The performance and modeling of liquid jet gas pumps

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This paper reports an experimental and theoretical study of a two-phase jet pump, involving a liquid primary flow and a gas secondary one. The ultimate objective was to produce a mathematical model and computer program in BASICA for use with microcomputers in the industrial design of such gas pumps, the experimental phase of the project being required to produce empirical values for some of the variables used in the computer program. Comparison with other, rather sparse, work on this type of jet pump shows the current one to be a worthy representative of the species, and cross-checks between the computer program predictions and rig results show good agreement. The major advance has been the ability of the mathematical model to predict performance under almost any operating conditions, previous restrictions to high-back-pressure cases being removed.

Keywords: jet pumps; induction pumps; ejectors; LJGP; two-phase flow

Introduction

Single-phase jet pumps, otherwise known as ejectors or induction pumps, dealing with primary and induced flows of the same fluid, have been used in engineering applications for many years, but over the last two decades a two-phase variant, using a liquid primary flow to induce a secondary gas flow, has become more widely known. These liquid jet gas pumps (LJGPs) have found application in, for example, the chlorination of water supplies, the removal of the gas phase from condensers, and in the addition of gas in biochemical reactions.

The LJGP was introduced by Hoeffer[1] in 1922, but a more thorough analysis of its performance was not made until 1952 by Takishima[2]. Even then, operating efficiency η, defined in terms of gas compression, was only about 10% because the devices were simply not dragging in enough gas. The ratio ϕ of gas to liquid volumetric flow rates was about 0.2, only a seventh of what might be expected in a modern device. Some improvement was obtained using four liquid jets instead of a single one, presumably because the jet surface area was thereby increased, but this trend, incidentally, is not confirmed by the results of the present project. High gas-to-liquid ratios and maximum efficiencies typical of modern versions of the LJGP (around 40%) were first achieved by Witte[3,4,5] in the 1960s.

Major papers on the LJGP were published in 1974 by Cunningham[6] and Cunningham and Dopkin[7]; the former was concerned mainly with performance, the latter with jet breakup distances. These two sources have had an important influence on the way in which the current project has been pursued, and reference will often be made to them. The comprehensive equations given by Cunningham can be simplified by approximation in most cases because the ratio of gas to liquid densities is always of the order 10^{-3}, thus rendering many terms in the equations negligible by comparison with the others. One disadvantage of Cunningham's results is that, in general, they apply only to cases where back pressure is high enough to ensure homogeneous flow at the downstream end of the mixing tube. This will be seen to be restrictive.

The prime purpose of the current project has been to develop simplified but realistic equations for jet pump performance so that a consequent mathematical model could be used to assemble a computer program for use by manufacturers in designing liquid/gas jet pumps for specified duties against known back pressures. The unknown parameters in the governing equations were found from the performance results of an experimental rig described later.

Conditions at state 2, which is of course a mixture, are used in the K_t term. This has the advantage of also being the input station for the diffuser and would be analogous to basing K_t on downstream conditions in a sudden expansion pipe loss calculation. There is no alternative here anyway, since conditions at state 1 are highly nonuniform.

If we normalize by division by A_2 and by the dynamic pressure Z in the liquid jet ($=\rho_1 V_1^2/2$) we have

$$\frac{p_2-p_1}{Z}=\frac{2m_1}{A_2\rho_1 V_1}+\frac{2m_g}{A_2\rho_1 V_1^2}-2b\frac{m_2}{m_1}\frac{V_2}{V_1}-K_t b\frac{m_2}{m_1}\frac{V_2}{V_1}$$

$$=2b+2b\frac{m_g}{m_1}\frac{V_g}{V_1}-(2+K_t)b\frac{m_2}{m_1}\frac{V_2}{V_1}$$

$$=2b\left[1+\frac{\gamma\phi_1^2}{c}-(1+\tfrac{1}{2}K_t)(1+\gamma\phi_1)\frac{V_2}{V_1}\right]$$

At this point, some approximations can be made since the liquid density is nearly always a thousand times that of the gas, so ϕ_1 will be of order 10^{-3} and $\gamma\phi_1^2/c$ will be potentially about the same. An approximate value of V_2/V_1 can be obtained by remembering that at station 2 the bulk of the mass of the froth is in the form of liquid, even if the volumetric flow rates of the two phases are comparable. We therefore have

$$\rho_2 A_2 V_2 \simeq \rho_1 A_j V_1$$

or

$$\frac{V_2}{V_1}\simeq\frac{A_j}{A_2}\frac{\rho_1}{\rho_2}\simeq b\frac{\rho_1}{\rho_2}$$

But

$$\rho_2=\frac{m_1+m_g}{Q_2}=\frac{1+m_g/m_1}{(1+\phi_2)/\rho_1}=\frac{1+\gamma\phi_1}{1+\phi_2}\rho_1$$

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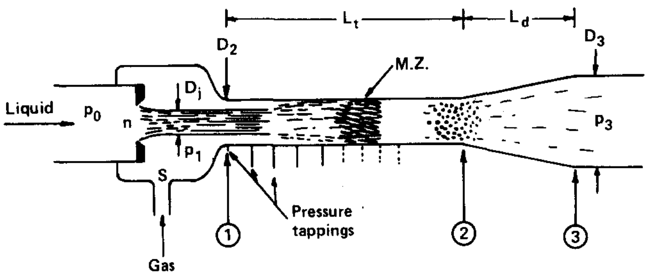

Figure 1 Notation diagram

so $\rho_1/\rho_2 \simeq 1+\phi_2$ since $\gamma\phi_1$ is of order 10^{-3}. We thus have a simple expression for the pressure recovery coefficient at station 2:

$$C_{p_2} = 2b[1 - b(1+\phi_2)(1+\tfrac{1}{2}K_t)] \tag{1}$$

Derivation of performance equations

The operation of the liquid/gas jet pump can be summarized in conjunction with Figure 1. A liquid jet is produced in the nozzle n by an upstream pressure p_0 and is fired along the axis of a mixing tube or throat. Shortly after the jet has left the nozzle, instabilities develop in its surface, leading eventually to breakup. The droplets formed induce a velocity in the surrounding gas, which has entered via the suction port s, and creates a partial vacuum p_1. At this mixing stage, the gas is considered to be the continuous medium, and the liquid droplets are dispersed within it. At a subsequent downstream location, the jet has completely reformed into a slower-moving mass of liquid and gas, and here the liquid is considered to be the continuous phase with gas bubbles interspersed. The liquid has therefore been decelerated with a concomitant increase in static pressure, whereas the gas has been accelerated by induction and then subsequently compressed.

A further deceleration is achieved using a diffuser to prepare the flow for its final pipe diameter and efficient diffusion will result in a further pressure increase here. However, conventional diffusers suffer a marked drop in efficiency when dealing with nonuniform inlet flows, so the matter of whether the major gas compression will occur in the mixing tube or in the diffuser will depend on the position of the mixing zone (MZ in Figure 1). In the case of low back pressure p_3 and a short mixing tube, the liquid jet may not have broken up before entering the diffuser, so the jet pump then operates at low efficiency, even though it may be drawing in large quantities of gas. Even if the mixing zone is upstream of the diffuser entry, the diffuser is still having to deal with a very frothy fluid so that efficiency may not be as high as one might expect in a single phase case.

Referring still to Figure 1, the flow process can be analyzed using a straightforward continuity, momentum, and energy approach. Losses, from whatever source, are accounted for by loss coefficients K, which can in general be found only by experiment. The pressure increase in the mixing tube can be handled by equating the net pressure force to the change of fluid momentum, a loss coefficient K_t being introduced to allow for frictional and sudden expansion losses.

$$(p_2 - p_1)A_2 = m_1 V_1 + m_g V_g - m_2 - A_2 K_t \rho_2 V_2^2/2$$

Conditions in the diffuser can be analyzed in the same way as for a single-phase case where an overall loss coefficient K_d accounts for all losses in total head between entry and exit and space-averaged quantities are used for mixing velocities, etc., so

$$p_2 + \tfrac{1}{2}\rho_2 V_2^2 - K_d(\tfrac{1}{2}\rho_2 V_2^2) = p_3 + \tfrac{1}{2}\rho_3 V_3^2$$

and therefore

$$\frac{p_3 - p_2}{\frac{1}{2}\rho_2 V_2^2} = 1 - K_d - \frac{\rho_3}{\rho_2}\frac{V_3^2}{V_2^2}$$

$$= 1 - K_d - \frac{1}{a^2}\frac{\rho_2}{\rho_3}$$

This could be transformed to a straightforward pressure recovery coefficient C_{p_3} if the denominator were changed to Z, so now we have

$$\frac{p_3 - p_2}{Z} = \frac{\rho_2 V_2^2}{\rho_1 V_1^2}\left[1 - K_d - \frac{1}{a^2}\frac{1+\phi_3}{1+\phi_2}\right]$$

We have already established that $V_2/V_1 \simeq b(\rho_1/\rho_2)$ and that $\rho_1/\rho_2 \simeq 1+\phi_2$ so the above expression becomes

$$\frac{p_3 - p_2}{Z} = b^2(1+\phi_2)\left[1 - K_d - \frac{1}{a^2}\frac{1+\phi_3}{1+\phi_2}\right]$$

The final term in brackets is very small compared to unity; in the experimental rig the value of $1/a^2$ was 0.0321 and $(1+\phi_3)/(1+\phi_2)$ will be only slightly less than 1 so this term is lumped in with K_d.

Notation

a	Area ratio A_3/A_2
A	Cross-sectional area
b	Area ratio A_j/A_2
c	Area ratio $(A_2 - A_j)/A_j$ $[=(1-b)/b]$
C_p	Pressure coefficients
D	Diameter
K	Loss coefficient $(=\Delta p/Z)$
L	Length
LR	Length-to-diameter ratio L_t/D_t
m	Mass flow rate
p	Static pressure
Q	Volumetric flow rate
Re	Jet Reynolds number $(=V_j D_j/\nu)$
V	Velocity
Z	Dynamic pressure of liquid jet
ρ	Density
γ	Density ratio ρ_g/ρ_l
ϕ	Gas:liquid ratio Q_g/Q_l
η	Efficiency $(=p_1\phi_1 \ln[p_3/p_1]/[p_0 - p_3])$

Subscripts

d	Diffuser
g	Gas phase
j	Jet
l	Liquid phase
n	Nozzle
0	Supply (stagnation value)
t	Throat (mixing tube)
1	Mixing tube entry section
2	Mixing tube exit section
3	Diffuser exit section

In addition, this simple analysis has ignored the work done in compressing the gas with increasing downstream pressure, a term that Cunningham[6] claims, quite rightly, is not negligible. Once again, this term is lumped with K_d. Since K_d is being determined experimentally, using equations such as the one above, it will not matter that it cannot be compared directly with Cunningham's value.

A pressure recovery coefficient C_{p_3} can now be set down for the whole compression process from vacuum chamber to diffuser exit.

$$C_{p_3}=\frac{p_3-p_1}{Z}=\frac{p_3-p_2}{Z}+\frac{p_2-p_1}{Z}$$
$$=2b[1-b(1+\phi_2)(1+\tfrac{1}{2}K_t)]+b^2(1+\phi_2)(1-K_d)$$
$$=2b[b^2(1+\phi_2)(1+K_t+K_d)] \qquad (2)$$

Formation of a computer program

The simple approach adopted in the previous section has produced equations that can be easily incorporated in a computer program for use by designers of jet pumps of the two-phase variety. The problems facing such designers fall into two broad types:

(a) It is required to know what dimensions should be given to a pump to satisfy a stated duty. The specification would normally be in terms of inducing a required gas flow against an imposed back pressure.
(b) The likely performance of a given jet pump of known dimensions is required to be predicted, for a given back pressure.

Case (b) is undoubtedly easier to handle, but the computer program must be able to predict results in both cases.

Case (a)

If jet pump dimensions are required for a specified duty, the input variables will be p_1, p_3, and Q_g. In addition, a designer may need to specify maximum tube length, since long jet pumps are sometimes unacceptable, and output pipe diameter, so that an upper limit can be set on mixing tube diameter. The gas constant R and temperature will also need to be specified so that gas density can be determined.

The calculation process must necessarily be an iterative one since the ability of the pump to induce a gas flow depends crucially on the position of the mixing zone in the tube, and this in turn depends on the relative magnitudes of p_0, p_1, and p_3. The pressure in the vacuum chamber can be given a typical value, but it tends to depend on gas availability. For example, if gas is being drawn from a gas bottle, a very severe vacuum can be experienced in the latter stages of draining the bottle.

The iteration could be started by assuming a high value of C_{p_3}, which in turn would imply low values of Z and p_0. Gas flow under these conditions would be very low, so ϕ_2 could be assumed zero in order to obtain a starting value of C_{p_2} from Equation 1. K_t can be found only by experiment, and a relationship is still needed between ϕ_1, p_0, and p_3 so that Q_l (and hence D_j and D_t) can be determined. As a result, a slightly more realistic (lower) value of C_{p_3} is obtained, and the process can then be repeated, using a nonzero ϕ_2 value, until realistic values of all the desired parameters are obtained. Experience shows that eight iterations are always sufficient.

Case (b)

In the case where the dimensions of an existing jet pump are already known, the input quantities to the program will be Q_g, p_3, D_n, D_t, L_t, D_3, R, and T. A calculation is started with a very high liquid flow rate, to allow for low-back-pressure cases. This produces a low value of calculated C_{p_3} because Z is artificially high. The link between ϕ and p_3 and p_0 is not required here because an assumed value of Q_l automatically implies a value of ϕ $(=Q_g/Q_l)$. The net result is calculated values of p_2 and p_3, and the liquid flow rate is then reduced in steps until the calculated value of p_3 equals the value set at the start and the iteration is stopped.

It is clear therefore that certain relationships between parameters are required in the constructions of this computer program, and these will in general be obtainable only from experimental test rig results. In particular, we need to know how ϕ_1 varies with p_3/p_0 for Case (a), and for both cases we need to find a relationship between the loss coefficients K_t and K_d and the variables upon which they might depend: jet-to-mixing-tube-area ratio b, mixing-tube-length-to-diameter ratio LR, and pressure recovery coefficient C_{p_3}, since these determine where the mixing zone shall be.

Experimental arrangements

A water/air test rig was constructed to study the operation of a two-phase jet pump and to provide empirical data for the relationship required in the computer model. The closed-circuit rig is shown in Figure 2.

The main body of the jet pump was made from perspex and comprised an inlet section that held the nozzle for producing the primary jet, a vacuum chamber that allowed air to enter, seven different sections that could be assembled to produce various lengths of mixing tube, and a conical diffuser section of total included angle 6.8° and area ratio 5.6. The mixing tube diameter was 21.5 mm and its maximum length 500 mm, or about 23.3 diameters. All sections were fitted with static pressure tappings to enable axial pressure plots to be obtained, since the mixing zone was not always accurately determined by eye. The pressure tappings were connected to a bank of vertical mercury/water manometers 1.5 m high.

Atmospheric air flow into the vacuum chamber was metered using a calibrated rotameter and could be controlled when needed by valve G. A centrifugal pump powered the water flow, which was metered using a calibrated orifice plate prior to entering the jet pump nozzle. Flow control was by gate valve W, placed well upstream of the orifice plate and pressures p_0 and p_3 were indicated by Budenberg gauges as shown, although these pressures were in practice available more accurately by summing the manometer readings.

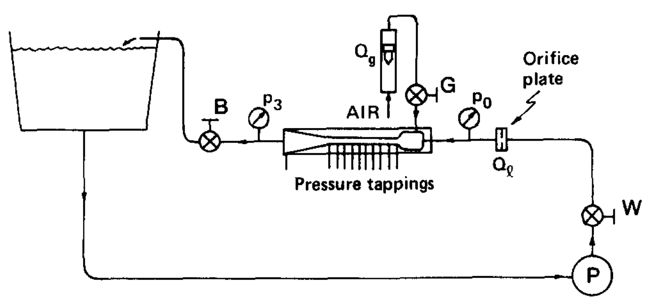

Figure 2 Experimental test rig (not to scale)

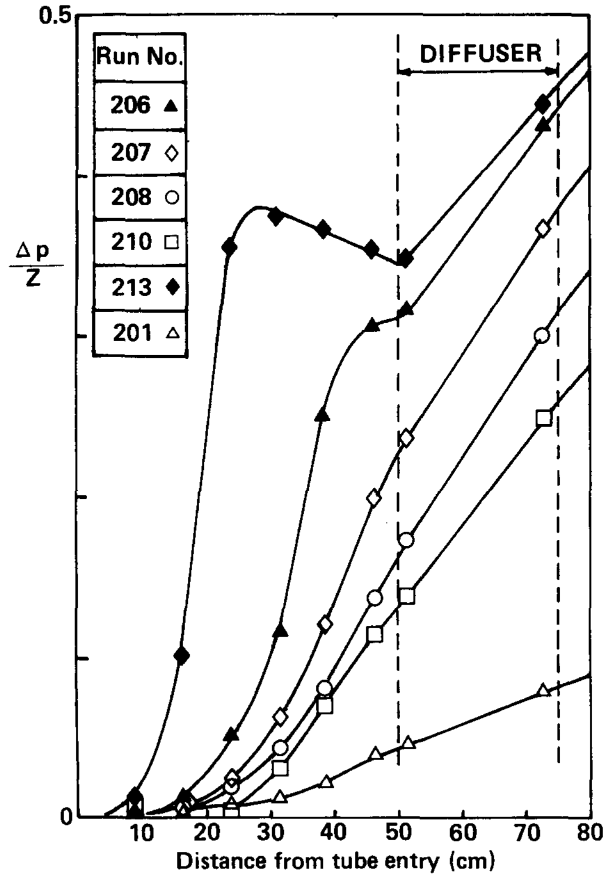

Figure 3 Effect of back pressure on pressure distribution: $LR=23.3$, $b=0.3$

Back pressure was imposed by partially closing valve B, although the rig was lightly pressurized even with B fully open by the head in the large downstream tank. The water level in this tank was above the manometer bank tops, to enable air to be expelled prior to each run.

Since b was known to be a variable of major importance, four different sharp-edged nozzles were available, three having single orifices to give jets of b value 0.3, 0.4, and 0.5, the fourth having four holes of equivalent (total) b value 0.3. Sharp-edged nozzles were used because they produce almost no friction losses, and their loss coefficients do not therefore need to be included in the overall analysis. In designing the nozzles, a contraction coefficient of 0.62 was assumed in each case.

The rig was a closed circuit, and the absence of a heat exchanger meant that the water temperature inevitably increased with time. The rate was found to be, on average, about 6°C/h, but this was considered to be acceptable since the average continuous run lasted about 40 min. The major parameter to be affected would be the Reynolds number, through the change in kinematic viscosity, but as the values of Re were always high and the mixing region flow always of the separated and highly turbulent variety, this was considered unimportant.

Testing was undertaken by measuring performance at all three b values for each of three mixing tube lengths. A computer program was available for rapid calculation of derived results, and loss coefficients K_t and K_d were evaluated from experimental values of C_{p_2} and C_{p_3} using Equations 1 and 2.

Discussion of experimental results

Nearly 300 experimental runs were carried out, and from the vast quantity of data available the results shown in this paper are those particularly applicable to understanding the modus operandi of the jet pump and the construction of a computer program (runs 201–287).

Pressure distribution in the mixing tube

Mixing tube axial pressure distribution has been plotted in Figures 3–5 in terms of static pressure difference from that of the first hole in the tube, normalized by division by the dynamic pressure Z in the water jet. This has been found to be a very useful and successful normalizing parameter with results for similar C_{p_3} conditions overlying each other despite differences of water flow rate of at least 4 to 1.

In all cases, there is a continuous transition from the rapid pressure rises in the mixing tube associated with high-back-pressure conditions to the much more placid rises with low back pressure. Figures 3–5 have been plotted with the same horizontal scales so that a realistic idea of pressure gradients is given.

Of particular interest in Figure 3 is the clear indication for run 213 of a straightforward frictional pressure loss in the mixing tube, after the initial pressure peak and before diffuser entry. With shorter mixing tubes, this region has no real chance to

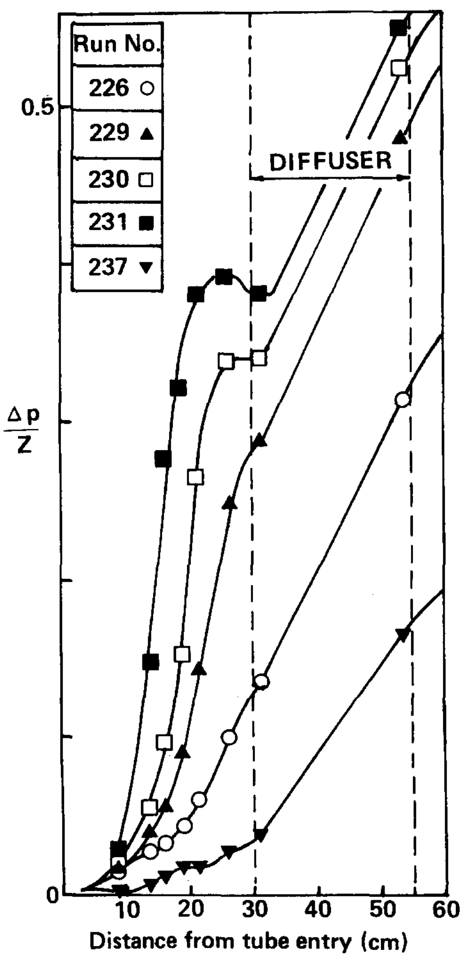

Figure 4 Effect of back pressure on pressure distribution: $LR=14$, $b=0.4$

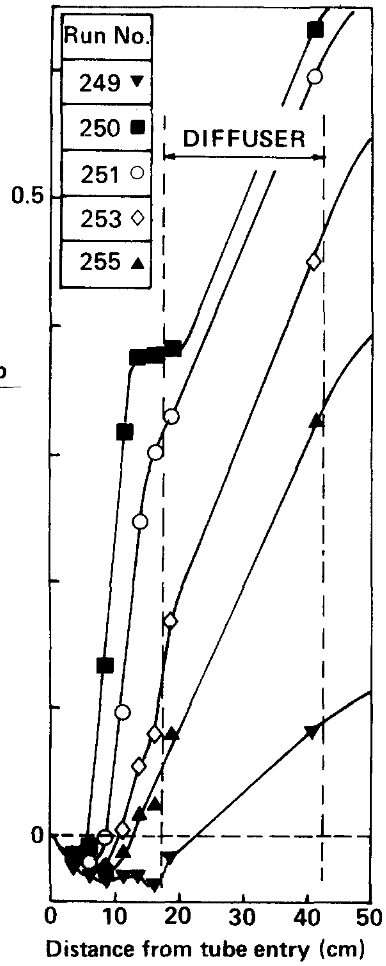

Figure 5 Effect of back pressure on pressure distribution: $LR=8.1$, $b=0.5$

develop. If the jet pump efficiency η of each run were to be inspected in conjunction with Figures 3–5, it would be clear that optimal operation always occurs at the highest back pressure not causing the frictional loss evident with run 213.

Although Figure 5 shows the rapid buildup of pressure inherent in using short mixing tubes at high p_3 values, it cannot show how sensitive the jet pump was under these conditions to a slight p_3 increase, causing a sudden flooding of the vacuum chamber and leading to obvious shutdown as a gas pump.

Pressure recovery versus gas ingestion

It will be recalled from comments in the theoretical performance equations section that a link was required between ϕ_1 and p_3/p_0; Figures 6–8 show this link for various nozzle sizes and mixing tube lengths. These three figures collectively confirm the trend, well known in jet pump circles for many years, that a high b value will enable high back pressures to be dealt with, but only at the expense of a lower maximum gas ingestion rate. For jet pumps in general, a b value of 0.4 would seem to have much to commend it, as a compromise.

The influence of tube length on performance is most evident for $b=0.3$ and virtually nonexistent at the other extreme of 0.5. This would be in keeping with the idea that a slimmer liquid jet in the longest possible tube has the greatest chance to break up and induce a gas flow over the longest possible distance. Figure 6 shows clearly that a tube length of 23.3 diameters produces the best performance in terms of the highest ϕ_1 value at a given back-pressure ratio.

Figure 6 also provides an opportunity to assess the value of the four-hole nozzle ($b=0.3$) assembly. This was tested at $LR=23.3$, but the results show no advantage over the single-hole nozzle at all; it might even be considered worse. The four-hole nozzle was therefore dropped from the testing schedule, and all future references to $b=0.3$ imply the single-hole case only.

Loss coefficients

The other important relationship needed for computer model construction was that between the loss coefficients K_t and K_d and mixing zone position as exemplified by C_{p_3} value. As an example of this, Figure 9 shows how K_d varies with C_{p_3} and b for

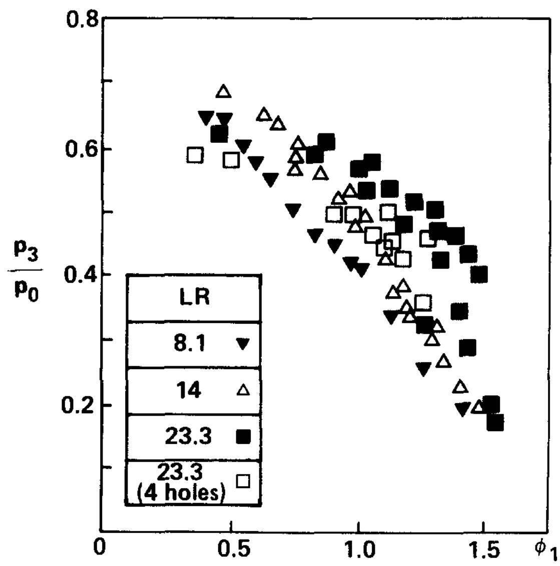

Figure 6 Effect of mixing tube length on gas ingestion: $b=0.3$

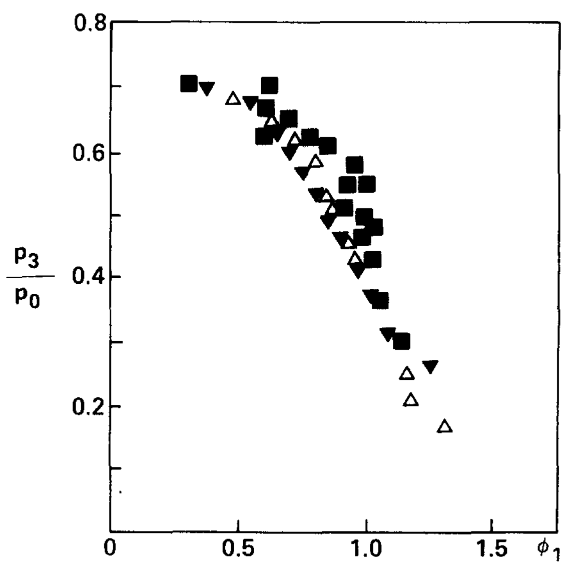

Figure 7 Effect of mixing tube length on gas ingestion: $b=0.4$; symbols as for Figure 6

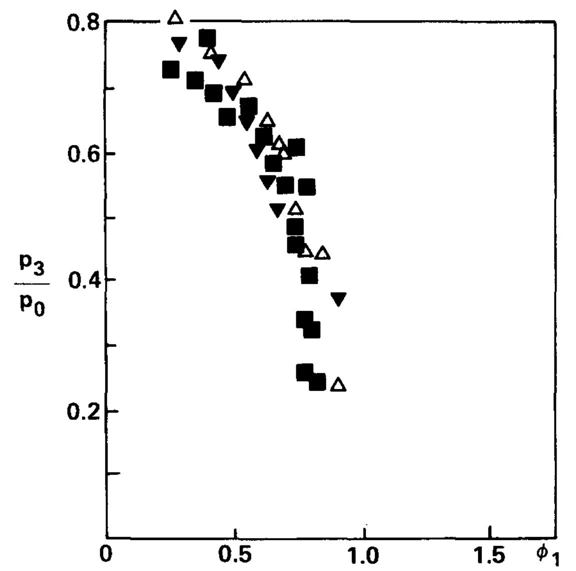

Figure 8 Effect of mixing tube length on gas ingestion: b=0.5; symbols as for Figure 6

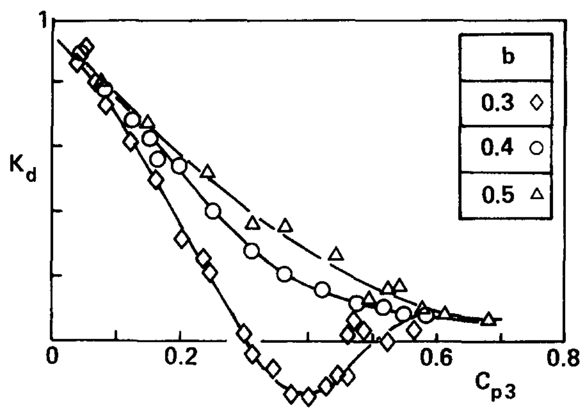

Figure 9 Effect of b value on diffuser loss coefficient: LR=14

the intermediate tube length of $LR=14$. At very low C_{p_3} values, corresponding to the mixing zone being in, or even downstream of, the diffuser, K_d values are close to unity, as expected, for what is essentially a sudden expansion type of loss. K_d falls progressively with increasing C_{p_3} and acquires small negative values in some cases with the smallest nozzle. These unrealistic values are probably the result of calculating them from Equation 2, which involves ϕ_2. This parameter, unlike ϕ_1, is not measured, and the assumption that $\phi_2=\phi_1 p_1/p_2$ may be slightly in error. This problem is also evident in Figure 10, which shows the influence of mixing tube length.

Figures 11 and 12 show the effects of b and L_t on mixing tube loss coefficient K_t. As stated before, one would expect sudden expansion losses to be greater than frictional ones, and these two figures show clearly that variation with b is more marked than with L_t, except that the shortest tube shows some interesting differences at intermediate pressure recovery coefficients.

Combined loss coefficient

One of the advantages of basing both K_t and K_d on the same normalizing quantity at station 2 is that they then appear in the same brackets in Equation 2, so a plot of K_t+K_d versus C_{p_3} is equally useful in constructing a computer model. The sum K $(=K_t+K_d)$ has been plotted in Figure 13, and it is pleasing to find that not only have the bulk of the negative K values disappeared but also that, in general, the results depend only on b and not on LR. Indeed, there is even a crossover point at about $C_{p_3}=0.35$ where they are not b dependent either. This is a major step forward in constructing a computer model, since the calculation of K from a C_{p_3} value will not need the prespecification of a mixing tube length.

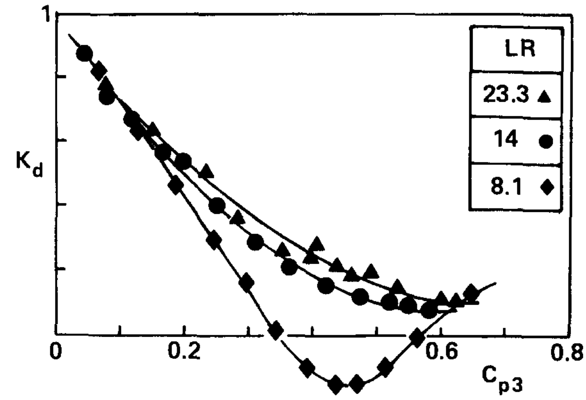

Figure 10 Effect of mixing tube length on diffuser loss coefficient: b=0.4

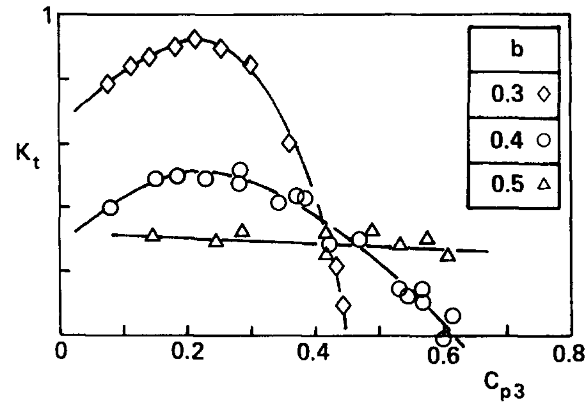

Figure 11 Effect of b value on mixing tube loss coefficient: LR=23.3

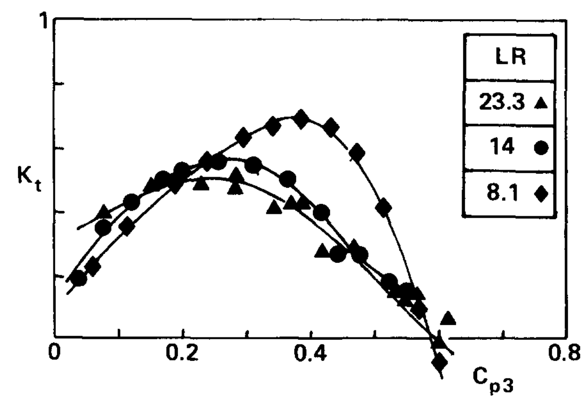

Figure 12 Effect of mixing tube length on loss coefficient: b=0.4

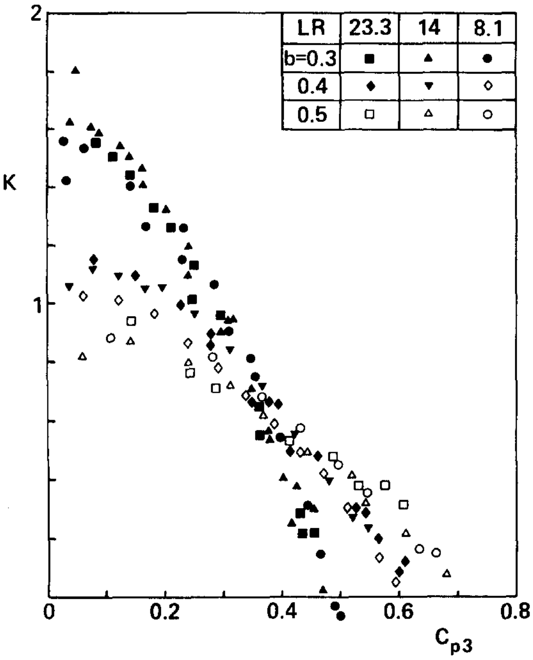

Figure 13 Overall loss coefficient K ($=K_t+K_d$)

Jet breakup length

As a check against other published work, Figure 14 shows a plot of nondimensional jet breakup length versus the value of $\phi_1/c\,\mathrm{Re}$, the parameters used by Cunningham and Dopkin[7]. The present results agree fairly closely with their data, represented by the shaded region. They used even longer mixing tubes and were able to plot results corresponding to both jet disintegration and water wall impact. The former, without exception, rose above the level represented by the equation

$$\frac{L_j}{cD_j}=7.86\times 10^6\,\frac{\phi_1}{c\,\mathrm{Re}} \qquad (3)$$

shown superimposed on Figure 14, the deviation starting at about $\phi_1/c\,\mathrm{Re}=1.8\times 10^{-6}$.

Incorporation into program P.21

A computer program called P.21 was developed using the empirical relationships obtainable from the graphical results, the digits 21 indicating the number of refinements needed during this process.

In Case (a), when only p_1, p_3, and Q_g are available and no dimensions, a link between ϕ_1, p_3/p_0 and b is required, and this can be derived from Figures 6–8. So far as mixing tube length is concerned, a choice of $LR=14$ could be made at this point to strike a balance between the danger of vacuum chamber flooding with shorter lengths and unwanted wall friction losses with longer ones.

A start is made by assuming a high C_{p_3} value (low Z) so that a required supply pressure p_0 results. The approximate relationship

$$\phi_1=3(0.85-p_3/p_0) \qquad (4)$$

from Figures 6–8 can now be used to obtain a first-stab value of ϕ_1. If ϕ_1 on this basis is above about 1.1, Figure 6 clearly suggests that a b value of 0.3 should be used, so this conditional statement is built into the program. For all other values of ϕ_1, the bulk of results presented here suggest that $b=0.4$ gives a useful compromise between high gas ingestion rates and high downstream-generated pressures. C_{p_2} can now be calculated from Equation 1, using a value of K_t from results such as Figures 11 and 12 and the appropriate LR and b values. A zero value is given to ϕ_2 for this initial step, and C_{p_3} is then obtainable from Equation 2 using K values from Figure 13.

The overall loss coefficient K is given with reasonable accuracy by

$$K=C_1+C_2C_{p_3}=2.5C_{p_3}^2 \qquad (5)$$

where

$$C_1=5.02-15.8b+15b^2 \quad \text{and} \quad C_2=-5.25+20b-17.95b^2$$

When a new (lower) C_{p_3} value has been calculated by this method, the process is repeated until changes from previous values are negligible. As stated before, eight iterations are sufficient.

In Case (b), where dimensions are known, and Q_g, p_1, and p_3 are specified, the initial link between ϕ_1 and p_3/p_0 is not required because the starting assumption of a suitable liquid flow will automatically define a starting value of ϕ_1. The value of K from Equation 5 can be used, as before, and a calculated value of p_3 is obtained. Q_l is then reduced in stages, and the iteration stops when calculated and imposed p_3 values are acceptably equal.

Typical run time for either Case (a) or Case (b) is about 10 to 12 s, although this is naturally reduced to under 2 s if a compiled version is used instead of the program in BASICA.

Comparison of predicted and experimental results

The use of Equations 2, 4, and 5 in a computer program clearly implies approximations, and one essential step in validating the program is to check whether, in its (b) mode, it will accurately predict the actual performance of the experimental rig for given p_1, p_3, and Q_g inputs. It is easy to modify the program to read experimental values of these three parameters from a data file, predict the performance of the jet pump and then compare the performance with the known values from the same data file. Two such outputs are shown in Figures 15 and 16 and in each case the disagreement between predicted and actual performance is seen to be no worse than the scatter involved in producing Equations 4 and 5 from experimental data.

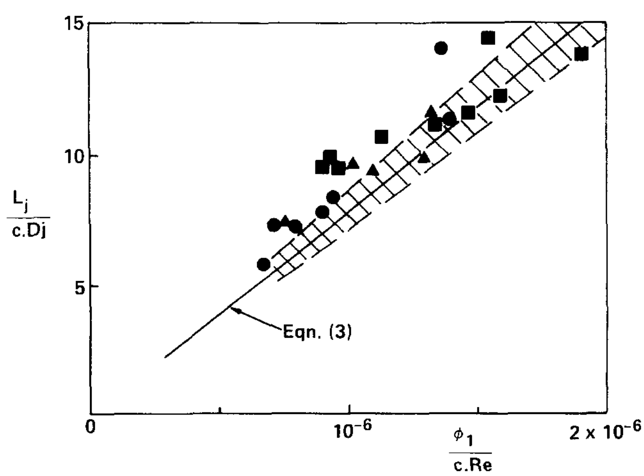

Figure 14 Liquid jet breakup lengths—comparison with Cunningham and Dopkin[7] (shaded area)

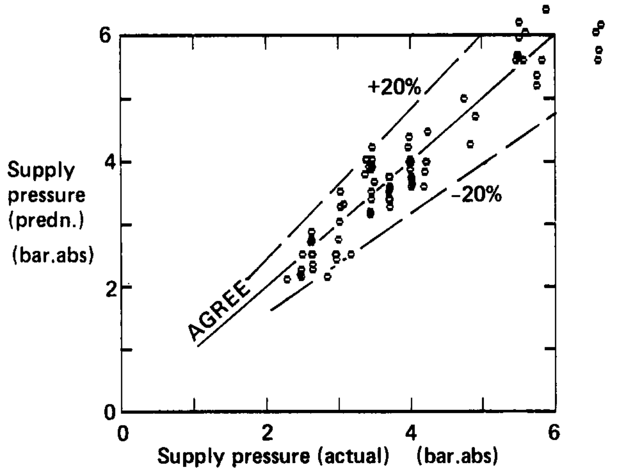

Figure 15 Accuracy of computer model predictions—supply pressure required

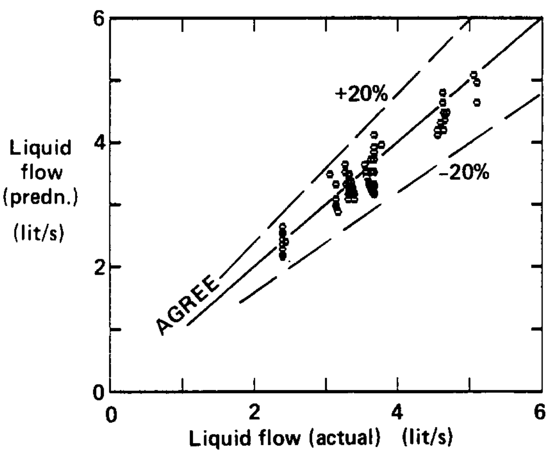

Figure 16 Accuracy of computer model predictions—liquid flow rate required

The program can also be checked in both (a) and (b) modes by using the former to generate a jet pump geometry from a given performance specification and then using that geometry in the (b) mode to confirm that the performance is the same as that forming the input to (a). Specimen numerical results are given in Table 1, and agreement is seen to be close.

Conclusions

This project had the dual goals of investigating the performance of a two-phase jet pump rig and then of assessing whether the relationships between geometrical and performance parameters were strong enough to warrant the development of a computer program for engineering design use.

Experimental results

By its very nature, the rapid induction and compression of a gas by a liquid jet is an unsteady process, and considerable scatter is therefore inherent in all the results presented here. The mixing zone was observed to be at least 30 mm long under most conditions and to move axially and periodically by about the same amount. Manometers, especially those containing mercury, have a fortunate propensity for damping such fluctuations, within themselves, and therefore firm conclusions can be justified.

The major factors affecting performance and the jet-to-mixing-tube-area ratio b, the relationship of back pressure p_3 to supply pressure p_0, and mixing-tube-length-to-diameter ratio LR.

(a) A b value of 0.4 represents a useful compromise between the ability of lower b value jets to induce high gas flow rates and that of higher b value jets to withstand higher back pressures.
(b) Under most circumstances (and certainly at high back pressures) a supply pressure of at least 2.5 times the back pressure will be required to avoid vacuum chamber flooding, absolute values being used in each case.
(c) A value of LR of between about 10 and 14 seems best; lower values involve the risk of vacuum chamber flooding at high back pressures, higher ones the complication of extra pipe friction losses in the mixing tube. It is also true, however, that frictional losses are usually small compared to those caused by sudden expansion and mixing processes.

Table 1 Numerical check of program P.21 (chlorine gas, R=117 J/kg K, T=18°C)

Input	Output
Mode (a)	
p_1 =40 mmHg vac	p_0 =5.20 bar abs
Q_g =4 L/s	Q_l =4.92 L/s
p_3 =2 bar g	D_t =23.2 mm
LR =14	D_n =18.62 mm (sharp-edged)
D_3 =60 mm	L_t =325 mm
	C_{p3} =0.478
	ϕ_1 =0.813
	m_g/m_l =2288 ppm
	power =2087 W
	η =40.7%
Mode (b)	
p_1 =40 mm Hg vac	p_0 =5.25 bar abs
Q_g =4 L/s	Q_l =4.942 L/s
p_3 =2 bar g	m_g/m_l =2278 ppm
D_n =18.62 mm (sharp-edged)	power =2117 W
D_t =23.2 mm	η =39.8%
L_t =325 mm	b =0.399
D_3 =60 mm	C_{p3} =0.475

Computer program

The various checks that have been applied to program P.21 in both its modes confirm that relationships derived from experimental tests can be used with confidence in building up a computer program for jet pump design, and a version of P.21 is already in use as a design tool in industry. Loss coefficients and other empirical links have enabled P.21 to predict performance to within about ±20% of true values, the majority of cases being within about ±10%.

The really major advance over the work of the most recent researchers in this field is that predictions can now confidently be made of LJGP performance under virtually any conditions. The previous restriction to cases where the mixing zone was in the mixing tube have been removed.

Scale effects

Although mathematical models are, by their very nature, intended to be amenable to dimension changes, it is probably unsafe to claim that program P.21 is applicable to any jet pump design, irrespective of size. Because of the highly turbulent flow state obtaining in the mixing tube, effects of viscosity and surface tension are likely to be unimportant compared with those applicable in two-phase flow. The dominant dimensionless term is likely to be the Froude number $\mathrm{Fr} = V^*/\sqrt{gD_t}$, where V^* is the superficial liquid velocity in the tube. This is a relatively weak inverse function of D_t, but since all experimental testing was carried out with $D_t = 21.5$ mm, it is probably unreliable to apply the computer model to tube diameters greater than about 120 mm.

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