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Two-dimensional Numerical Simulations on the Performance of an Annular Jet Pump

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Abstract: Two-dimensional numerical simulations were carried out to investigate the effects of the shape of mixing chamber on the performance of annular jet pumps for several jet flow rates. We confined the computational domain to a jet, a mixing chamber, and the outlet of the pump. Several computations have been carried out to seek the effects of reducing angles on the suction performance and the efficiency of the jet pump. Some numerical results regarding suction flow rates, head ratios, the efficiencies of the pump, flow fields and pressure distributions are shown and discussed.

Keywords: annular jet pump, computational fluid dynamics, mixing chamber.

1. Introduction

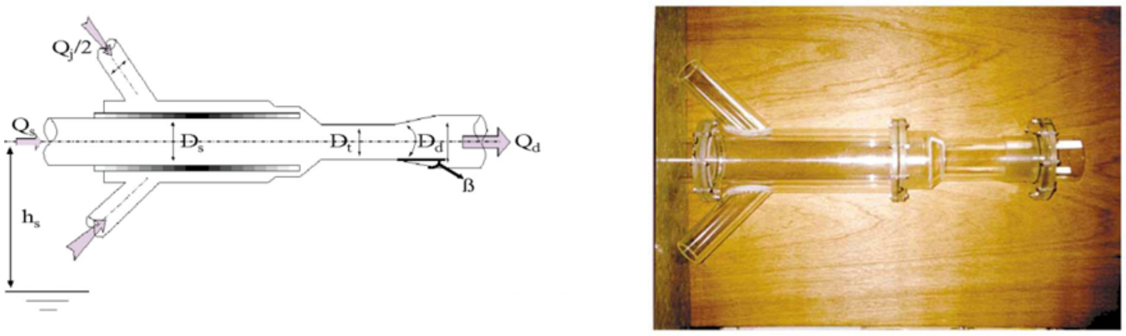
Due to its simplicity in structure, absence of moving parts and convenience of maintenance, jet pumps have been used widely in many fields for various purposes. Jet pumps are used widely in such fields as deep well pumping, dredging, pumping chemicals and transporting solids or fish. The jet pump works on the principle of a driving jet entraining the suction flow. In general, a conventional jet pump, a central jet pump, has a nozzle in the center where a primary driving fluid is injected, and an annular suction part. Many investigators (e.g., Ueda, 1954; Sanger, 1970; Yano, 1990; Jo, 1996) have studied the performance of this type of jet pump.

Another type of jet pump, an annular jet pump, has an annular nozzle on the outside and the suction pipe in the center. Many investigators (e.g. Shimizu and Nakamura, 1987; Donald and Sam, 1994) have studied the annular type jet pump. However, most of the works done by these investigators are confined to a given shape of the jet pump. Thus, there are not enough data for various shapes of the annular jet pump. There are not many reported numerical simulations for this type of jet pump.

To see the effects of the shape of mixing chamber on the performance of the jet pump for several jet flow rates, an annular jet pump tested by one of the authors (Oh and Kwon, 1998), which was designed for the transportation of fish, was modeled in this paper. Several computations are done and the results are discussed to provide better data for further development of annular jet pump.

2. A Numerical Model

In this study, the jet pump, which was designed and tested by Oh and Kwon, for the transportations of fish, was modeled. The jet pump used in their experiment is an annular type jet pump. A schematic diagram and photograph of which are shown in Fig. 1. The primary fluid enters the jet pump through the annular nozzle of the pump while the secondary fluid flows in the suction pipe. Here, we set the potential heads h_s and h_d as constant. We call the contraction part and the throat of the jet pump the mixing chamber.



(a) Schematic diagram (b) Photograph

Fig.1. Schematic diagram and a photo of annular type jet pump.

We confined the computational domain to the jet, the mixing chamber and the outlet of the pump as shown in Fig. 2. We also assume the flows are mainly axisymmetric, thus enabling the two-dimensional calculations. The coordinate system and some notations regarding the computational domain are shown in Fig. 2. We set the length of the outlet long enough to ensure to apply the outlet condition of the flow. Some detailed dimensions of the computational domain are shown in Table 1.

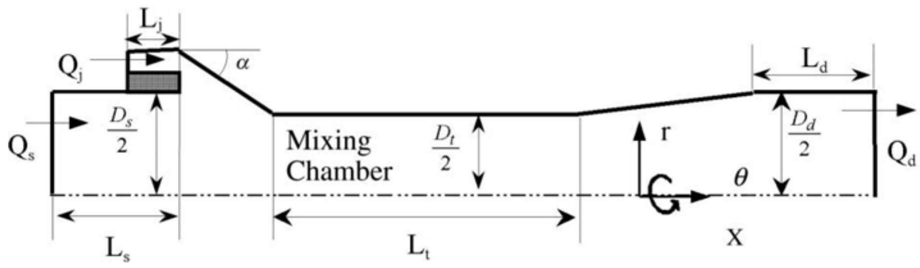


Fig. 2. Computational domain and coordinate system.

Table 1. Definition and dimensions of symbols.

Symbol	Definition	Dimension
D_s	Diameter of suction flow pipe	55 mm
D_d	Outlet diameter	55 mm
D_t	Diameter of throat flow pipe	47 mm
a	Reducing Angle	
b	Angle of divergence of diffuser	4°
L_s	Length of suction	430 mm
L_d	Outlet length	520 mm
L_t	Length of throat flow pipe	150 mm
h_s	Potential head of suction flow	1.4 m

3. Numerical Simulations

The standard k-ε model and the RNG k-ε model were used in this simulation. A finite difference method was used for the discretization of the continuity equation and the momentum equations, and the hybrid scheme was used for the convection-diffusion terms. The grid for the computation is shown in Fig. 3. Although we must use dense grid at the entire computational domain to get more accurate result, we used the following appropriate grid system shown in Fig. 3, resulting from the trial and error approach for the grid system. Instead of using dense grid at the entire computational domain to get accurate solutions, we used irregular grid system. Denser grid was used at the jet, near the wall, and at the entrance region of the computational domain, because rapid changes of velocities and pressures happen there, while coarser grid was used near the axis of the jet pump.

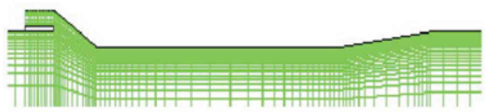


Fig. 3. Grid used for the computations.

A total pressure at the suction $p_{s,\,\mathrm{total}}$ was used as a boundary condition at the entrance of the suction flow and a jet flow rate, Q_j, as a boundary condition at the entrance of the primary fluid. The atmospheric pressure was used as the outlet boundary of the computational domain. Assuming no friction loss between the free surface of the suction flow and the entrance of the suction flow, we used the calculated total pressure at the entrance $p_{s,\,\mathrm{total}}$ from the following equation using the data from the experiment (Oh and Kwon, 1998).

$$p_{s,\,\mathrm{total}} = \frac{1}{2} rU_s^2 + p_s = p_{\mathrm{atm}} - rgh_s \tag{1}$$

The primary and secondary fluids are considered incompressible. The head ratio of the jet pump N and the ratio of flow rates M can be defined as:

$$N = \frac{H_d - H_s}{H_j - H_d}, \quad M = \frac{Q_s}{Q_j} \tag{2}$$

where H is total head. The subscripts d, s and j stand for discharge, suction, and jet. The efficiency of the jet, in general, is defined as a ratio of the energy used to induce the suction flow to the energy applied to the driving fluid as represented by the following equation.

$$h = \frac{rgQ_s(H_d - H_s)}{rgQ_j(H_j - H_d)} = M \cdot N \tag{3}$$

4. Numerical Results and Discussions

First, we carried out some computations for several flow rates using the grid system and the dimensions mentioned above in order to compare the simulated results with the experimental data (Oh and Kwon, 1998). FLUENT was used for the cmputations. Both the standard k-ε model and the RNG k-ε model have been tested. Figure 4 shows the dependence of the suction flow rate on the jet flow rate, while Figure 5 shows the dependence of the head ratio on the ratio of flow rate. The results using the standard k-ε model and the RNG k-ε model are shown along with the experimental data. As shown in Fig. 4, the suction flow rate increases with the jet flow rate. Unlike Fig. 4, the head ratio of the jet pump decreases with the ratio of flow rate. The computational results show relatively good agreement with the experimental data, while the RNG k-ε model shows slightly better agreement with the experimental data than the standard k-ε method. Hence, the RNG k-ε model was used for further calculations.

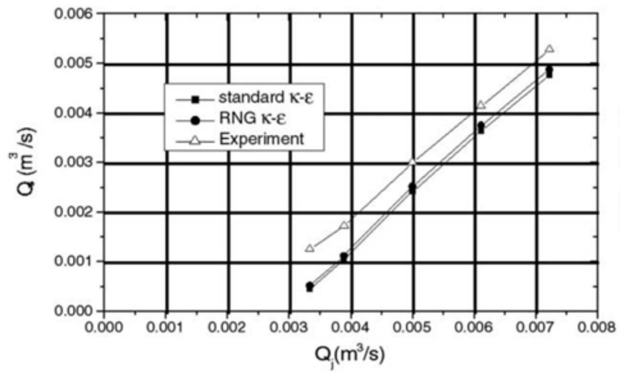


Fig. 4. Jet flow rate v.s. suction flow rate.

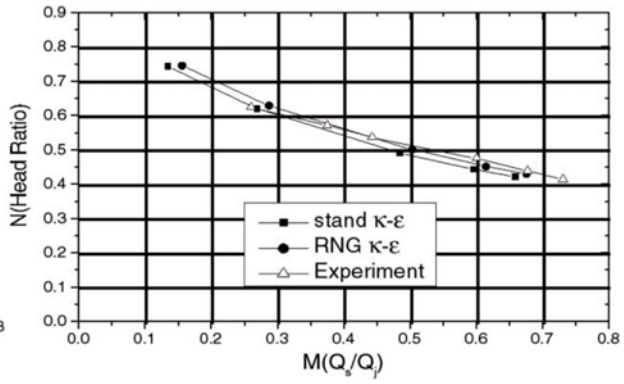
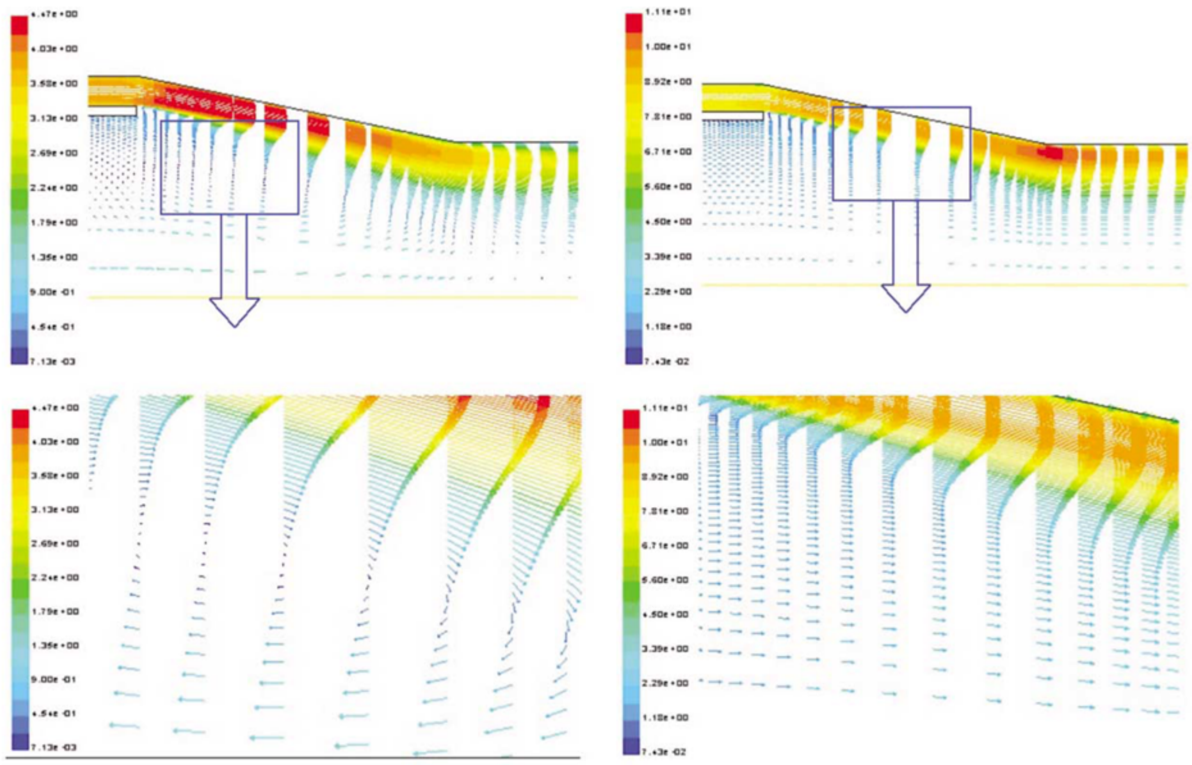


Fig. 5. Head ratio v.s. ratio of flow rate.



(a) Low flow rate (Q_j = 0.00333 m³/s) (b) High flow rate (Q_j = 0.00719 m³/s)

Fig. 6. Velocity profiles near the nozzle.

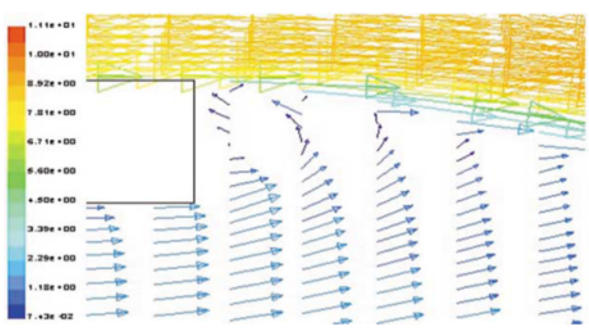


Fig. 7. Velocity profiles at the jet (Q_j = 0.00719 m³/s).

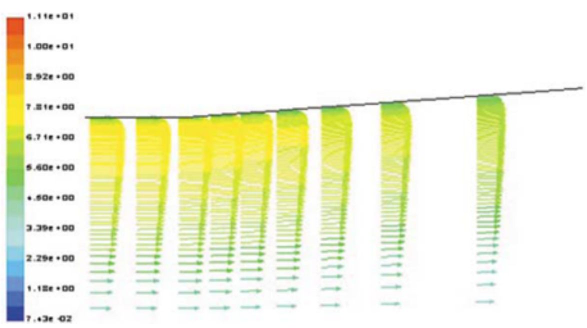


Fig. 8. Velocity profiles at the diffuser (Q_j = 0.00719 m³/s).

The velocity vectors of the flows in the mixing chamber are illustrated in Fig. 6 for two different jet flow rates and the detailed velocity vectors of the flows are also shown. Part of the primary jet fluid flows into the section pipe at the low flow rate, while the whole primary jet fluid flows toward the throat and induce the suction flow at the high flow rate, which explains why the efficiency is usually high at the high flow rate.

Figure 7 shows the detailed velocity vectors just after the nozzle for high flow rate. From this figure, we can see the jet flows induce the suction flows. The velocity profiles at the diffuser are shown in Fig. 8 where we can see the flow fields are developing along the axis.

Several computations have been carried out to seek the effects of the reducing angles on the suction performance and the efficiency of the jet pump. The reducing angles of 8°, 12°, 16°, 20°, 24°, 28° and 32° were used in this computation for several flow rates. Figure 9 shows the efficiency as a function of suction flow rate for several values of the reducing angle. The efficiencies increase with the ratios of the flow rates for every value of the reducing angle. The highest efficiency happens around at the reducing angle of 12°.

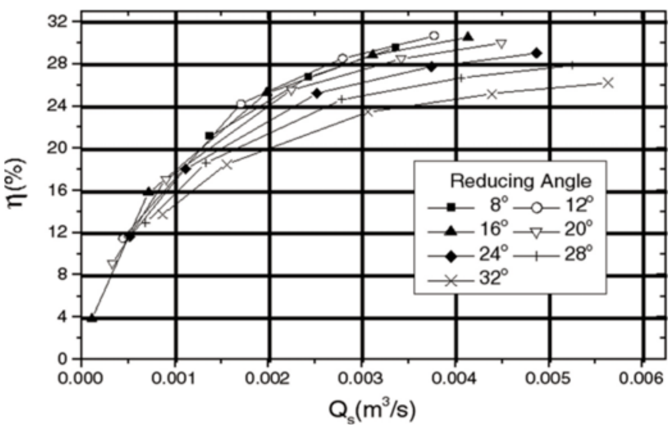


Fig. 9. The efficiency of the jet pump with various reducing angles.

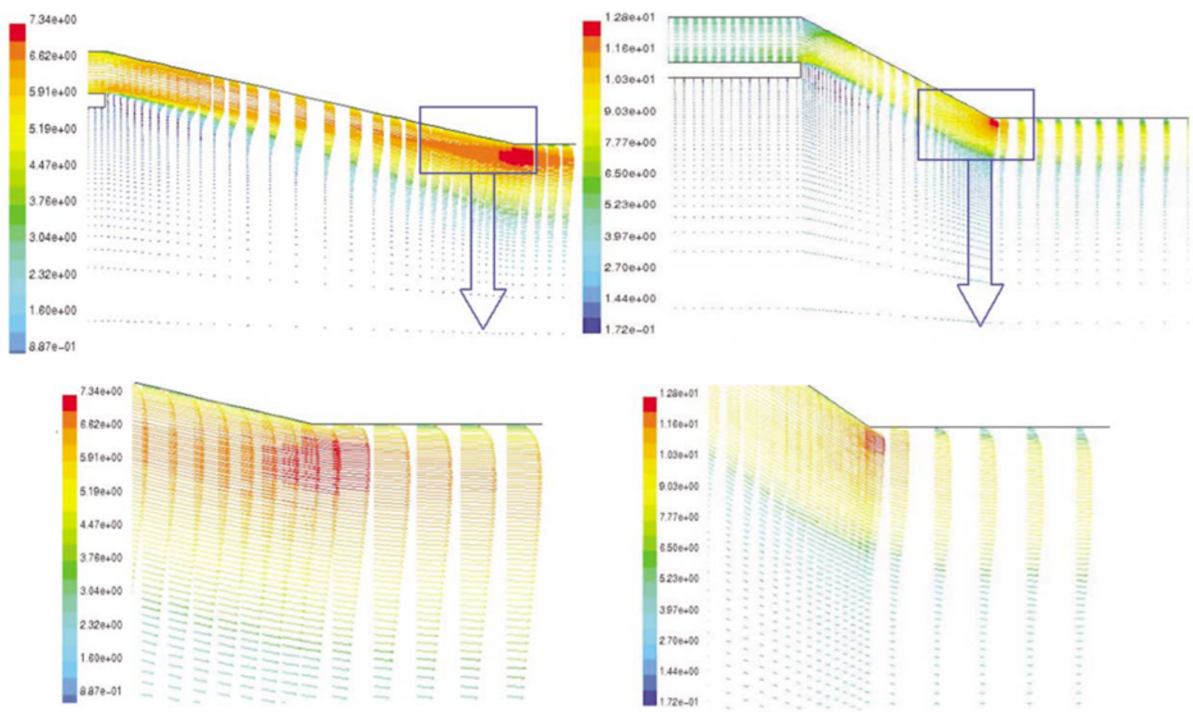


Fig. 10. Velocity profiles at the mixing chamber (jet flow rate: 0.005 m³/s).

Figure 10 shows the velocity profiles near the wall at the mixing chamber for two different reducing angles. It seems that there are big differences between these profiles, revealing that the suction flow is induced relatively well by the primary flow for the reducing angle of 12°, while it is induced poorly by the primary flow for the reducing angle of 32°. Hence, the jet pump with the reducing angle of 12° is more efficient than the one with the reducing angle of 32°, as is apparent in Fig. 9.

The pressure distributions along the axis of the pump for several flow rates are shown in Fig. 11. Pressure changes rapidly at the end of the jet, where a vacuum occurs. For high jet velocity, the pressure distribution shows the typically desired shape. The pressure drops rapidly at the entrance of the throat and increases at the diffuser part of the pump.

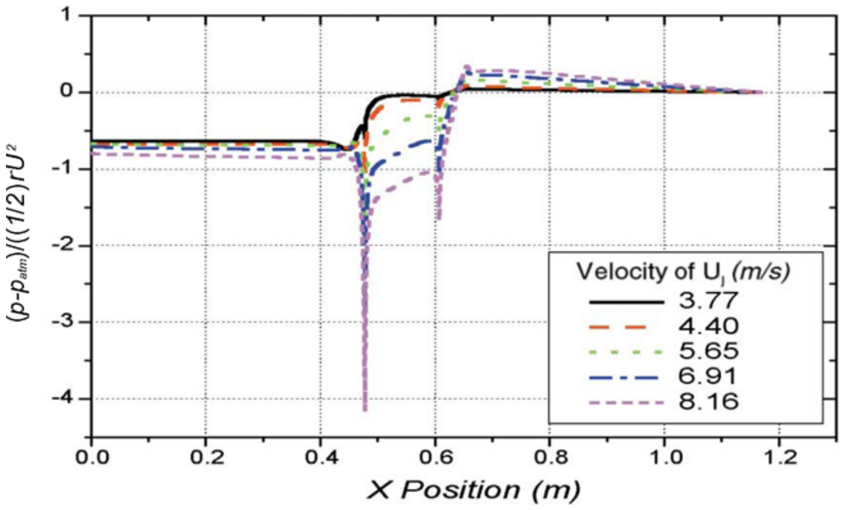
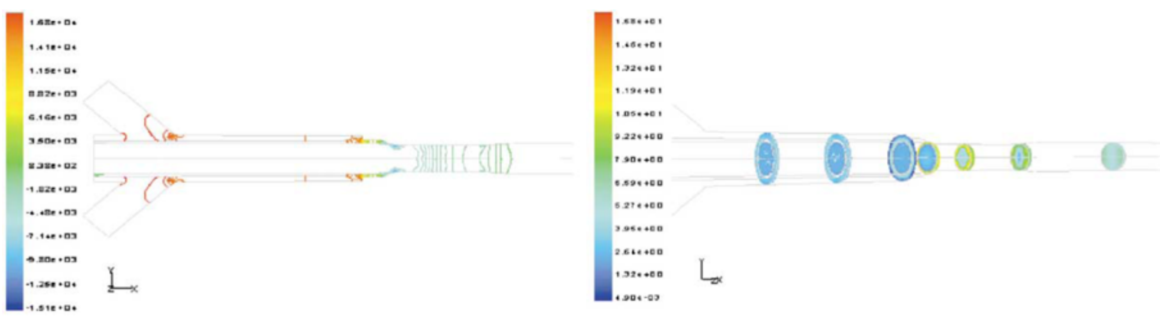


Fig. 11. Pressure distribution along the axis.

We carried out three-dimensional computations modeling the real jet pump tested by Oh and Kwon (1998). Although we have not carried out the entire simulations with several parameters of the pump, we present some velocity and pressure distributions. The computed pressure distributions and some velocity profiles are shown in Fig. 12 and the detailed velocity profiles at the location of 0.2 m from the suction inlet in Fig. 13. The outer velocity vectors of Fig. 13 represent the velocities at the outer chamber, just before the nozzle, while the inner velocities represent the velocities at the suction pipe. Figure 14 shows the velocities for the entire pump at the vertical section crossing the axis. The velocities are very high near the nozzle while the velocities at the center are very small. At the exit of the pump, the jet flows and the flows near the center get mixed well and it seems that there are no big differences between them.



(a) Pressure distributions (b) Velocity profiles

Fig. 12. Pressure distributions and velocity profiles for the three-dimensional simulations of the annular jet pump.

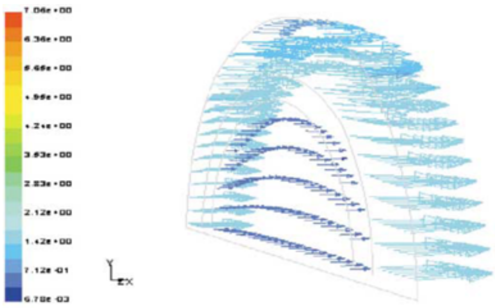


Fig. 13. Detailed velocity profiles at the location of 0.2 m from the suction inlet (jet flow rate: 0.00333 m^3/s).

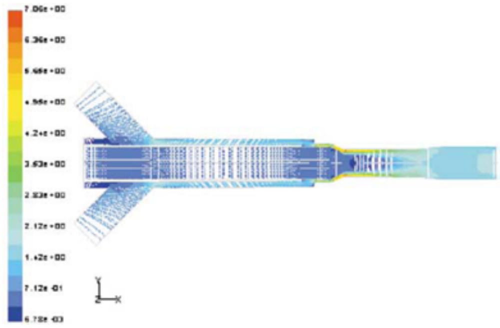


Fig. 14. Velocity profiles at the vertical section of the three-dimensional pump (jet flow rate: 0.00333 m^3/s).

5. Conclusions

Some computations have been carried out to study the effects of the shape of mixing chamber and the reducing angle of the jet pump for several jet flow rates. From the simulations of the annular jet pump, we can conclude as follows.

(1) The computational results have a relatively good agreement with the experimental data, while the RNG k-ε shows a little better agreement with the experimental data than the standard k-ε method.

(2) The efficiency of the pump with the reducing angle of 12° is the highest among the tested.

(3) There exist some re-circulations or reverse flows at the low flow rate.

(4) Some pressure distributions and velocity profiles are shown for two-dimensional and three-dimensional cases.

Acknowledgments

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