Effect of the collector tube profile on Pitot pump performances

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Abstract. The pitot pump is composed of the rotating casing with the impeller channel and the pitot tube type collector as the discharge line. The radial impeller feeds water to the rotating casing. The water rotating together with the casing is caught by the stationary pitot tube type collector, and then discharges to the outside. This type pump, as the extra high head pump, is provided mainly for boiler feed systems, and has been designed by trial and error. To optimize the pump profiles, it is desirable to investigate not only performances but also internal flow conditions. This paper discusses experimentally and numerically the relation between the pump performances and the flow conditions in the rotating casing. The moderately larger dimensions of the collector make the pump head and the discharge high with the higher hydraulic efficiency. The flow in the casing is almost the forced vortex type whose velocity is in proportion to the radius but the core velocity is affected with the drag force of the stationary collector. Based upon the above results, the profile of the pitot tube type collector was optimized with the numerical simulation.

1. Introduction
To get the sustainable society and to live together with the developing countries, it is required, as the advanced technologies, to guarantee the performance of the pumping system in the future infrastructure construction. In response to such requirements, this paper challenges to improve the efficiency of the pitot type pump [1], which is provided for boiler feed systems, cleaning of machined die-casting systems, welding machine cooling systems, reverse osmosis systems and so on. The pitot type pump has seriously weak points difficult to get fruitfully the higher head, discharge and/or efficiency, though the pump has noteworthy advantages that the pump is assembled with few parts and the maintenance is very easy. Many researchers have investigated the flow interaction between the rotating casing and the stationary pitot tube type collector (called sometimes pick-up tube), namely the discharge tube [2], how to reduce the friction loss in the pick-up tube [3], the modifications of the pick-up tube inlet profile form [4], but the relation between the pump performances and the internal flow conditions has been out of the investigations.

On such poor technologies, the authors have investigated that the flow conditions in the rotating casing is close to the forced vortex [5]. Continuously, this paper numerically discussed the effect of the pick-up tube profile on the pump performances, to get not only the higher head but also the higher discharge with the higher efficiency.

2. Model pump and experiments
2.1. Model pump
The model pitot type pump shown in figure 1 is composed of the rotating casing with the impeller channels, the stationary manifold connecting to the inlet pipe, and the stationary pick-up tube. The inlet of the pick-up tube is closed to the maximum inner diameter of the rotating casing. The flow velocity against the inlet of the pick-up tube is almost the same as the rotational speed of the casing. In the pick-up tube, the part of the dynamic energy is transformed into the pressure energy through the diffuser passage, and the water is discharged outside. That is, the pumping work quite differs from one of the traditional type pumps where the impeller gives the hydraulic power.

2.2. Experiments
The model was set as shown in figure 2. The pump discharge was measured by the electromagnetic flow meter, and adjusted by the gate valve at the outlet piping system. The pump head $H$ is estimated with the static and the dynamic pressures at the pump inlet and the outlet pipes, and the input power $P$ are estimated with the rotational torque $T$ and rotational speed $n$ without mechanical power such as water seals, bearings and so on. The rotational speed is kept constant with an inverter system.

3. Numerical flow simulation
The flow conditions of the rotating casing were predicted numerically by the commercial CFD code of ANSYS CFX ver.12.1 with SST turbulent model at the steady state. The flow field were divided into the rotational domain with the impeller channel which is enclosed by the rotating casing and the stationary domains with the inlet pipe, the pick-up tube and the discharge pipe, as shown in figure 3. Both domains were connected with the Frozen Rotor interface. The mass flow rate is given at the inlet, and the pressure is given at the outlet, as the boundary condition. Besides, the all walls have no slip condition, namely the velocity is zero at the stationary wall and the rotational speed at the rotating wall. The prism mesh was applied to the boundary layer and the tetra mesh was done to the main flow region. The number of the node is about 300,000 in the rotational domain and about 1,100,000 in the stationary domain, where the node was closed up near the wall.

![Figure 1. Meridian view of the model pitot pump.](image1)

![Figure 2. Experimental set up.](image2)
4. Verification of predicted results in comparison with experiments

Figure 4 shows the experimental-pump head $H$, the shaft power $P$, and the hydraulic efficiency $\eta$ in comparison with the numerical results, where $Q$ is the discharge and the subscript $\text{Exp-bep}$ means the value at the maximum efficiency in the experiment. The head decreases and the input power increases with the increase of the discharge, in a similar to the performances of the traditional type centrifugal pump. The simulated head and shaft power are lower than those of the experiment, but the distribution against the discharge is almost the same as the experimental result.

5. Optimization of the pick-up tube profile

5.1. Models provided for the flow simulation

To improve the pump performances, it is very important to take effectively the dynamic energy of the rotating fluid and to reduce the hydraulic loss in the pick-up tube. The typical profile of the pick-up tube is shown in figure 5 and the models satisfying the following conditions were prepared for the flow simulation.

- The tube curves smoothly from the tube inlet to the diffuser end.
- The tube has the circular cross section from the tube inlet to the diffuser end.
- The radial positions of the tube inlet and the diffuser end are independent of the model.
- The diameters of the tube inlet and the diffuser end are independent of the model.
- The diameter $d$ of the cross sectional area changes in proportion to the length $l_d$ from the tube inlet to the diffuser end [see figure 6, where $l_{dra}$ is the channel length of the traditional pick-up tube (the diameter is not in proportion to $l_d$) and the $d_{out}$ is the diameter of diffuser end].

Four models shown in figure 7 were prepared for the flow simulation, where Model Tube A has the ellipsoidal cross-section and is in the traditionally practical use.

5.2. Diffuser efficiency
The flow in the model pick-up tube channel was predicted numerically and was evaluated with the diffuser efficiency $\varepsilon$ defined as equation (1), where $p$ is the static pressure, $\rho$ is the density of the fluid, $v$ is the velocity, the subscript 1 and 2 means the tube inlet and the diffuser end.

$$\varepsilon = \frac{(p_2 - p_1)}{\left(\frac{\rho v_2^2}{2} - \frac{\rho v_1^2}{2}\right)}$$

Figure 8 shows the diffuser efficiency $\varepsilon$ predicted by CFD. The efficiency $\varepsilon$ increases obviously with the increase of the tube length $l_d / l_{dtra}$ and takes the maximum value at $l_d / l_{dtra} = 3$, where $\varepsilon = 0.3$ at $l_d / l_{dtra}$ for Model Tube A.

5.3. Pump performance

Figure 8 suggests that Model Tube D is the best profile in consideration of the diffuser efficiency. The Model Tubes D and E, however, are not practical about the manufacturability. Then, Model Tubes B and C were installed in the model pump shown in figures 1 and 3 to predict the pump performances shown in figure 9, where each value is divided by the value at the best efficiency point (subscript $tra$-be$p$) of Model Tube A. The pump heads of Model Tubes B and C are higher than the head of Model Tube A, and then the efficiencies are also improved considerably by making the tube length long. The head and the efficiency are affected not only by the diffuser efficiency but also by the outer profile whose drag contributes to the flow velocity, namely the dynamic pressure, in the rotating casing. Based on the above discussions, this paper recommends Model Tube B to make the pump head increase with the higher efficiency.

5.4. Modification of the cross-section
The cross section of Model Tube B was modified to improve the efficiency more and more, where the cross section was changed as shown in figure 10. The channel has the circular cross section from the inlet to the section A. The circular cross section changes gradually to the ellipsoidal cross section from the sections A to B, where the width ratio $t/c = 0.25$ or $2.25$. The ellipsoidal cross section also changes gradually to the circular cross section from the section C to the outlet.

The effect of the cross section on the pump performances is shown in figure 11. The blunt profile with $t/c = 2.25$ brings the velocity defect of the rotational moment as recognized in figure 12 ($r/r_e$: the dimensionless radius divided by the outer casing radius, $\omega$: the angular velocity, $v_\theta$: the absolutely swirling velocity) and makes the shaft power increase, and then the efficiency deteriorates markedly.

![Figure 8. Diffuser efficiency.](image1)

![Figure 9. Pump performances.](image2)

![Figure 10. The channel of cross-sectional with ellipsoidal form.](image3)
6. Concluding remarks

As a preparatory step to make the head and the discharge increase with the higher efficiency, the pick-up tube installed in the pitot pump was optimized with the numerical simulation.

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