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Technical Notes

TECHNICAL NOTES are short manuscripts describing new developments or important results of a preliminary nature. These Notes should not exceed 2500 words (where a figure or table counts as 200 words). Following informal review by the Editors, they may be published within a few months of the date of receipt. Style requirements are the same as for regular contributions (see inside back cover).

Numerical Study of Flow Inside an Annular Jet Pump

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Nomenclature

A_J	=	sectional area of annular jet
A_o, D_o	=	area, inner diameter of suction and delivery lines
A_{so}, D_{so}	=	area, exit diameter of central suction nozzle
A_t, D_t	=	area, inner diameter of mixing chamber
C_p	=	pressure coefficient $= [(P_x - P_s)]/[0.5\rho V_J^2]$
L_t	=	length of mixing chamber
L'	=	total length of the convergent nozzle and the mixing chamber
M	=	mass flow ratio $= [\rho_s Q_s/\rho_J Q_J]$
N	=	head ratio $= [(P_d - P_s)/(P_J - P_d)]$
P_j, P_s, P_d	=	wall pressure (jet nozzle, suction pipe, delivery pipe)
P_x	=	total pressure at location (X)
Q, Q	=	flow rate (jet, suction)
R	=	area ratio of jet and mixing chamber $= [A_J/A_t]$
r	=	distance from centerline along the radius
V_J, V_t	=	mean axial velocity of the jet nozzle
x	=	axial position from the jet exit, mixing chamber
η	=	efficiency $= [M \cdot N]$
Φ	=	angle of convergent nozzle

Subscripts

d	=	delivery
j	=	annular jet
t	=	mixing chamber

Introduction

THE jet pump is a device that performs its pumping action by the transfer of energy from a high-velocity jet to a low-velocity stream.

In general, there are two types of such pumps: a jet pump with the primary jet nozzle placed along the centerline of the pump and the second stream around it (the central type), and a pump with the suction stream in the center and the annular driving jet on the outside (the annular jet type). Configuration and dimensions of the annular jet pump are shown in Fig. 1.

The primary power is provided by a high-pressure stream of fluid directed through the nozzle to produce the highest possible velocity. Due to the high velocity of the primary fluid, low pressure in the suction chamber creates suction of the secondary stream. Consequently, turbulent mixing between the two streams (with two different velocities) occurs, with increased pressure due to change of momentum; then the pressure of the combined flow is increased further in the outlet diffuser. Even through their efficiency is low, such pumps are widely used because of their simplicity and high reliability (as a consequence of having no moving parts). The jet pump has a very wide range of applications, such as in deep well pumping, booster pumping , dredging, priming devices, and slurry pumping.

James Thomson made the first known application of the water-jet pump in 1852. He designed a pump for the specific purpose of removing water from the pits of submerged water wheels. The theory of the mixing of two streams of a liquid was developed by Rankine[1] in 1870. He based his study on the one–dimensional continuity and momentum equations, which are still used in the present day.

However, jet pumps received little attention prior to 1930, because the maximum efficiency level was about 24%. In the early 1930s Gosline and O'Brien[2] proposed a theory of the jet pump and obtained an efficiency on the order of 31%. The efficiency of the jet pump is defined as the ratio of energy transferred to the secondary stream to energy transferred from the primary stream, that is,

$$\eta = [(Q_s/Q_j)] \cdot [(P_d - P_s)/(P_j - P_d)] = M \cdot N$$

Reddy and Kar[3] carried out a theoretical and experimental study. The experimental work in their research was divided into two parts; tests with the components, and tests with an assembled jet pump. The experiments were carried out for a wide range of different dimensions of the components and configurations of the assembly of the jet pump. In the theoretical work, he deduced an equation for theoretical efficiency,

$$\eta = 2[M/(1 + M)^2]$$

For the annular jet-pump type the experimental information is limited because only a few studies have been carried out, such as Shimizu et al.[4] and Yokota et al.[5] The experiments of Yokota cover a wide range of flow ratios, but show low efficiency, about 14%.

Shimizu et al.[4] investigated experimentally the relation between configuration and efficiency of annular-type jet pumps and compared it with that for central-type jet pumps. Twenty-five different kinds of pumps were used in the experiments. Those pumps reached a maximum efficiency of 36% (within uncertainty percentage ±05). This corresponds with that of the conventional central-type jet pump. The present work includes comparison of the obtained numerical results and experimental results of Shimizu.[4]

Numerical Technique

The numerical computational was carried out using CFX-TASC flow.[6] This code solves Reynolds averaged Navier–Stokes equations

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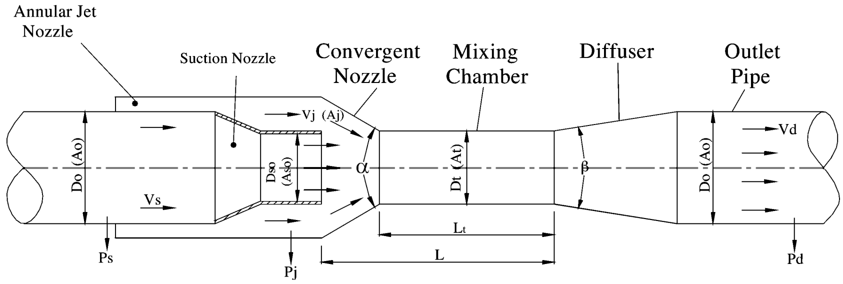

Fig. 1 Configuration and dimensions of annular jet pump.

in primitive variable form. The standard K–ε turbulence model is used to study the effect of turbulence of fluid flow. Also, in order to study the flow near the wall, a wall function is used to resolve the wall flows. This code uses a finite-element-based finite-volume method. Finite-difference, finite-element, and finite-volume methods can all be shown to belong to a broader class called the method of weighted residuals, as presented in Ref. 7. To achieve a numerical solution that is convergent, consistent, and stable, various numerical schemes have been implemented in CFX-TASC flow code.

Boundary Conditions

To get an accurate simulation, the same dimensions of the jet pump used by Shimizu[8] were adapted. Also, the boundary conditions of fluid flow were the same as those set in the experimental results. The mass flow rate of each primary and secondary that gives the correct mass-flow ratio and the outlet static pressure were applied in the code. The turbulent intensity and the eddy length were 0.03 and 0.03, respectively, whereas the average Reynolds number was 7.0×10^5. The maximum residuals were less than 10^{-4}. To reach good convergences the number of iterations was 120 for a time of calculation of about 5.5 h, for each condition of the fluid flow.

Grid Generation

A high-quality mesh is produced using a multiblock [H] grid inside the jet pump. This type of grid, for this problem, gives a better minimum skew angle, which should not be less than 20 deg, and a better maximum aspect ratio, which should not be more than 100. The total number of grid nodes is about 430,000.

Results and Discussion

Characteristic Curves

The main characteristic curves showing the performance of a jet pump are the M–N and M–η curves. The optimum efficiency in the experimental results of Shimizu et al.[4] is at $A_t/A_o = 0.48$, $A_{so}/A_o = 0.61$, $R = 0.57$. Thus the prediction of the characteristic curves is carried out with these ratios. The main variables are the ratio of mixing chamber length to inner diameter of suction line (L_t/D_o) and convergent nozzle angle (Φ). Figure 2.a shows the relation between head ratio (N) and mass flow ratio (M) at constant value of $(L_t/D_o = 1.86)$ and for different convergent nozzle angles $(\Phi = 18, 30)$, while corresponding M–η curves are illustrated in Fig. 2.b.

Static Wall-Pressure Distribution in the Flow Direction

The static wall-pressure distribution inside the convergent nozzle, the mixing chamber, and the diffuser is a good pointer to know what occurs inside the jet pump. Figure 3, shows the pressure coefficient C_p versus dimensionless axial position (X/D_o) from the jet exit. These results are a sample of the experimental results for various values of the convergent angle $\Phi = 18$ deg, $A_t/A_o = 0.48$, and $L'/D_o = 4.24$.

Figure 3, shows the static wall pressure coefficient for an area ratio $R = 0.57$ and $A_{so}/A_o = 0.61$ for mass flow ratios $M = 0.3$, 0.58. In every case, the static pressure distribution in the mixing chamber drops at $(X/D_o) = 0$ to 3.8 and then recovers in the diffuser $(X/D_o) = 3.8$ to 6.9 until it becomes almost constant.

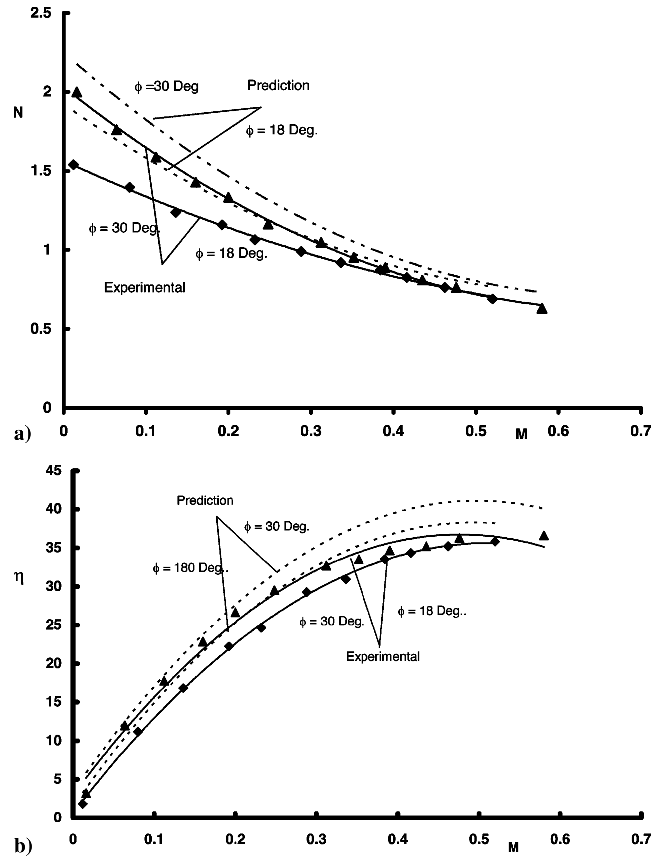

Fig. 2 a) M–N and b) M–η characteristic curves for L_t/D_o 1.86 and two values of Φ.

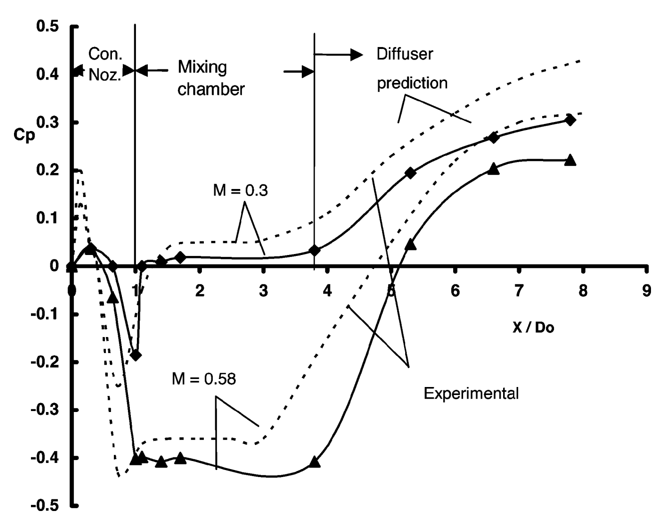

Fig. 3 Wall static pressure distribution along the pump, for $R = 0.57$.

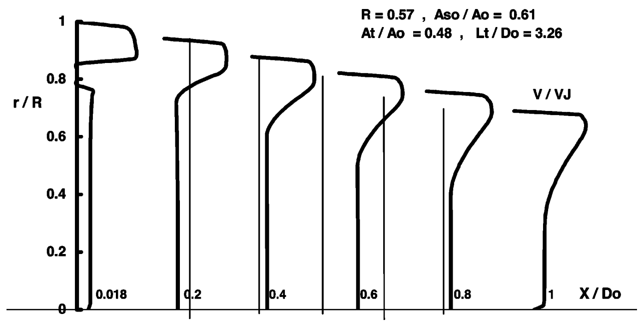

Fig. 4 Dimensionless velocity distribution along the convergent nozzle, and $M = 0.58$.

Velocity Distribution

The prediction of velocity distribution inside the convergent nozzle is carried out at $R = 0.57$, $A_{so}/A_o = 0.61$, $A_t/A_o = 0.48$, $L'/D_o = 4.24$, $\varphi = 18$ deg, and mass flow ratio 0.58. Figure 4 shows the prediction of the dimensionless velocity distributions at six sections from $X/D_o = 0.018$ to 1 along the axis of the convergent nozzle. The velocity distributions are referred to the velocity at the exit of the annular jet nozzle (V/V_J).

Conclusions

In this study the simulation of the flow inside annular jet pumps is carried out using CFX-TASC flow code. The experimental data were obtained from the extensive results obtained by Shimizu.[4] Comparison between the experimental results and prediction results is carried out, for the characteristics and performance curves of jet pumps at $A_t/A_o = 0.48$, $A_{so}/A_o = 0.61$, $R = 0.57$ for two convergent nozzle angles $\Phi = 18$ deg, 30 deg, and for two configurations $L_t/D_o = 18$, 3.26. The static wall pressure distribution in pressure coefficient form (C_p) is calculated for two area ratios ($R = 0.57, 0.82$) for mass flow rates $M = 0.19$–0.58. Good agreement between them is found. Further, the dimensionless velocity distribution at various sections is carried out for $R = 0.57$ ($L_t/D_o = 3.26$) along the axis. The pressure contours are obtained for the same configuration and condition of flow. Finally, the turbulent kinetic energy and the dissipation of turbulent kinetic energy are calculated.

The numerical results illustrate the mechanism of the mixing process inside both the convergent nozzle and the mixing chamber. The comparison proves the validity of the used code, when properly applied, and opens the possibility of extensive study of the effect of the various kinematic and geometric ratios of the pump, without the need for resort to more expensive and time-consuming experimental research.

References

[1]Rankine, J. M., Proceedings, Royal Society of London, No. 123, 1870, pp. 80–83.

[2]Gosline, J. E., and O'Brien, M. P., "The Water Jet Pump," *University of California Publications in Engineering*, Vol. 3, No. 3, 1934, pp. 167–190.

[3]Reddy, Y. R., and Kar, S., "Theory and Performance of Water Jet Pump," *Journal of the Hydraulics Division*, Vol. HY5, Sept. 1968, pp. 1261–1281.

[4]Shimizu, Y., Nakamura, S., Kuzuhara, S., and Kurata, S., "Studies of the Configuration and Performance of Annular Jet Type pumps," *Journal of Fluids Engineering*, Vol. 109, Sept. 1987, pp. 205–212.

[5]Yokota, A., "On the Annular Ring Type Jet Pump," Japan Miner. Engineers Society, No. 90-1038, 1974, p. 539.

[6]ASC, "CFX-TASCflow Documentation Version 2.9.0," Advanced Scientific Computing, Waterloo, Ontario, 1999.

[7]Raithby, G. D., and Galpin, P. F., "The Practical Application of Computational Fluid Mechanics," Conference Paper, Canadian Congress of Applied Mechanics, June 1991.