Experiments and numerical simulations of a flow instability in a low-specific-speed pump-turbine.

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Abstract. In order to clarify the mechanism of the fluid oscillation, which occurs for a shut-down sequence of a pump-turbine, the experiment and numerical simulation of the pump-turbine were performed. The pressure fluctuation is measured at inlet, outlet and four points between the runner and guide vanes. Parameters are flow rate, rotating speed and guide vane opening ratio. Results show that the characteristic curve shows so-called “S-shaped” and the pressure oscillation occurs in the “S-shaped” region.

Key words: Pump-turbine, Flow Oscillation, Surging, Stall

1. Introduction
A pumped-storage hydroelectricity can generate and store the electric power. It can respond to variation of a power demand. In the pumped-storage hydroelectricity, a pump-turbine is used. When an electrical trouble occurs in the generating operation, the generator is cut off and consequently the pump-turbine operates without load. Under the no-load condition, it was reported that flow oscillation occurred in the operating conditions between a turbine-brake and a reverse-pump modes. The flow oscillation caused the large vibration of the power plant [1-2]. However, the flow characteristics and the mechanism of flow oscillation are still unclarified. In the present study, the flow characteristics in a pump-turbine were investigated by the water tunnel experiment and the numerical simulation.

2. Experimental Methods
Figure 1 shows the schematics of the experimental facilities. The water tunnel is closed type and consisted of the pump-turbine, the booster pump and the tank.

Figure 2 shows the schematics of the pump-turbine part and the pressure measurement points. The pressures \(p_i\) and \(p_o\) indicate the ones at the inlet and outlet. The pressures \(p_1\)~\(p_4\) between the guide vane and the runner were measured as the pressures at the circumferential locations of 72, 142, 182 and 242
degrees, respectively, from the tongue in a counterclockwise direction. The pressures sensors (KYOWA DENKI, PGMC-A) were used. For inlet and outlet pressure measuring, the sampling frequency is 2500 Hz and the sampling data number $2^{15}=32768$. For pressure measuring between the runner and guide vane, the sampling frequency is 12500 Hz and the sampling data number $2^{16}=65536$.

Table 1 shows the specification of the runner and guide vane. The tip diameter of the diameter is $D=0.134$ m. The specific speed of the pump-turbine is $N_{SO}=nQ^{1/2}/H^{3/4}=36$. The rotating frequency of the runner is $f_n=15$ Hz. The design flow rate is $Q/\sqrt{H}=0.0020$.

The torque was measured by the torque meter (YUNIPARUSU, UTMII-20Nm). The flow rate was measured by the magnetic flowmeter (OVAL, LS5377-400B, Measurable range: 10~60 L/min) and the raising flowmeter (Osaka Flow Meter Manufacturing, RHS Type, Measurable range: 50~500 L/min).

| Runner     |                  |                  |
|------------|------------------|------------------|
| Number of blades, $N_R$ | 7                |                  |
| Inlet diameter, $D_{R1}$ [m] | 0.134            |                  |
| Outlet diameter, $D_R$ [m] | 0.063            |                  |
| Inlet blade angle, $\beta_1$ [deg] | 24.0             |                  |
| Outlet blade angle, $\beta_2$ [deg] | 29.5             |                  |

| Guide Vane |                  |                  |
|------------|------------------|------------------|
| Number of blades, $N_G$ | 20               |                  |
| Inlet diameter, $D_{G1}$ [m] | 0.180            |                  |
| Outlet diameter, $D_{G2}$ [m] | 0.135            |                  |
| Blade angle, $\alpha$ [deg] | 20.0             |                  |

![Figure 1. Experimental facilities.](image1)

![Figure 2. Schematic of experimental pump-turbine.](image2)
3. Simulation Methods
The ANSYS-CFX 17.2 was used for the numerical simulations. The governing equations are the continuity equation and the incompressible Navier-Stokes ones. The SST turbulence model was used. The unsteady three dimensional simulations were carried out.

Figure 3 shows the computational domains. The numerical configuration is identical with the experimental one. The inlet of the inlet pipe is located upstream from 3.5D of the turbine inlet, i.e. the tongue leading edge. The outlet of the outlet pipe is located downstream from 2.0D of the turbine outlet. The runner domain was treated with in the rotating frame. The others were calculated in the stationary frame. The transient rotor/stator model was applied at the interfaces between the stationary and rotating frames. The non-slip wall condition was applied at the wall surface. The total pressure and the mass flow rate were set at the inlet and the outlet boundaries, respectively. The computational domains for the runner and the guide vane were shown by green and red, respectively. The numbers of structure-grid cells for the runner and the guide vane were about 900,000 and 800,000, respectively. The sum of those is about 2,300,000.

4. Experimental and Numerical Results

4.1. Characteristic Curve
Figure 4 shows the pressure characteristic curves. The abscissa and ordinates indicate \( n/\sqrt{H} \) [rad\,m\(^{1/2}\)s\(^{-1}\)] and \( Q/\sqrt{H} \) [m\(^{5/2}\)s\(^{-1}\)], respectively. The numerical value of \( n/\sqrt{H} \) is 11% larger than the experimental one at \( Q/\sqrt{H} = 0.0013 \). However, as shown by the blue and red broken lines, both the numerical and the experimental pressure characteristic curves show S-shape. Therefore, the numerical result can be said to simulate the experimental one, qualitatively.

Figure 5 shows the torque characteristic curves. The abscissa and ordinates indicate the flow rate \( Q/\sqrt{H} \) and the torque \( \tau=4T/\rho Q\omega D^2 \), respectively. Here, \( T \) is the dimensional torque and \( \omega=2\pi f_n \) is the angular velocity of the runner. The numerical value of \( \tau \) is 20% larger than the experimental one at \( Q/\sqrt{H} = 0.0013 \). However, both the numerical and the experimental torque characteristic curves show the tendency that \( \tau \) increases as \( Q/\sqrt{H} \) increases. Therefore, the numerical result can be said to simulate the experimental one, qualitatively.
4.2. Pressure Spectrum Cascade

Figure 6 shows the spectrum cascade of the measured pressure at the point p₁ in Fig.2. The abscissa and ordinates indicate \( f/f_n \) and \( \tilde{\psi} = 2\tilde{p}/\rho U^2 \), respectively. The depth direction indicates \( Q/\sqrt{H} \). Here, \( \tilde{p} \) is the spectra, i.e. pressure amplitude, of \( p_1 \). All the pressures of \( p_i \), \( p_o \), \( p_1 \sim p_4 \) showed similar tendency.

In the region of \( Q/\sqrt{H} = 0.0005 \sim 0.0023 \), the peaks of the pressure oscillations surrounded by the red broken line are observed in the range of \( f/f_n = 1 \sim 1.8 \). The red broken line show a convex (U-shape) in the right direction, i.e. C-shape. At \( Q/\sqrt{H} = 0.0015 \), both the frequency \( f/f_n \) and the pressure amplitude \( \tilde{\psi} \) become maximum. Here, in Fig.4, the S-shaped curve is confirmed in \( Q/\sqrt{H} = 0.0006 \sim 0.0023 \). Therefore, it is considered that the flow oscillation occurs in the S-shaped region. Since the torque \( \tau \) becomes zero at \( Q/\sqrt{H} = 0.0015 \), the flow oscillation can be considered to occur near the zero torque point. Since the circumferential phase differences for all peaks are about zero, this flow oscillation is identified as a surge-like oscillation.

Figure 7 shows the spectrum of the numerical pressure at the point \( p_o \) in Fig.2, at \( Q/\sqrt{H} = 0.0018 \), where the torque becomes zero in the simulation. The abscissa and ordinates indicate \( f/f_n \) and \( \tilde{\psi} \), respectively. The numerical peak frequency \( f/f_n = 1.47 \) in Fig. 7 is almost same as the experimental one \( f/f_n = 1.45 \) in Fig. 6, where the torque becomes zero in the experiment. Therefore, the numerical result can be said to simulate the experimental one, quantitatively.
5. The Considerations for Details by Numerical Simulation

Figure 8 shows the numerical distributions of the radial velocity in the turbine at $Q/\sqrt{H} = 0.0018$. Here, $T_s = 1/f_s = 0.045$ s is the period of the surge. The red means that the radial velocity is positive, i.e. backflow.

The backflow is observed near the pressure side of all the runner blade leading edges. That is, all runner blades are stalled. The backflow is also observed near the suction side of all the runner blade trailing edges. Since the distribution in Fig. 8(a), especially backflow, is almost same as the one in Fig. 8(c) despite the runner rotation, the effect of the runner rotation on the velocity field is quite small. It is considered because the flow rate is fixed at the inlet boundary.

Figure 9 shows the numerical distributions of the vorticity in the turbine at $Q/\sqrt{H} = 0.0018$. The red means that the vorticity is positive and clockwise.

At the pressure side near the runner blade leading edge, the red regions in Fig. 9 approximately correspond to the red ones in Fig. 8, that is the clockwise vorticity is mainly produced by the backflow. The distributions between the runner blade trailing edge and the runner outlet show the larger temporal change in Fig. 9 than in Fig. 8. Therefore, the vorticity can be said to cause the surge-like oscillation mentioned in Fig. 7.

![Figure 8. Radial velocity distribution ($Q/\sqrt{H} = 0.0018$).](image)

![Figure 9. Vorticity distribution ($Q/\sqrt{H} = 0.0018$).](image)
6. Conclusion
The flow characteristics were investigated by the experiments and numerical simulations. The following conclusions were obtained.

- The pressure and torque characteristic curve of the experiments and the tendency is confirmed by numerical simulation.
- The peaks of the pressure oscillations are confirmed in the S-shaped region by experiments.
- The flow oscillation is observed near the zero torque point.
- The pressure oscillation is caused by the vorticity between the runner blade trailing edge and the runner outlet.

References
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