcomer tube up to the point of gas introduction. The need to account for these losses has been emphasized by Dabolt and Plummer (1980) and Husain and Spedding (1976).

The top of the air lift tube, height x_2, will discharge to atmosphere, or into a vessel at a known pressure, P_2. Acceleration effects due to the expansion of the gas over the pump height are a very small fraction, typically less than 1%, of the overall gradient in the tube and can be neglected. Therefore, where the small contribution due to the air phase density in the air lift tube is also disregarded, the total differential for the pressure gradient in the air lift tube is given by

$$-dP = [\varrho_1 g(1 - \epsilon) + F]dx \tag{2}$$

where ϵ is the cross-sectional average air void fraction, and F is the irreversible loss per unit pipe length, at a height x in the pipe. By integrating Eq. 2 between the entrance condition ($x = 0, P = P_o$), and the exit condition at the top of the air lift ($x = x_2, P = P_2$), it is possible to obtain an explicit formula to predict the total height of the pump, x_2, and hence the lift, $x_2 - x_1$. However, in order to integrate, both the air void fraction and the frictional pressure loss must be predicted as a function of air and liquid flow rates. Stenning and Martin (1968) chose to evaluate the void fraction, ϵ, at the mean pressure in the pump, but this approach may be inaccurate in the case of tall pumps, where ϵ may vary greatly over the tube height.

In vertical slug flow one may not simply equate the *in situ* air void fraction, ϵ, with the volumetric flowing air fraction, $W_g/(W_g + W_l)$, because the velocities of the air and liquid phases differ in the pipe. First, the slugs of air rise relative to the liquid, so that liquid flows back in the annular film surrounding the air slug. Second, in occupying the central region of the pipe, the slug is traveling in a region of flow which is moving faster than the cross-sectional average velocity. Both of these factors contribute to raising the slip ratio, or ratio of air to liquid average velocity, U_g/U_l; most air lift pumps operate at low liquid velocities, with slip ratios between 1.5 and 2.5 (Stenning and Martin, 1968). The air voidage, ϵ, may be found from the relationship

$$W_g/W_l = U_g(1 - \epsilon)/U_l\epsilon \tag{3}$$

where W_g and W_l are the air and liquid superficial velocities in the pipe. In practice, the air void fraction is accurately predicted by using the drift-flux model described by Nicklin et al. (1962) and by Zuber and Findlay (1965).

$$W_g/\epsilon = C_o(W_g + W_l) + V_v \tag{4}$$

where C_o is a constant accounting for the location of the slug in the faster region of flow, and V_v is the drift velocity between the air phase and the total flow. The drift-flux model has been used extensively to predict holdup in a variety of vertical two phase flows (Clark and Flemmer, 1984, 1985; Ardron and Hall, 1980; Clark, 1984b; Miller et al., 1984). A range of values for C_o in bubble and slug flow have been reported in the literature to date (Ishii, 1977; Ishii and Grolmes, 1978; Clark and Flemmer, 1984, 1985), and recently Miller et al. (1984) have presented values for C_o in gas-slurry slug flow. Although C_o may not be predicted universally, the best value currently available for slug flow is $C_o = 1.2$. Data have also been presented by Shipley (1984) to show that a value of $C_o = 1.2$ can be used for two-phase flow in large diameter pipes. Drift velocity, V_v, is given by the formula

$$V_v = 0.35(gd)^{0.5} \tag{5}$$

where d is the pipe diameter.

Frictional loss in slug flow is less readily predicted. However, because air lift pumps operate at relatively low liquid velocities, the hydrostatic head is much larger than the frictional head loss and an approximate relationship for the frictional contribution will suffice. For example, in most air lift pumps frictional losses represent only 2 to 10% of the total head, so that the use of a simple pressure-loss expression is justified in the interests of obtaining a closed-form design equation, and will cause little loss of accuracy. Models to predict pressure loss in slug flow are reviewed below. Nicklin et al. (1962) have argued from continuity that the velocity of the liquid slug in the pipe is equal to the total superficial velocity, $(W_g + W_l)$. Since the liquid slugs are in contact with a fraction of approximately $(1 - \epsilon)$ of the pipe wall, this suggests that the frictional loss per unit length due to the liquid slugs is given by

$$F = 4\varrho_l f(W_l + W_g)^2(1 - \epsilon)/2d \qquad (6)$$

where the friction factor is found for the total flow velocity using the liquid phase properties. The same equation was reached by Orkizewski (1967), while Griffith and Wallis (1961) proposed that

$$F = [4\varrho_l f(W_l + W_g)^2/2d][W_l/(W_l + W_g)] \qquad (7)$$

Stenning and Martin (1968) chose the Griffith and Wallis equation to describe pressure loss in their analysis of air lift pumps, but found experimentally that the friction factor was larger than the friction factor diagram predicted, although the flow in their short length of pipe may not have been fully developed. A disadvantage of these simple correlations when frictional losses must be known precisely is that none takes into account the shear stress generated at the wall by the annular film of liquid around the bubble. However, in air lift pump design this contribution to the overall head may be considered negligible.

The approach of Lockhart and Martinelli (1949) has gained wide acceptance. Although not the most sophisticated approach, it provides a simple expression sufficiently accurate for air lift analysis. Irreversible two-phase pressure loss in the pipe is given by the product of a two-phase flow multiplier, ϕ_l^2, and the frictional pressure loss per unit length of pipe, D, that would occur if the liquid alone were flowing in the pipe.

$$F = D\phi_l^2 \qquad (8)$$

The pressure loss that would occur with the liquid flowing alone is given by

$$D = 4\varrho_l f W_l^2/2d \qquad (9)$$

where f is the friction factor, found from a conventional diagram. Values for the two-phase flow multiplier are obtained graphically from a correlation with a factor X^2, defined as the ratio of the pressure loss that would occur if the liquid were flowing alone in the pipe, to the loss that would occur if the air were flowing alone in the pipe. Oshinowo and Charles (1975) found that frictional losses in slug flow were slightly lower than are predicted by the Lockhart and Martinelli correlation, and that at low liquid velocities the direction of shear stress at the wall could even reverse due to the liquid flowing back around the rising slug. However, when it occurs, this negative shear is very small relative to the hydrostatic head, and deviation of the Lockhart-Martinelli prediction in this region would not affect the overall air lift pressure gradient significantly. Of all the models presented above, that of Lockhart and Martinelli was chosen to predict frictional pressure loss. For air voidages below 50%, as encountered in the efficient operating range of air lift pumps, the graphical Lockhart and Martinelli correlation is well represented by an expression of the form

$$F = D(1 + n\epsilon) \qquad (10)$$

In this range of air voidage, using $n = 1.5$, Eq. 10 deviates from the Lockhart and Martinelli correlation typically by 1 or 2% and at most by 6%. Moreover, Eq. 10 does not deviate significantly from the equations proposed by Nicklin et al. (1962) and Stenning and Martin (1968) over the effective range of air lift pumps. Accordingly, Eq. 10 was chosen to describe the frictional losses in the pump. Although it is acknowledged that more sophisticated models might be adopted to predict the slug flow pressure loss, these will cause only a small increase in overall accuracy for practical air lift pump designs, and will prohibit the development of a design equation in closed form. Combining Eqs. 2 and 10 results in:

$$-dP = [\varrho_l g(1 - \epsilon) + D(1 + n\epsilon)]dx. \qquad (11)$$

Next, Eq. 4 for the prediction of air void fraction is substituted into the total pressure differential, Eq. 11. Clearly, the superficial air velocity in Eq. 4, W_g, will vary with pressure over the height of the pump, and account must be taken of this variation in the total differential. It is valid to assume that the air behaves in an ideal fashion, and that expansion of gas in the pump is isothermal. Where the mass flow rate of air to the base of the pump is G, the superficial air velocity, W_g, is given by

$$W_g = GP_a/AP\varrho_g \qquad (12)$$

where P_a is atmospheric pressure, A is the pipe cross-sectional area, and ϱ_g is the density of the air at atmospheric pressure and at the temperature of the pump. Setting $M = GP_a/A\rho_g$, and combining Eqs. 4, 11, and 12,

$$-dP = \left[\varrho_l g\left(1 - \frac{M}{C_o(M + W_l P) + V_v P}\right) + D\left(1 + \frac{nM}{C_o(M + W_l P) + V_v P}\right)\right]dx \qquad (13)$$

Multiplying through by the term $C_o(M + W_l P) + V_v P$, and collecting pressure terms on the lefthand side of the equation,

$$\left(\frac{C_o M + (C_o W_l + V_v)P}{\varrho_l g C_o M - \varrho_l g M + DC_o M + DnM + (\varrho_l g + D)(C_o W_l + V_v)P}\right)dP + dx = 0 \qquad (14)$$

Integrating between the points $(x = 0,\ P = P_o)$ and $(x = x_2,\ P = P_2)$,

$$\frac{P_2 - P_o}{(\varrho_l g + D)} - \left(\frac{DnM - \varrho_l g M}{(\varrho_l g + D)^2 S}\right)\log\left(\frac{R + (\varrho_l g + D)SP_2}{R + (\varrho_l g + D)SP_0}\right) + x_2 = 0 \qquad (15)$$

where $R = \varrho_l g M(C_o - 1) + DM(C_o + n)$ and $S = C_o W_l + V_v$. Since every variable except x_2 is known in Eq. 15, an explicit equation for the evaluation of x_2 has been developed, thus avoiding the use of incremental methods in the design of air lift pumps. The lift of the pump is then given by $x_2 - x_1$, and the submergence ratio, or ratio of submergence to total pump length, by x_1/x_2.

Although Eq. 15 can be used directly to evaluate the lift of the pump, given flow rates and pump submergence, it cannot be rearranged in closed form to give the liquid flow rate as a function of gas flow rate if the lift is already known. Therefore, to generate an operating curve Eq. 15 must be used on a trial and error basis, adjusting the liquid flow rate for a given gas flow rate until the predicted lift matches the actual pump lift. However, this method is still rapid in use, and when applied to tall pumps is superior to earlier momentum balances, which required an incremental analysis over the length of the air lift tube as well as a trial and error test.

Experimental operating curves of liquid flow rate vs. gas flow rate for air lift pumps demonstrate that the same liquid flow rate can often be obtained using two different air rates. Liquid flow rate rises with increasing gas flow rate, through a point of maximum energy efficiency, to a point of maximum liquid flow rate (Figure 2). Thereafter, higher has superficial velocities in the pump cause increasing frictional losses, and the liquid flow rate falls off. In this region of high gas flow, exit effects, flow acceleration, and entrance effects at the gas sparger or orifice become significant. Moreover, the pump can sometimes operate in annular flow or very irregular pulsating flow rather than slug flow in this region, so that holdup will not be predicted by the analysis used above. Not only is prediction of the pump performance difficult at very high gas rates, but no pumps are operated in this region in practice. Most pumps are operated at or near the point of maximum efficiency on the rising section of the operating curve. Equation 15 is therefore intended to predict the first (low gas rate) solution only, since this is the only solution of interest in industrial air lift operation.

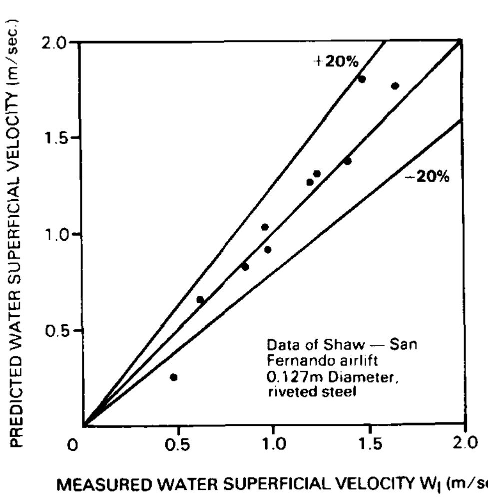

Figure 3. Comparison of the tall air lift data of Shaw (1920) with the prediction of Eq. 15.

COMPARISON OF ANALYSIS WITH EXISTING DATA

The most stringent test for the theory developed above is whether or not it can predict the behavior of very tall air lifts, where the gas volumetric flowrate changes significantly over the height of the pump. For short air lifts, it is sufficient to use an average gas flowrate in the pump (Stenning and Martin, 1968), but this approach is not valid for taller pumps. Shaw (1920) has supplied data on a variety of tall air lifts (up to 400 m high) used to dewater the Tiro and San Fernando mine shafts in Mexico. Some of Shaw's pumps had variable diameters, so that they could not be compared with Eq. 15, but the pump used in the San Fernando shaft was of constant 5 in. (127 mm) diameter so that it was suited to comparison with the theory. (Variable diameter can offer increased efficiency for very high single-stage lifts, but is seldom used in modern designs.) In Table 13 (p. 436) of Shaw's paper, operating data for five different submergence ratios and heights of the San Fernando lift are supplied. For each configuration, data are given for the point of maximum flow rate for the pump, and for a point in the range of higher operating efficiency, so that a total of ten test points was available.

Shaw's data compare favorably with the predictions of the new equation; as shown in Figure 3. The pipes were of spiral riveted steel, which has a high equivalent roughness, so that a friction factor of 0.01 was chosen as typical for the flow. A value of 1.2 was used for the profile constant, C_o, the exponent n was taken as 1.5, and the drift velocity, V_v, was calculated at 0.39 m/s. Agreement between Shaw's data and the new design equation is good, with all predictions but one falling within the 20% lines on the parity plot, Figure 3. However, it is suspected that the one point in disagreement may be reported in error, since it does not follow the trends in the rest of Shaw's data. Small errors in predicting the remaining points may be due to a phase change from slug to annular flow near the top of the San Fernando pumps, since air superficial velocities were very high in this region as a result of the low pump submergence used. The theory developed above was for slug flow, and will become inaccurate in predicting annular flow behavior, in particular the liquid holdup in annular flow.

Comparison with Shaw's data has demonstrated that the new design equation is effective in predicting the performance of tall air lift pumps, and that a value of $C_o = 1.2$ is satisfactory for use in large diameter air lifts. Experimental data described below were used to test its accuracy in the design of mid-size air lift pumps used in nuclear fuel reprocessing.

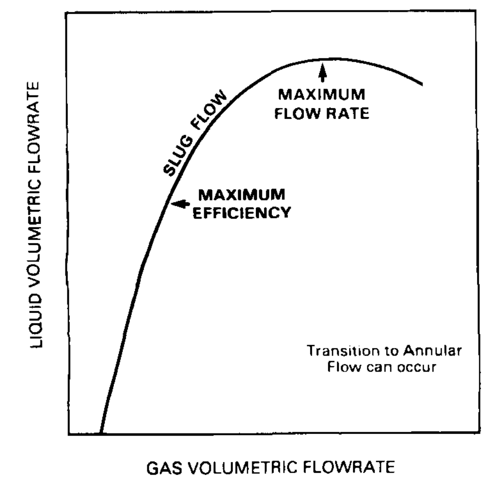

Figure 2. Typical air lift pump operating curve. Equation 15 is valid for the ascending portion of the curve, up to the maximum liquid flow rate.

EXPERIMENTAL

Tests were undertaken using an air lift installation with a 38.1 mm dia. tube, described in detail by Dabolt and Plummer (1980), and diagramed in Figure 4. The lift was constructed of schedule 40 stainless steel pipe, and gas inlets were situated so that submergence ratios of 60, 70, and 80% could be chosen. The head pot of the pump was constructed of plexiglass and stainless steel, and contained two baffles and a weir for measuring flowrate of liquid. Air supply was measured using a rotameter.

A 276 L tank was used as the feed reservoir, and the lift discharged liquid via the head pot into a smaller 62 L reservoir. Liquid was lifted back to the larger reservoir by a separate 50 mm dia. air lift pump. In addition, the equipment vent lines were connected via PVC piping to the off-gas header, with valves and pressure gauges installed in each line. This allowed either pressure or vacuum to be applied to each piece of equipment as desired.

Eleven test runs, of approximately ten points each, were made to determine the effects of submergence, liquid properties, and vacuum/pressure on the system. Data were gathered for the ascending section of the air lift operating curve, since operating points to the right of the point of maximum flow are of no practical interest to the air lift operator. Conditions for the runs are listed in Table 1. Three nonaqueous liquids used in the runs were 30% TBP in dodecane, the same solution saturated with sodium carbonate, and with nitric acid.

Air flow rate was measured accurately with a rotameter, while expected errors in liquid flow rate measurement are shown by error bars in Figure 5.

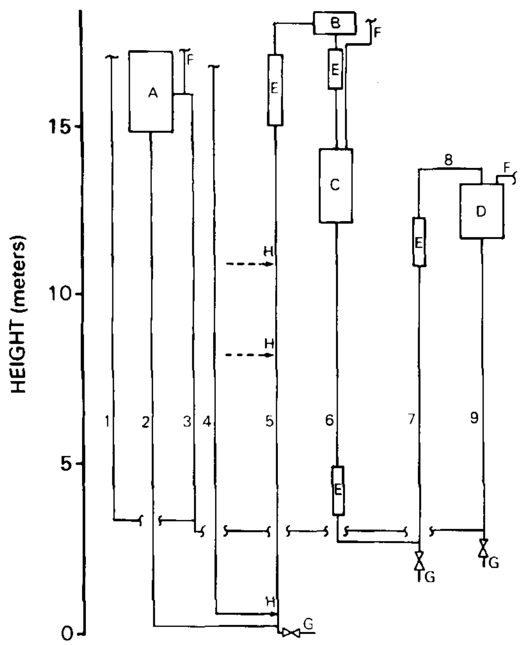

Figure 4. Experimental test installation.

A. 276 L tank
B. Metering head pot
C. Separator
D. 62 L tank
E. Glass viewing sections
F. Vents
G. Drains
H. Points of air introduction

1. 19 mm air supply
2. 38 mm liquid supply
3. 50 mm air lift
4. 19 mm air supply
5. 38 mm experimental air lift
6, 7. 38 mm drain from head pot
8. 150 mm horizontal feed to tank
9. 50 mm tank drain

RESULTS

Points on operating curves for the runs, as plots of liquid flow rate vs. free air flow rate, are shown in Figures 5 to 9. In each case the given curve is predicted using the new design equation, with a friction factor of 0.01, $n = 1.5$, $C_o = 1.2$, and the drift velocity $V_v = 0.21$. It should be stressed that each of these constants is known prior to obtaining the experimental results, and is not found by regression. The friction factor of 0.01 was chosen as representative of the flows from a conventional friction factor diagram. More accurate evaluation of the friction factor throughout the operating range was found to produce no significant change in the curves, because the frictional losses were small at the low liquid velocities used. This amplifies the claim that very accurate frictional loss models are not required for practical air lift design. Also, losses due to the liquid flow in the downcomer were small, and were neglected.

Agreement between the data and model is good, except in the case of run 2-8, where trends in the data suggest an operating curve with a lower slope than is predicted. However, noting the operating conditions, it is clear that the curves for runs 2-1 and 2-8 should not cross at any point, while the data suggest that they do. It was concluded that the first three experimental points for run 2-

Table 1. Conditions for Experimental Runs

Run No.	Submergence, %	Pressure	Fluid
2-1	82	Atmospheric	Water
2-2	82	− 25.4 cm water	Water
2-3	70	− 25.4 cm water	Water
2-4	60	− 25.4 cm water	Water
2-5-1	82	− 25.4 cm water	Solvent*
2-5-2	82	Atmospheric	Solvent*
2-6-1	82	− 25.4 cm water	Solvent + carbonate**
2-6-2	82	Atmospheric	Solvent + carbonate**
2-7-1	82	− 25.4 cm water	Solvent + nitric acid†
2-7-2	82	Atmospheric	Solvent + nitric acid†

*30% TBP in dodecane, sg. 0.8227.
**Solvent saturated with carbonate, sg. 0.8295.
†Solvent saturated with nitric acid, sg 0.8305.

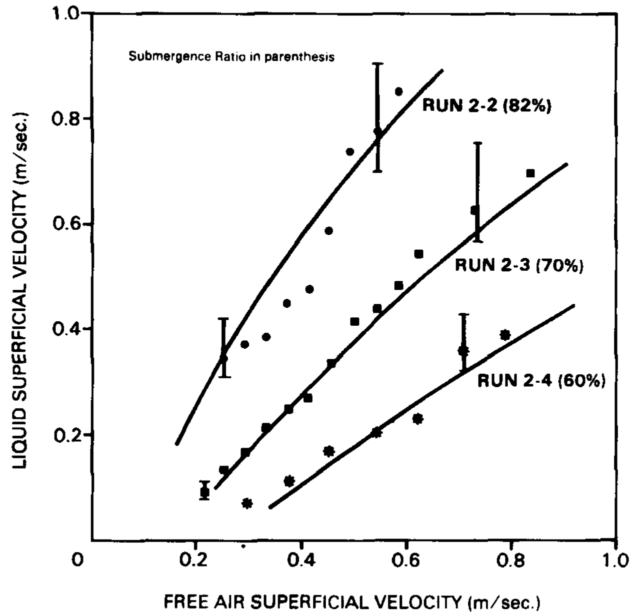

Figure 5. Comparison fo test data for water runs with operating curves generated using the new design equation.

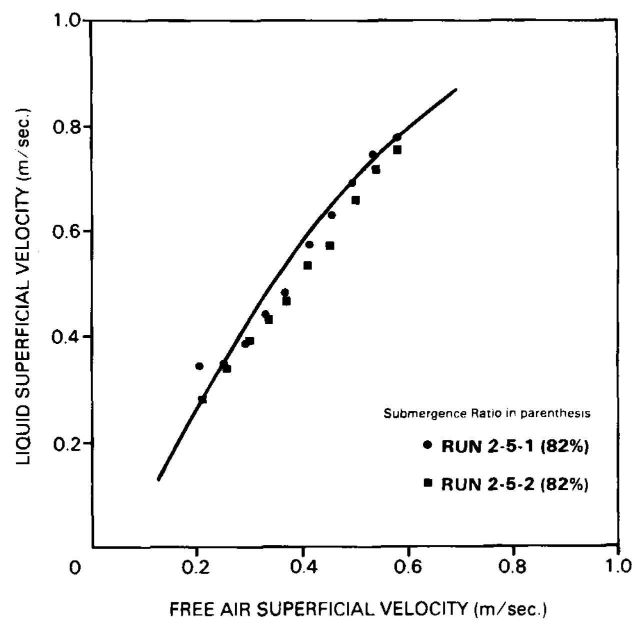

Figure 6. Comparison of test data for solvent-only runs with operating curves generated using the new design equation.

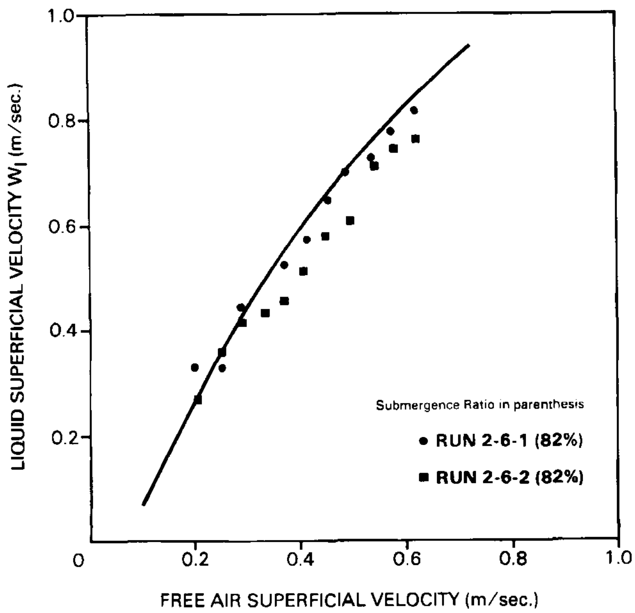

Figure 7. Comparison of test data for solvent + carbonate runs with operating curves generated using the new design equation.

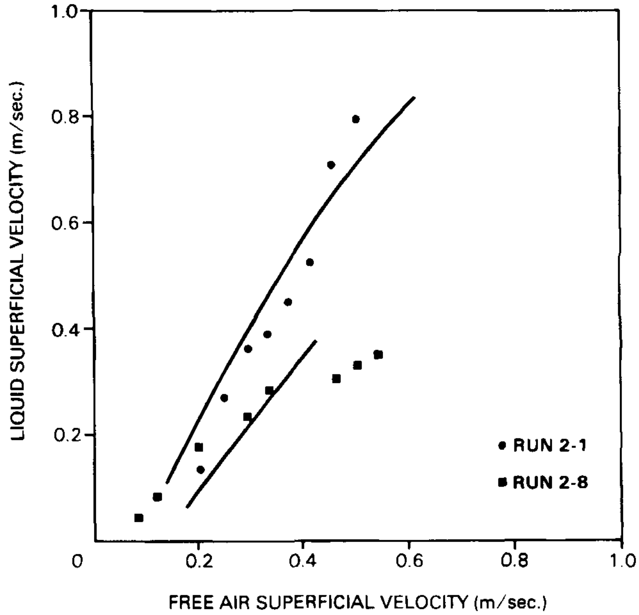

Figure 9. Comparison of test data with operating curves generated using the new design equation.

8 represent too high a liquid flow rate, although the cause has not been identified.

In many of the experimental runs, the liquid flow rate was slightly lower than predicted over the mid-range of the data. This depression in the operating curve has not been fully explained, but might be attributed to the stability of the two-phase flow in the U-tube apparatus. Theory of pump stability has been discussed mathematically by Hjalmars (1973) and Apazadis (1980, 1982, 1983), but has not been related directly to such perturbations in the operating curve. However, the overall agreement between theory and data was good, thus demonstrating that it is possible to predict pump performance without resorting to regression on existing pump data.

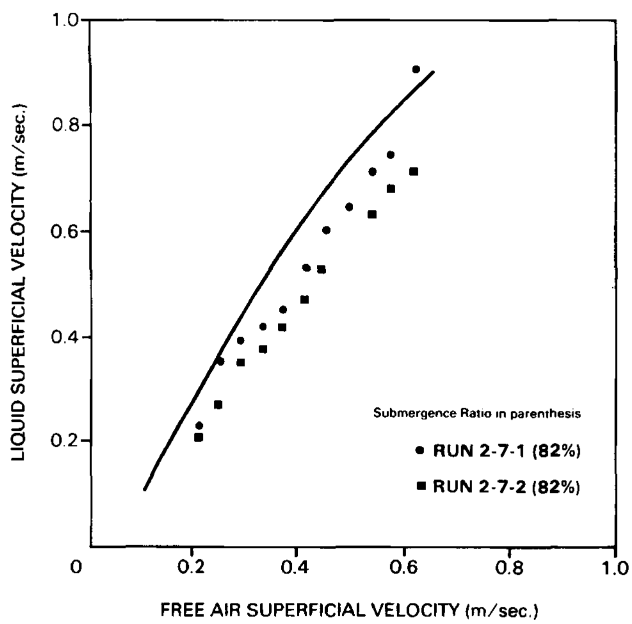

Figure 8. Comparison of test data for solvent + nitric acid runs with operating curves generated using the new design equation.

COMPARISON OF DATA WITH OTHER MODELS

Several other design equations were also compared with run 2-1, as shown in Figure 10. The Husain and Spedding (1976) analysis predicted too low a liquid flow rate when solved for the operating conditions in run 2-1, using the constants supplied by Husain and Spedding in their paper. It was deduced that although the

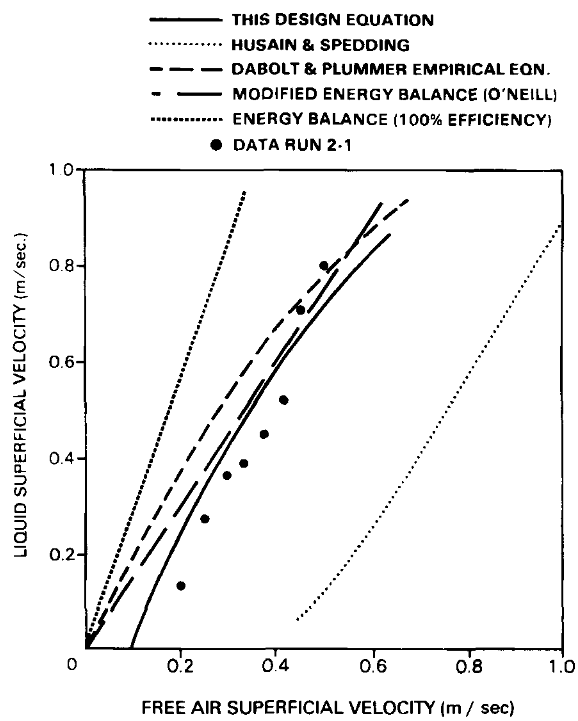

Figure 10. Comparison of several design equations with the data of run 2-1.

The ideal energy balance (100% efficiency) and Husain and Spedding method are both inaccurate. The Ingersoll-Rand equation presented by O'Neill (1975) cannot predict the operating curve except as a straight line.

form of the Husain and Spedding equation might reflect the air lift performance, the terms in the equation were not necessarily constant over a range of pump geometries. Moreover, the Husain and Spedding analysis is in error at very low gas flow rates, where it still predicts a very small liquid flow rate; in other words, the predicted operating curve does not cross the horizontal axis at a positive value for air flow rate. A correct solution will predict an operating curve which does cross the horizontal axis into the fourth quadrant, this region corresponding to a hypothetical downflow of liquid in the pump, if the whole length of the air lift tube is to remain full of a gas-liquid mixture. Only the new equation developed above predicts this situation correctly.

The Ingersoll-Rand equation (O'Neill, 1975) agrees quite favorably with the data, particularly in the region of maximum efficiency. However, since it is a modified energy balance with a constant efficiency assumed from operating experience, the operating curve is a straight line which becomes inaccurate at very high or very low gas flow rates.

The empirical equation of Dabolt and Plummer (1980) and the new equation developed above find superior agreement with the experimental data in Figure 10.

DISCUSSION

The new model agrees with experimental results and data from the literature. Equation 15 reduces correctly to the single-phase pressure differential when the air rate is zero, and has found favorable comparison with the empirical Ingersoll-Rand equation (O'Neill, 1975) in the region of maximum efficiency. However, the Ingersoll-Rand equation predicts only the volume of air required to raise a given volume of water, given the lift and submergence, and so cannot take into account frictional losses, which may become significant in pipes of smaller bore. In addition, the slip velocity will differ between the case of a high flow-velocity pump with a small pipe bore, and a low-velocity large bore pump, although the volume rate of liquid lifted may be identical in each case. The Ingersoll-Rand approach cannot account for this difference. Equation 15 takes into account both the variation of slip velocity and the presence of frictional loss, and may therefore be applied over a wide range of operating conditions for air lift pumps in slug flow.

The linear assumption for the relationship between pressure loss and voidage, Eq. 10, will cause negligible error in the final prediction of lift. Equation 10 deviates from the widely used Lockhart and Martinelli (1949) model by no more than 6% at air voidages up to 0.5. Moreover, the frictional loss in typical air lift pumps is at least an order of magnitude smaller than the hydrostatic head, so that overall errors incurred by this linear assumption would total no more than 0.5%. In pipes of smaller bore, for example the extreme case of a 50 mm pipe carrying a flow at 2 to 3 m/s, the frictional loss is only of the same order of magnitude as the hydrostatic head, so that the error incurred by the linear assumption would be less than 3%.

Using the new equation, air lift pumps are rapidly designed. In conjunction with an equation describing the efficiency of the pump, such as that presented by Nicklin (1962), the closed solution developed in this article provides an efficient means for optimizing such variables as pipe diameter, gas flow rate, and submergence ratio. The design of a pump with minimum energy requirements by using this method is more rapid than by using incremental methods, and more accurate than by using existing empirical equations.

NOTATION

A = pipe cross-sectional area, m^2
C_o = constant for the drift-flux model
D = frictional pressure loss per unit length in single phase flow, Pa/m
d = pipe diameter, m
F = frictional pressure loss per unit pipe length in two-phase flow, Pa/m
f = friction factor
G = mass flowrate of air, kg/s
g = acceleration due to gravity, m/s^2
M = a product of gas superficial velocity and pressure, Pa•m/s
n = constant
P = pressure, Pa
R = relationship extracted for clarity in Eq. 15
S = relationship extracted for clarity in Eq. 15
U = average velocity over pipe cross section, m/s
V_v = drift velocity of slug, m/s
W = average superficial velocity over pipe cross section, m/s
x = height above base of pump, m
ϵ = gas void fraction
ϱ = density, kg/m^3
ϕ_l^2 = two-phase flow multiplier

Subscripts

a = atmospheric
g = air, gas
l = liquid
o = at gas sparger (except C_o)
w = water
1 = at liquid level in reservoir
2 = at top of pump

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