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DIFFUSER PERFORMANCE IN TWO-PHASE JET PUMPS

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Abstract—Experimental results are reported from tests on a two-phase variable geometry jet pump with a view to assessing how well the included diffuser handles inlet flows with considerable non-uniformities of both velocity and density. Although the diffuser has an important contribution to make in such an application, results show that its pressure recovering abilities are markedly influenced by increasing non-uniformities at entry and by those potentially developing in the diffuser itself. In two-phase jet pumps, short mixing tubes and thin primary jets should generally be avoided.

Key Words: jet pumps, induction pumps, two-phase, diffusers, pressure recovery

1. INTRODUCTION

Single-phase jet (or induction) pumps have been used for many years with varying degrees of efficiency for compression or evacuation purposes. Over the last two decades, however, two-phase versions have been evolved using a liquid as the primary jet and a gas as the secondary flow. Early models were very inefficient as gas compressors but by the middle 1960s, Witte (1965) was reporting results typical of modern optimum values and further, more detailed, research was reported by Cunningham (1974) and Cunningham & Dopkin (1974).

Industrially sponsored work by Neve (1988), using variable geometry devices, showed the importance of the mixing tube and diffuser in achieving peak efficiencies and results were sufficiently repeatable for a mathematical model to be assembled. A computer program based on that model is now used successfully both for the performance prediction of given devices and for the *ab initio* design of two-phase jet pumps for a given task. Applications include the chlorination of water and the addition of gas to biochemical reactions.

Results so far have indicated that the diffuser fitted to the downstream end of the jet pump's mixing tube has a major influence on the operating efficiency, accounting for at least half the static pressure recovery in a typical device operating against useful back pressures and thus contributing at least half of the efficiency figure. Since diffuser efficiency is known to depend crucially on inlet conditions, especially the uniformity of the velocity profile, this paper reports the results from 86 experimental runs of a variable geometry jet pump, so that guidance can be given on how static pressure recovery in a diffuser varies over a wide range of two-phase inlet flows. The fluids used were water and air and the variables of major importance were found to be the homogeneous void fraction ε, the jet to mixing tube area ratio b and the mixing tube length to diameter ratio LR.

2. TWO-PHASE JET PUMPS

A diagram showing the typical operation of a two-phase jet pump is shown in figure 1. Liquid at supply pressure p_0 is forced through a nozzle to produce a jet coaxial with, but of smaller diameter than, a mixing tube. Surface instabilities cause jet breakup and the resulting droplets drag gas from the interaction chamber into the mixing tube, causing a partial vacuum. At some downstream mixing zone, labelled MZ in figure 1, the flow changes from liquid droplets in a gas phase to gas bubbles in a liquid phase but this zone is neither well-defined nor particularly stable. One is therefore dealing with time-averaged conditions from now on.

The ability of the liquid jet to induce a gas flow clearly depends on the ratio of the jet area to tube area and on the length to diameter ratio of the mixing tube itself. The diffuser then has the job of returning as much as possible of the dynamic pressure to static pressure in the two-phase mixture at station 2.

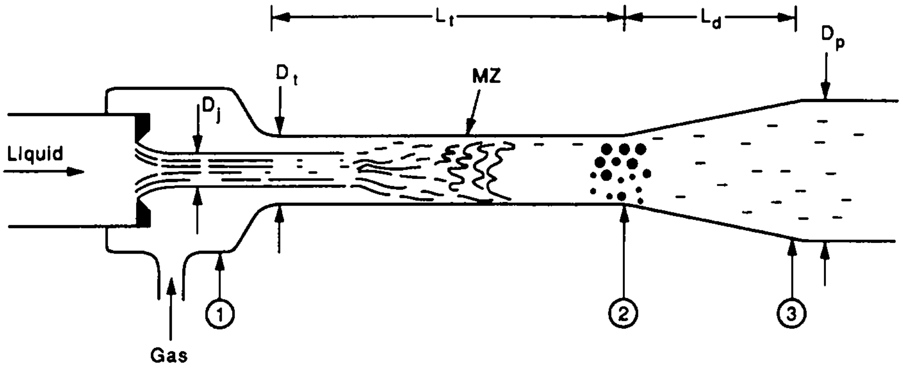

Figure 1. The two-phase jet pump (notation diagram).

Results given by Neve (1988) show that when b is low the pump can draw most gas but has a poor performance against higher back pressure p_3, whereas the reverse is true for high b values. A value of $b = 0.4$ seems a worthwhile compromise for most applications.

3. DIFFUSER PERFORMANCE

The vast majority of past research into diffusers has dealt with single-phase flows, the prime requirement being that pressure be recovered from a moving fluid as efficiently as possible. This conversion to static pressure is never achieved without loss of total (stagnation) pressure and the most realistic way of defining efficiency involves that loss. Unfortunately, such an estimation would require an extensive velocity and pressure traverse at the inlet and outlet and this has resulted in diffuser performance normally being assessed in terms of a pressure recovery coefficient C_p. This is a measure of the wall static pressure recovery, normalized by division by the inlet dynamic pressure and by an allowance $(1 - \sigma^2)$ for the area ratio $\sigma(=A_2/A_3)$, since complete diffusion could occur only if the area tended to infinity and therefore the exit velocity to zero.

E.S.D.U. (1976) have published C_p figures for many conical diffusers in terms of σ, the normalized diffuser length L_d/D_2 and conical angle and designers can confidently predict performance for given geometries provided the inlet flow is single phase and reasonably uniform. In particular, the best performance is achieved if an area ratio of about $\sigma = 0.2$ and a total cone angle of about 7° are used. However, C_p is not a very satisfactory measure to use because it is based on the space-mean velocity at entry $\bar{V}_2$. In a highly non-uniform velocity profile, the square of the maximum inlet velocity (which relates to the actual local static pressure p_2) can be very much greater than $\bar{V}_2^2$ so C_p can acquire misleadingly high values (far greater than unity) because p_2 and $\bar{V}_2$ can both be artificially low when used in the C_p definition. Such large non-uniformities are not always evened out by the time the flow reaches the diffuser exit section and considerable diffusion may remain to be done in the downstream pipe. When the pipe flow has become fully established again, the static pressure will fall linearly with downstream travel because of wall friction. Diffuser performance with non-uniform single-phase inflows has been dealt with by Tyler & Williamson (1967) in the case of purely axial flow and by Neve & Thakker (1981) for flows having rotational components.

In the case of two-phase diffuser flows, few results have been published for either conical or two-dimensional types. Thang & Davis (1979, 1981) were essentially concerned with gas/liquid flows in venturis, the conical diffusers thus being the downstream ends of their devices and Hench & Johnston (1972) have published results for similar flows in two-dimensional diffusers. However, each of these papers provides important comparative material for the current work.

4. EXPERIMENTAL ARRANGEMENTS

The experimental rig used in these diffuser tests has been fully described by Neve (1988) so a brief outline only is given here. The rig was of the closed circuit type with water as the primary fluid and air as the secondary. Various Perspex sections were clamped together by tie rods to give different *LR* values and three different nozzles gave *b* values of 0.3, 0.4 and 0.5. The mixing tube diameter was constant at 21.5 mm and if all tube sections were used at once, a maximum length of 500 mm was attainable ($LR = 23.3$).

Water flow was metered by a calibrated orifice plate; air flow by a rotameter, prior to entering the vacuum chamber. Static pressure was measured at many relevant points, the values at the diffuser inlet and outlet being of greatest interest here. Air volumetric flow rate at the diffuser inlet was calculated from the metered value, assuming it to be inversely proportional to the local static pressure; the homogeneous void fraction was based on that value. This is effectively an assumption of isothermal conditions and is in line with that made by Thang & Davis (1981), which was justified by their results.

The diffuser used in these tests was close to the "optimum" geometry with an area ratio A_3/A_2 of 5.6 ($\sigma = 0.18$) and a total conical angle of 6.8°. With single-phase uniform flow, this diffuser would be expected to have a C_p value around 0.8, according to the E.S.D.U. data sheets.

5. DISCUSSION OF RESULTS

Figures 2–5 show how the pressure recovery coefficient depends on ϵ, *b* and *LR* but it is important to suggest here two parameters of which pressure recovery is probably not a function, within the boundaries of these tests. E.S.D.U. (1976) suggests that the Reynolds number (Re) becomes an increasingly important parameter below about 36×10^3 because the inlet section boundary layers then become proportionally thick enough to constitute a non-uniformity. In the current tests, Re is based as usual for two-phase flows on the homogeneous mixture density, the sum of the superficial velocities for each phase, the inlet diameter and the liquid viscosity. On that basis, all tests were carried out in the range $1.42 \times 10^5 \leqslant \text{Re} \leqslant 3.02 \times 10^5$; this is considered to be well above any likely transitional regimes.

The Froude number (Fr) is also an important parameter is liquid/gas flows and this is usually defined in terms of the superficial liquid velocity and the tube diameter. In these tests, Froude numbers were in the range $14.4 \leqslant \text{Fr} \leqslant 41.7$; once again, well above any transitional difficulties.

The greatest uniformity of inlet flow is obtained when the largest jet pump nozzle ($b = 0.5$) and the longest mixing tube ($LR = 23.3$) are used, since a minimum area of annular mixing and a maximum time for doing so are then involved. Figure 2 shows that the mixing tube length has virtually no noticeable effect over the range $8.1 \leqslant LR \leqslant 23.3$, but that C_p suffers considerably as the homogeneous void fraction is increased. No data points have been plotted for $\epsilon = 0$ (liquid only) since they would be unrealistic. The flow would have undergone a severe separation and mixing process in the vacuum chamber with no air being admitted; conditions at the diffuser entrance could therefore hardly be described as uniform. The trend of points in figure 2 is however towards an $\epsilon = 0$ figure similar to the E.S.D.U. prediction of $C_p = 0.8$.

The appearance of figure 2 is very similar to the plots of C_p vs gas volume flow fraction given by Hench & Johnston (1972), even though they were dealing with vertically upward flow in shorter, two-dimensional diffusers. Their appendix gives an analysis predicting the amount by which C_p will suffer with increasing void fraction and they make the point that for churn–turbulent conditions, the flow behaves very much as if it were homogeneous. The flows contributing to figure 2 had a very high turbulence intensity, largely because of their history prior to entering the diffuser and this made it virtually impossible to relate the form of the flow, as seen by the human eye, to the value of C_p being achieved. The internal flow structure was completely obliterated by the flow close to the walls so visual study was pointless. This difficulty also afflicted Thang & Davis (1979), who resorted to immersed probes to measure bubble size in their venturi diffusers.

The density used in defining C_p in these results is the homogeneous one, $\rho = [\epsilon\rho_G + (1 - \epsilon)\rho_L]$. This is justified on the grounds that all the conventional definitions give roughly the same figure at the homogeneous void fractions used here. Only as ϵ approached 0.9 are any real deviations evident. In assessing his own results for sudden enlargement type diffusers, Wadle (1989) reviews the various density definitions and even suggests a new one to give a better correlation for his results. At homogeneous void fractions below about 0.8, Wadle's mixture density is close to that of the liquid and would have the effect of lowering all the data points in figure 2.

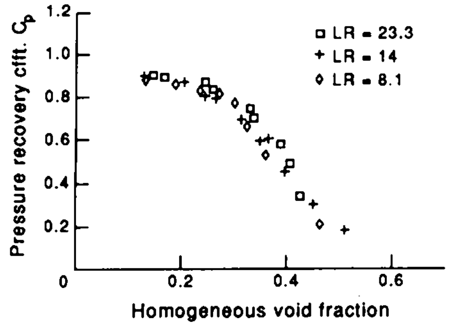

Figure 2. Effect of mixing tube length on pressure recovery ($b = 0.5$).

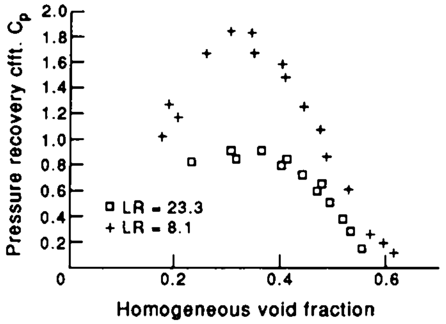

Figure 3. Effect of mixing tube length on pressure recovery ($b = 0.3$).

The decrease of C_p with ϵ in this figure suggests that either yet another density definition is needed or that some other phenomenon is becoming evident, since any data points which could be obtained at the extreme right-hand end of the figure ($\epsilon = 1.0$) would presumably return to $C_p \simeq 0.8$ with gas-only operation. The E.S.D.U. predictions are not fluid-specific, provided Reynolds number limitations are not contravened.

This author suggests that the fall-off is caused by non-uniformity of the density in the diffuser itself. In any flow separation caused by directional change, with concomitant streamline curvature, the gas phase will tend towards the centre of curvature, the liquid phase away from it, purely from centrifugation. If, therefore, there is a tendency for the mixture to separate from the diffuser walls, the liquid will tend towards the axis, the gas towards the walls, giving a highly non-uniform density profile. The effective density therefore tends towards the liquid value and C_p falls because of this non-uniformity.

Similar trends are evident in results reported by Tapucu *et al.* (1989) in their investigations of pressure losses due to rapid area changes in two-phase flows. The parameter they plot is the conventional loss coefficient (pressure drop divided by inlet dynamic pressure) and values far in excess of single-phase ones are achieved as the void fraction is increased from 0 to 0.6. The diffuser results presented in the present paper show the same trend since decreasing C_p values imply increasing loss coefficients.

Further evidence of density non-uniformity is given by Thang & Davis (1979), where plots are shown of void fraction vs position in the flow at the inlet to, and the outlet from, their venturi diffusers. In most cases, although the void fraction at the inlet is either roughly uniform or slightly peaked towards the centreline, in every case of outlet flow a double-hump profile for ϵ is obtained, indicating a movement of gas towards the walls.

Although the effect of inlet tube length may be minimal when the widest liquid jet is used, it is certainly not so with the slimmest jet ($b = 0.3$), as figure 3 shows. The data points for LR = 23.3 are similar to those in figure 2 but the shortest tube has produced C_p values well above unity, indicating presumably that the inlet flow is still in its infancy of development. Such high C_p values were mentioned as a possibility in section 3 and were also encountered by Wadle (1989).

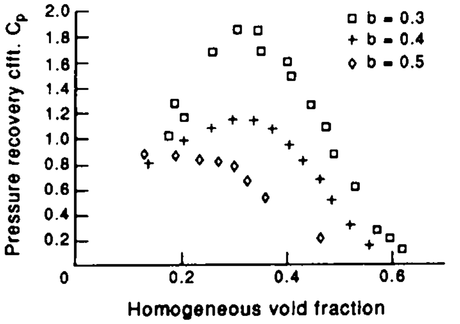

Figure 4. Effect of jet size on pressure recovery (short tube: $LR = 8.1$).

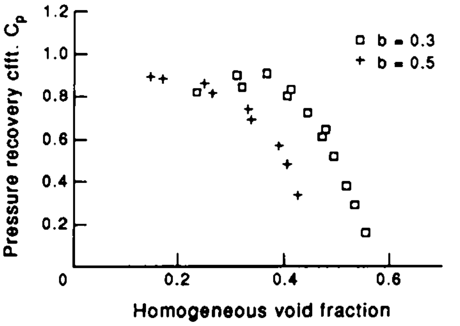

Figure 5. Effect of jet size on pressure recovery (long tube: $LR = 23.3$).

To complete the picture, figure 4 shows the influence of liquid jet width with the shortest mixing tube and figure 5 that for the longest tube. Clearly, jet size is much more important with the shorter tube, the longer one ironing out many of the non-uniformities.

6. ACCURACY

It is difficult to give realistic accuracy figures in experimental tests where two fluid phases are interacting in a far from stable manner. All plots are in the form of C_p vs ϵ. In the former case, the measurement of pressure differences is helped by the fortunate propensity of manometers for damping out high-frequency pressure oscillations. In the case of the void fraction at the diffuser inlet, any major inaccuracy stems from the assumption that the air volumetric flow rate (Q_G) is inversely proportional to the local absolute static pressure, the initial value of Q_G being more accurately metered. From the scatter evident in figures 2–5, it is probably fairest to suggest that the experimental accuracy is no better than about $\pm 5\%$.

7. CONCLUSIONS

The performance of diffuser/mixing tube combinations in two-phase flows is more difficult to understand than for the single-phase case. Problems associated with the non-uniformity of density are added to those associated with velocity non-uniformities in single-phase flows.

If a value of $b = 0.5$ is used in a jet pump, a mixing tube as short as $LR = 8$ may be used with some confidence that the two phases will have mixed sufficiently before diffuser entry. If a value as low as $b = 0.3$ is used (to gain a higher gas ingestion rate, for example) a mixing tube as long as $LR = 23$ is needed, if severe non-uniformities are to be avoided.

Even with the most favourable combinations of geometry, the pressure recovery performance of the diffuser suffers with increasing homogeneous void fraction. At ϵ values around 0.3, thin water jets and short mixing tubes cause high C_p values for the reasons given in section 3, whereas at ϵ values around 0.6 the diffuser performance is uniformly poor, irrespective of the b and LR values involved. Ironically, high C_p values do not necessarily mean good pressure recovery since p_2 may be depressed rather than p_3 being elevated.

The suggestion is made that for thin jets and short tubes high C_p values result from the considerable non-uniformity of velocity at the diffuser inlet associated with the mixing zone having passed into the diffuser, whereas the low C_p values recorded at higher void fractions result from massive non-uniformities of density throughout the whole diffuser and its tail pipe.

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