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Solution of unstable problem for hydraulic turbine in order to promote usage of hydraulic energy

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Abstract. Hydraulic power generation is one of the most commonly used electric power generation by renewable energy. It is necessary to solve some practical and operational problems relating to hydraulic power plant in order to spread usage of hydraulic power generation. S-shape characteristics, which is one of unstable problems in hydraulic power plant, should be solved for designing of hydraulic turbines. It is important for numerical flow simulation to predict the S-shape curve with high accuracy, because a lot of design ideas to prevent the S-shape characteristic curve can be generated by analysis of numerical results of flow patterns. In this research, Francis turbine with low specific speed was simulated by steady and unsteady analysis in two cases, such as guide vane opening of 100% and 40%. Numerical results of turbine head and torque agreed well with experimental ones at guide vane opening of 100%. In case of guide vane opening of 40%, the simulation predicted successfully an increase of head which appeared in the experimental result. As the result, S-shape appears in a numerical characteristic curve similarly to the experimental one. The guide vane opening of 100% had a small backward flow in a radial velocity at the chamber between guide vanes and runner blades. On the other hand, the opening of 40% had two different flow patterns in first half and last half of the S-shape characteristics curve. In first half, there are almost no back flow in radial velocity at the chamber between guide vanes and runner blades. In last half of S-shape characteristics, a stable vortex flow which contained two forward flows in crown and band side, and a backward flow in the middle region between crown and band appears. The runner torque changed to be minus when the backward flow appeared, and the minus torque means that the runner gave kinematic energy to the water. The water with high energy was transported to the upstream guide vane by the back flow and this flow patterns caused the head increase. And this head increase caused the S-shape character in characteristic curve of guide vane opening of 40%.

1. Introduction

Usage of renewable energy resources is important to solve global environment problems. Hydraulic power generation is one of the most commonly used electric power generation by renewable energy and 80% of it is generated by hydraulic power plant. However, it is necessary to solve some practical and operational problems relating to hydraulic power plant such as low efficient and unreliable operation in order to spread the usage of hydraulic power generation.



Francis turbine is one of the major hydraulic turbine systems used in hydraulic power plant. On Francis turbines including pump turbines, a turbine characteristic curve indicates S-shape relation at high speed factor and low discharge factor for a constant guide vane opening, in particular for small guide vane opening. Runaway speed point with zero turbine output appears and finally the turbine output changes to be negative, called as turbine brake around this operating region. This S-shape characteristic is formed by multiple discharge factor Q_{ED} against a given speed factor n_{ED} and causes an unstable operating condition at turbine start and load rejection. When a load rejection is caused in hydraulic turbine system, rotational speed is increased and the guide vane is going to be closed. And then, pressure on penstock is increased. If Francis turbine has S-shape characteristic, it may cause a pressure increase again. There are some cases in which this secondary pressure increase is higher than first one. In worst cases, the secondary peak value becomes higher than penstock design pressure. Therefore, it is important to create a new design method of Francis turbine which can prevent the occurrence of S-shape characteristic. And it will be possible to enhance the hydraulic power energy by using Francis turbine with no S-shape characteristic. The aim of this paper is to conduct a comprehensive analysis of flow behavior for S-shape characteristic by using numerical simulation technology.

At high speed factor and low discharge factor around the turbine brake region, a backward flow caused by separating flow from runner inlet appears in chamber between guide vane outlet and runner inlet. This backward flow takes hydraulic energy from runner to guide vane. In case of smaller guide vane opening, the backward flow does not impinge on guide vane because guide vane outlet diameter is large and distance from runner to guide vane is far. It means shock loss at guide vane outlet is decreased and turbine head can be increased by backward flow containing energy [1]. This increased turbine head makes speed factor n_{ED} to be smaller, then S-shape curve appears in 3rd quadrant region of turbine characteristics.

Model test for pump-turbine was conducted with zero condition of turbine discharge in order to measure pressure and radial velocity at chamber between runner blades and guide vanes [1]. It was investigated how turbine head could be affected by guide vane position and guide vane opening. S-shape region of reversible pump-turbine on turbine mode was analyzed by transient computational fluid dynamics with large grid scale [2]. Turbine characteristics curve, head and torque were predicted well with comparison of experimental results and flow phenomena in runner blade were analyzed in detail.

In this paper, numerical analysis for low specific speed Francis turbine is presented with the guide vane opening condition of 100% and 40%. Flow behaviors were investigated particularly at the chamber between runner and guide vane in order to demonstrate how S-shape curve was caused in the condition of smaller guide vane opening.

2. Numerical method

2.1. Computational model

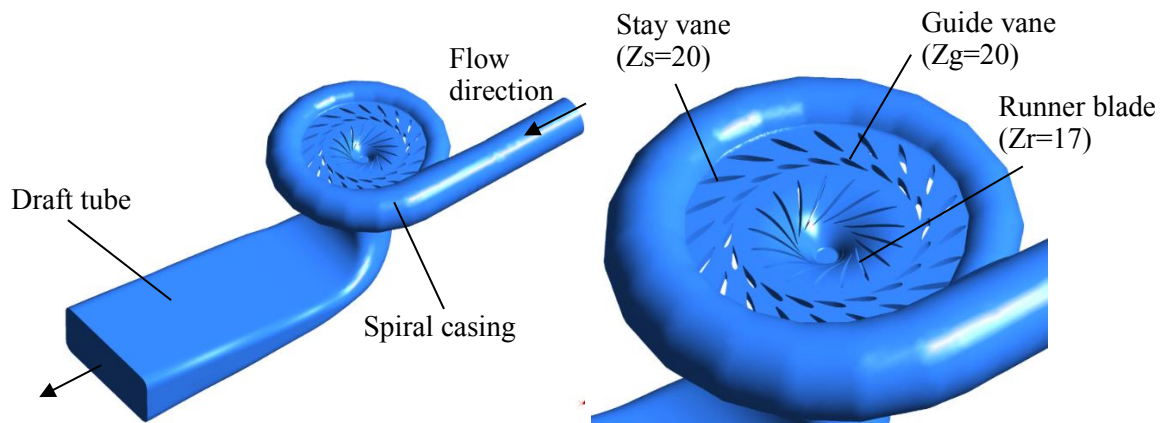
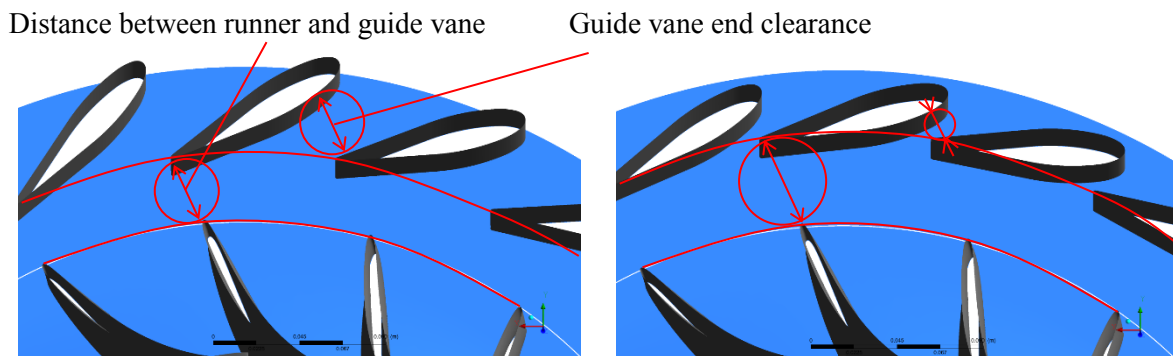
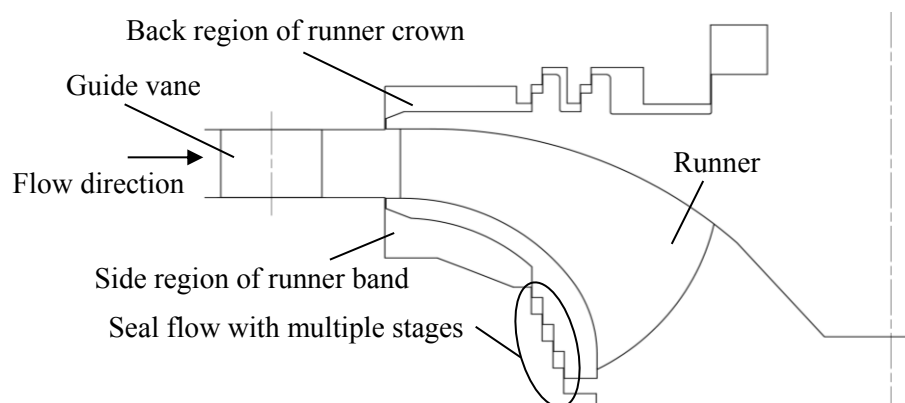
The numerical simulation and model test of low specific speed Francis turbine were carried out. The specification of the model turbine is shown in **Table 1**. **Figure 1**. shows computational domain considered in this research. The rotational part of turbine consists of a runner with 17 blades, and the stationary part consists of a spiral casing with 20 stay vanes, 20 guide vanes and draft tube.

The guide vane opening conditions are two cases of 100% and 40% explained in **Figure 2**. The guide vane end clearance is wider and guide vane outlet diameter is shorter in guide vane opening of 100%. On the other hand, the guide vane end clearance is narrower and the guide vane outlet diameter is larger in guide vane opening of 40%. Therefore, the distance between runner inlet and guide vane outlet is narrower in guide vane opening of 100% and wider in guide vane opening of 40%.

Leakage flow is generated in the back region of runner crown and runner band. The back regions were involved in a numerical domain in order to predict torque accurately in this research. **Figure 3**. indicates the numerical domain of back region. Leakage flows are generated in the seal flow passage with multiple stages.

Table 1. Specification of model turbine

Specific speed	N_{QE}	(-)	0.076
Runner outlet diameter	D_2	(mm)	0.285
Reynolds number	Re	(-)	$3 \sim 5 \times 10^6$

**Figure 1.** Computational domain**Figure 2.** Guide vane opening conditions (Left : 100%, Right : 40%)**Figure 3.** Numerical domain for back region of runner crown and side region of runner band

2.2. Computational method

The information of grid number applied in this research is summarized in **Table 2**. The numerical domain consists of all components of Francis turbine and is divided into six subdomains such as the spiral casing with stay vane, guide vane, runner, back region of runner crown, back region of runner band and draft tube. Mixed mesh with tetra and prism topology was prepared for spiral casing and hexahedron topology was prepared for other parts.

Table 2. Numerical grid number

Numerical part	Mesh Type	Grid number (million)	
		Guide vane opening of 100%	Guide vane opening of 40%
Spiral casing with stay vane	Tetra-Prism	0.3	←
Guide vane	Hexahedron	4.3	4.4
Runner	↑	3.7	←
Back region of runner crown	↑	1.5	←
Back region of runner band	↑	0.9	←
Draft tube	↑	0.8	←
Total node	—	11.5	11.6

Commercial software ANSYS CFX ver. 18 was applied for numerical simulation. Shear Stress Transport model was employed for turbulence model. Steady simulations were conducted for guide vane opening of 100% and 40%. In case of 40%, unsteady simulation was conducted with the steady result used as the initial condition. In the unsteady simulation, time step was set to be two degrees of the runner rotation. The interface model between stationary and rotating domains was frozen rotor in steady simulation, transient rotor stator in unsteady simulation. Numerical boundary conditions for speed factor and discharge factor are described at **Table 3**. These conditions are located around the turbine brake and including operating conditions in which S-shape curve is revealed guide vane opening of 40%.

Table 3. Simulation range

Guide vane opening	GVO	(%)	100	40
Speed factor	n_{ED}	(-)	0.298~0.288	0.293~0.275
Discharge factor	Q_{ED}	(-)	0.11~0.03	0.05~0.01

3. Numerical results and analysis

3.1. Turbine characteristics curve

The 3rd quadrant region in four quadrant characteristics of a Francis turbine was simulated in this research. **Figure 4**. indicates numerical results of turbine characteristics quadrant, head coefficient and torque coefficient for guide vane opening of 100% and 40% respectively. Q is volumetric discharge, D is outlet diameter of runner, g is gravity acceleration, H is turbine head, U is circumferential velocity based on outlet diameter of runner, ω is angular velocity of runner, m is torque, ρ is water density. Both of numerical simulations at guide vane opening of 100% and 40% were conducted by steady method. In this paper, directions of runner rotation and discharge on turbine mode are indicated as positive value, and thus n_{ED} and Q_{ED} at 3rd quadrant are described as positive value in order to make it easier to understand the relation between n_{ED} and Q_{ED} . The operating conditions that speed factor n_{ED} is higher and discharge factor Q_{ED} is lower and finally zero in the 3rd quadrant region are indicated in **Figure 4**.

In case of guide vane opening of 100%, the characteristic curve is not formed to be S-shape, and the head is decreasing together with the decrease of discharge. Torque is also decreasing and finally it

becomes negative over the point of runaway speed condition. In case of guide vane opening of 40%, S-shape is apparent in turbine characteristic curve. The speed factor n_{ED} is decreased at operating condition of B4 in the turbine characteristic. The head is decreasing from B1 to B3, but B4 shows the increase of head. This increasing head at B4-B8 causes S-shape in turbine characteristic curve. Torques at B3-B8 are negative and it means that turbine output is not generated. In both of guide vane opening of 100% and 40%, experimental results can be predicted well by our numerical simulation technology.

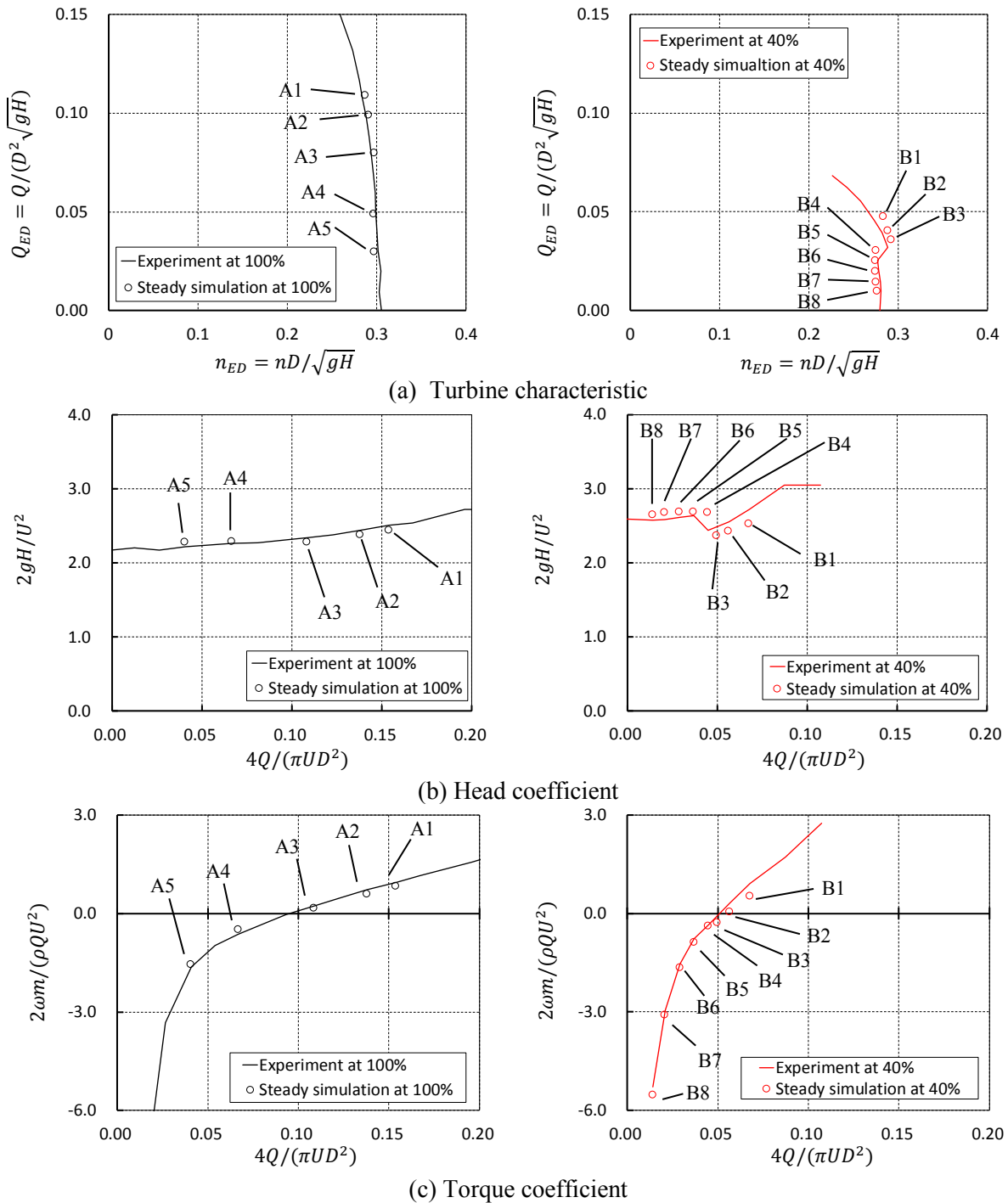


Figure 4. Numerical results of turbine characteristic, head coefficient and torque coefficient

3.2. Analysis of velocity distribution

Figure 5. and **Figure 6.** show radial velocity distribution at leading edge of runner inlet for guide vane opening of 100% and 40% respectively. This radial velocity is averaged one through the circumferential direction. The vertical value is axial position normalized by height of runner inlet and top value of 1.0 indicates runner crown, bottom value of 0.0 indicates runner band. Negative radial velocity is forward flow from guide vane to runner, positive is backward flow from runner to guide vane. In case of guide vane opening of 100%, backward flows do not appear at all operating conditions without A5. On the other hand, in case of 40%, backward flows appear at B4, B5 and B6. When the discharge is lower, the centrifugal force effect from runner to guide vane is stronger than the main flow going from guide vane to runner. This relationship causes backward flows in the middle region between runner crown and band. B4, B5 and B6 have similarity of radial velocity distribution, and all of torques are negative at those conditions. These backward flows contain energy supplied by rotating runner, and the increase of turbine head is generated by this backward flow in **Figure 4.**

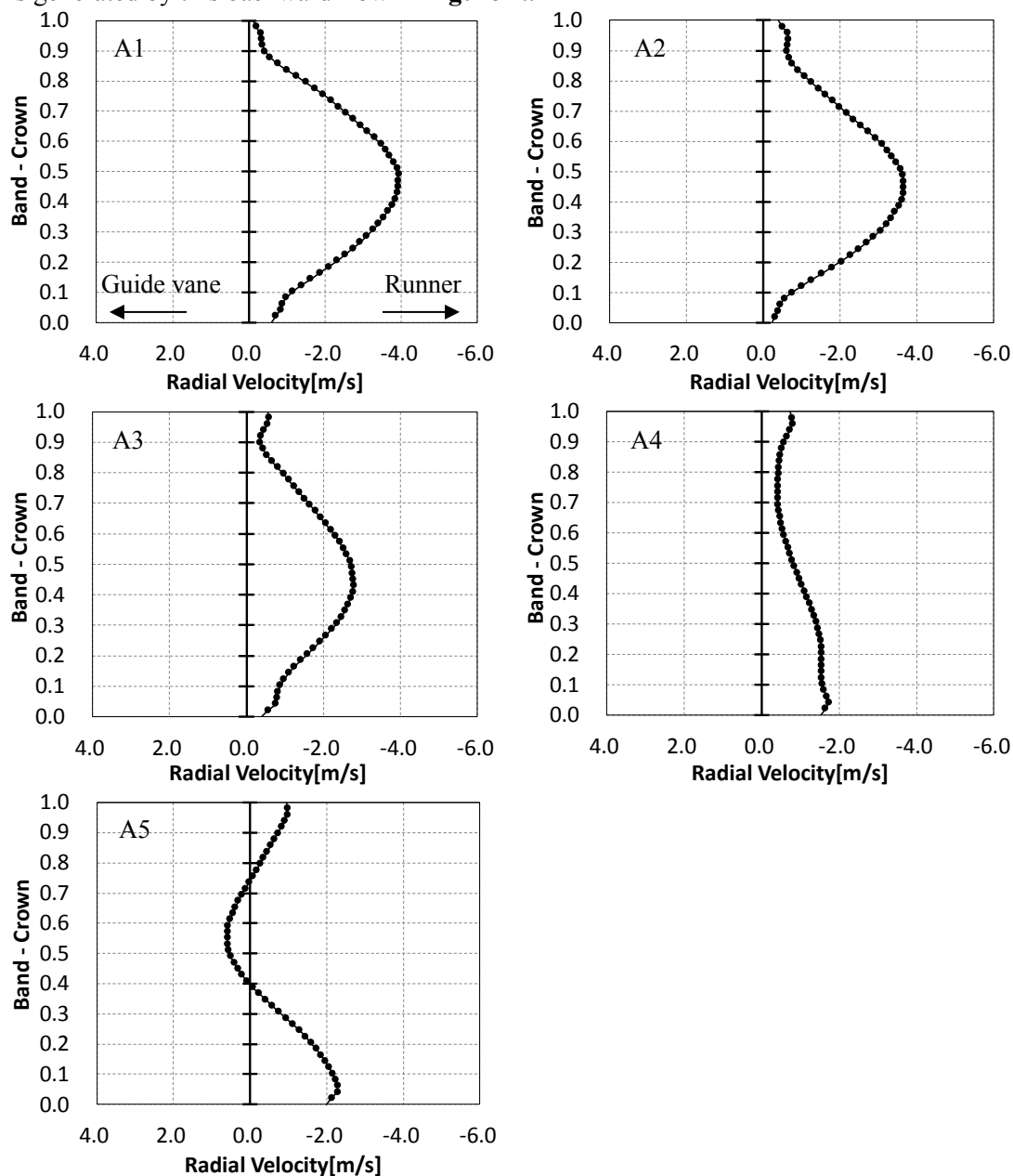


Figure 5. Radial velocity distribution at leading edge of runner for guide vane opening of 100%

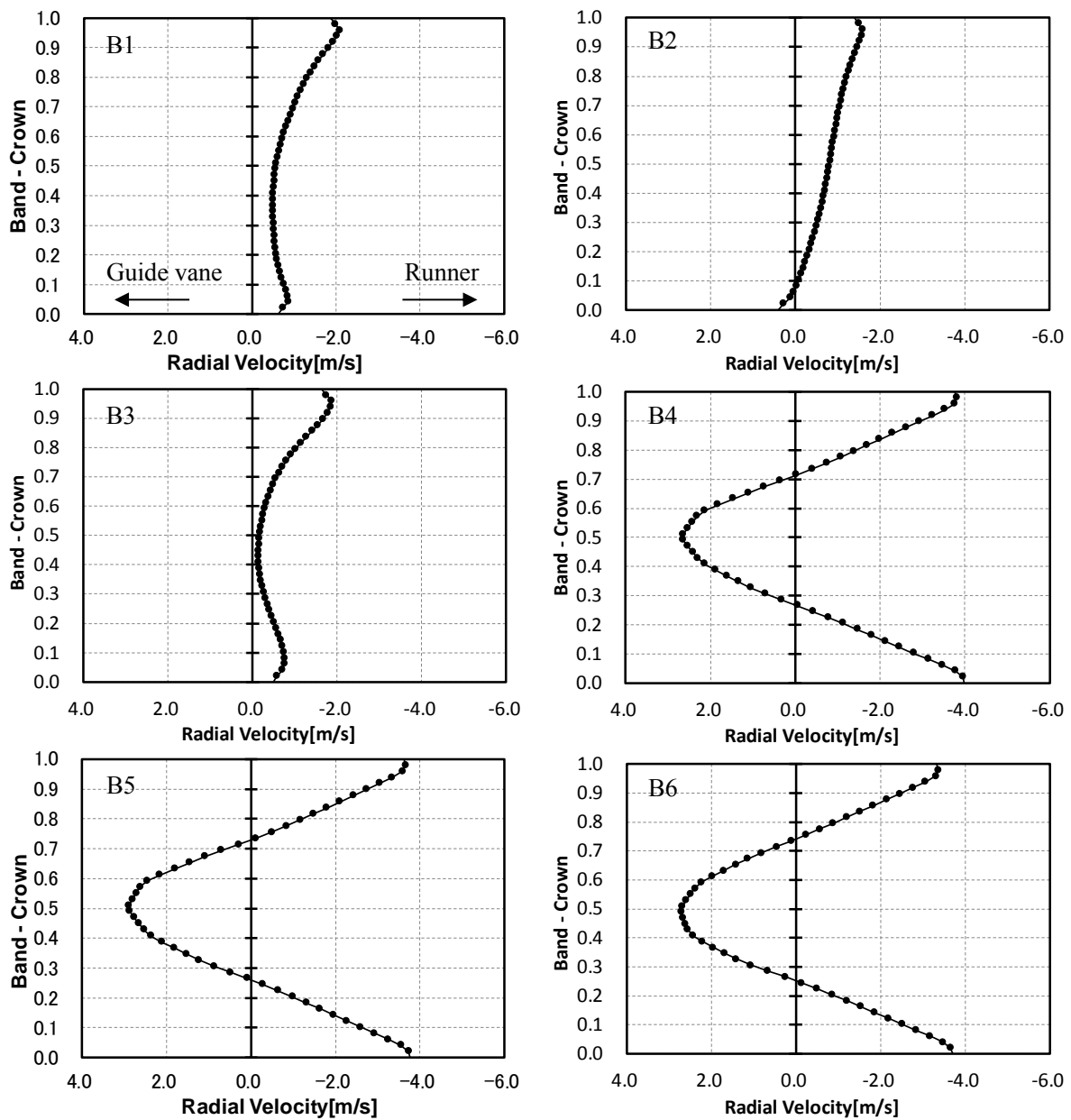


Figure 6. Radial velocity distribution at leading edge of runner for guide vane opening of 40%

3.3. Unsteady numerical simulation

Unsteady numerical simulations were conducted at B1, B3, B4 and B5 for guide vane opening of 40% shown in **Figure 7**. S-shape curve can be predicted more correctly than that of steady simulation. The turbine head and torque curves can be predicted correctly as well as turbine characteristic and numerical accuracy is better than that of steady simulation.

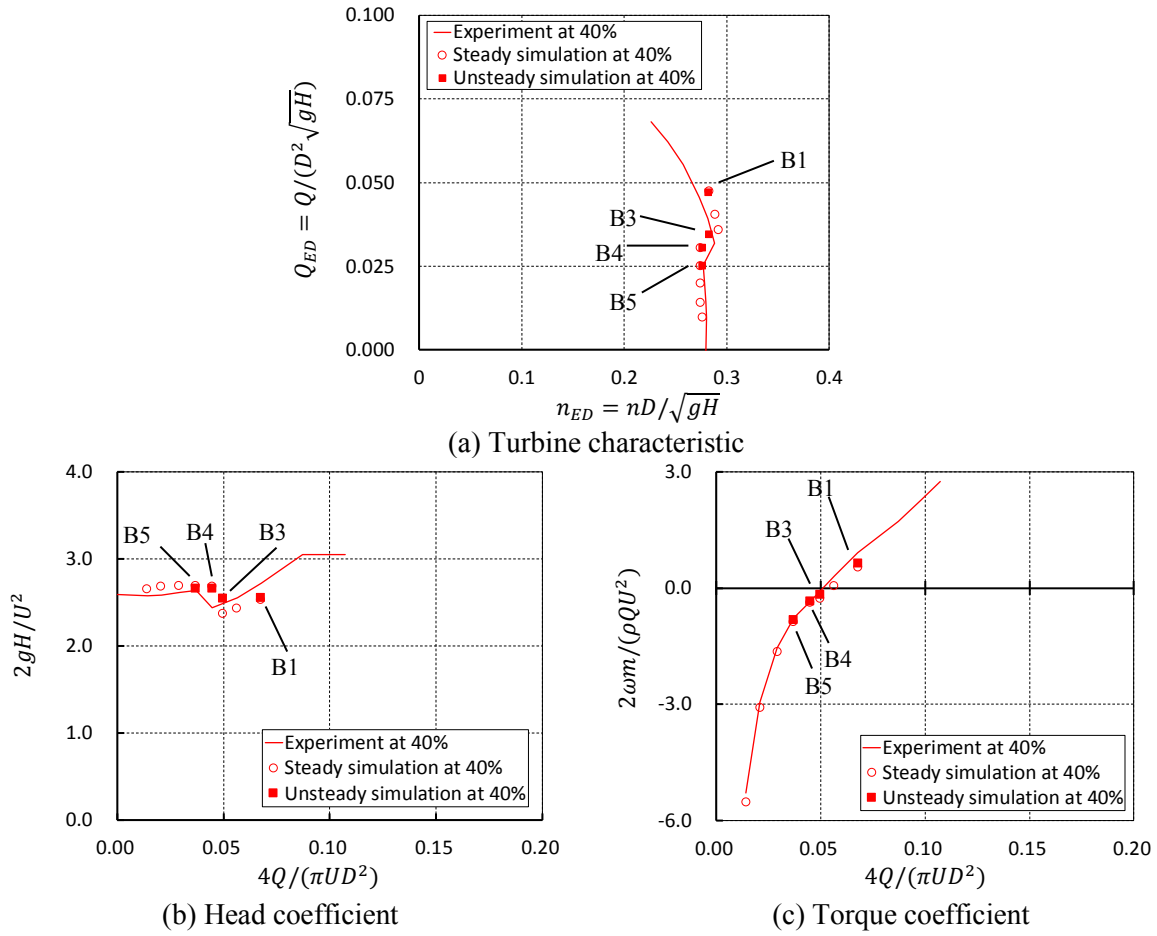


Figure 7. Numerical results of guide vane opening of 40% by unsteady simulation (Top : Turbine characteristic, Bottom left : Head coefficient, Bottom right : Torque coefficient)

Flow phenomena between guide vane and runner at B5 and B3 were investigated by using unsteady numerical results. B3 is before occurrence of head increase and B5 is after occurrence of head increase. **Figure 8.** and **Figure 9.** shows iso-surfaces of radial velocity with value of 2m/s around the leading edge of runner blade at B3 and B5 respectively. Positive radial velocity value indicates backward flow from runner to guide vane. So those flow patterns can be recognized as backward flows. Contour distribution on wall surface of band shows pressure coefficient defined as follows.

$$C_p = \frac{p - p_{inlet}}{1/2 \rho U^2} \quad (1)$$

Time t is normalized by rotational period of runner T_0 . Rotation direction of runner is counterclockwise. In case of B3, multiple backward flow regions appear between guide vane and runner. Besides, these backward flows are different sizes from each other and changing these sizes as time progress. When focusing on the backward flow around blade2 at $t/T_0=0$, it takes $t/T_0=6.67 \times 10^{-2}$ for this backward flow to move to the next blade1 in **Figure 6.** It means that those backward flows are revolving in circumferential direction with a rotating speed which is lower than that of runner. In case of B5 when discharge is reduced from B3, multiple backward flows around each blade are connected and formed to be a round ring shape. This round ring is revolving at the same speed as that of runner.

Figure 10. shows Fast Fourier Transform (FFT) analysis of unsteady pressure at the chamber between guide vane and runner at B5 and B3. Vertical axis indicates the amplitude of fluctuating

pressure and horizontal axis indicates frequency divided by frequency of runner rotation. f_0 is frequency of runner rotation, Z is number of runner blade. B5 has so stable flow pattern at the chamber between guide vane and runner in **Figure 9**. that FFT result reveals only one significant peak at blade passing frequency (BPF). In case of B3, dominant peaks occur not only at BPF, but also at other lower frequencies such as three quarters, half, and one quarter of BPF. Those three peaks are generated by multiple backward flows rotating in the chamber between guide vane and runner which are demonstrated in **Figure 8**.

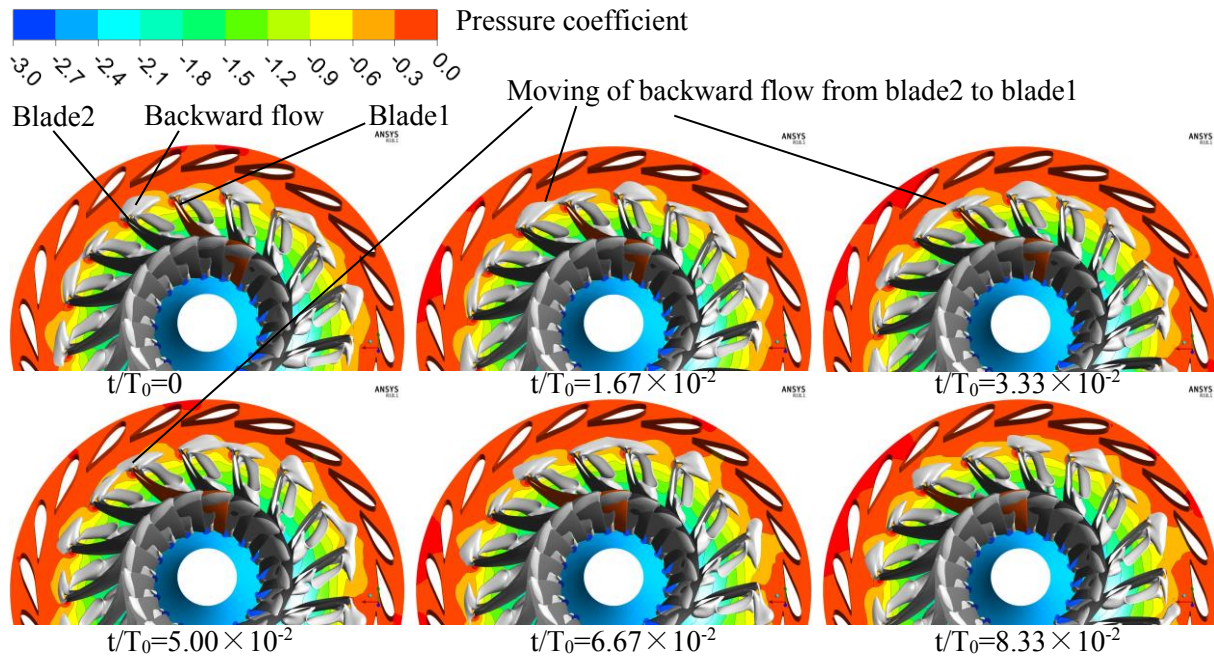


Figure 8. Iso-surface of radial velocity of 2m/s and contour distribution of pressure coefficient at B3

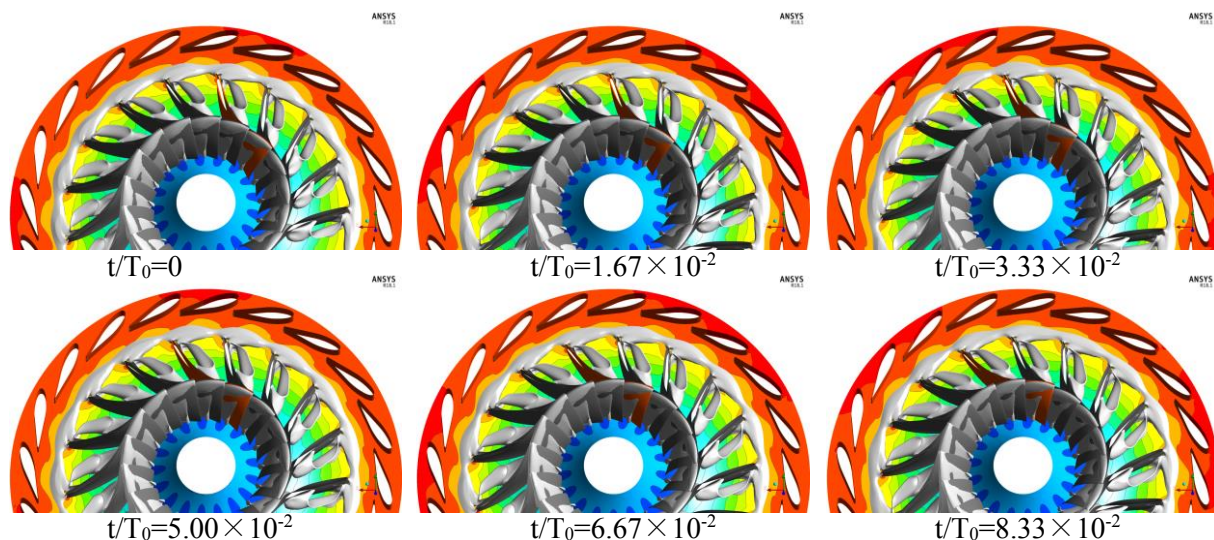


Figure 9. Iso-surface of radial velocity of 2m/s and contour distribution of pressure coefficient at B5

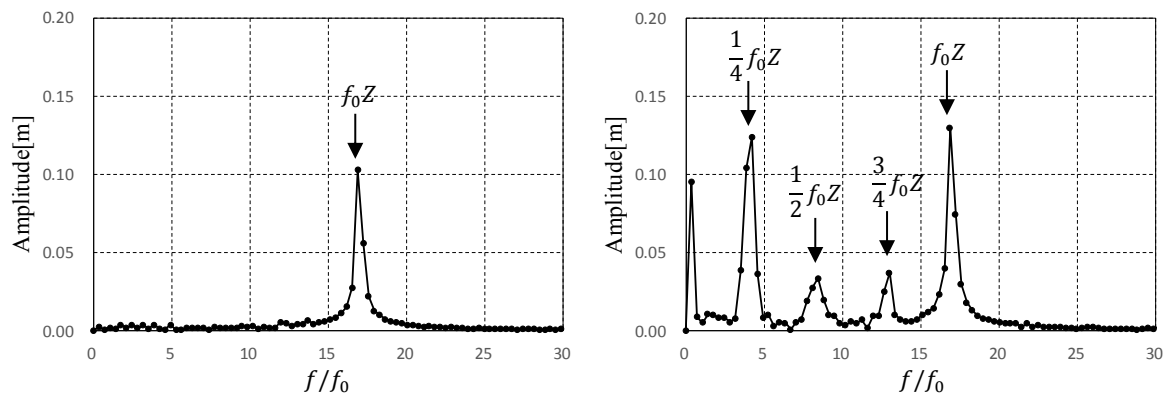


Figure 10. FFT analysis of transient pressure at positions between guide vane and runner (Left : B5, Right : B3)

4. Conclusion

There are some practical and operational problems relating to hydraulic power plant and it is important to solve those problems in order to spread the hydraulic power generation among the renewable energy resources. S-shape characteristic is one of design problems in Francis turbine used in hydraulic power plant. Therefore, investigation of S-shape characteristics for Francis turbine was carried out by numerical simulation analysis.

Steady numerical simulations were conducted for guide vane opening of 100% and 40%. In case of 100%, S-shape characteristic curve is not formed in experimental result, and numerical simulation could predict the characteristic curve accurately. In case of 40%, S-shape curve is revealed in experiment, and the numerical simulation could predict the S-shape characteristics accurately. The numerical result of turbine head showing the increase at lower discharge had good agreement with experimental one. It was found by numerical analysis that the increase of turbine head was caused by the backward flow from runner to guide vane. The backward flow makes head larger and it causes S-shape characteristic curve in guide vane opening of 40%. At lower discharge and after head increasing conditions of B4-B8, radial velocity distributions are similar curves with each other. The backward flow appears in the middle region and forward flows appear in both of runner crown and runner band region.

Unsteady numerical simulation was conducted to predict turbine characteristics for guide vane opening of 40%. The agreement of numerical result with experimental one was better in unsteady method than steady one. Multiple backward flows could be found in the chamber between guide vane and runner at lower discharge and before the head increase was caused. These backward flow regions were revolving in circumferential direction around leading edge of runner blade. At less discharge, finally these backward flow regions were connected and formed to be a round ring shape. That unified ring caused the head increase and S-shape appeared in the turbine characteristic curve. Pressure fluctuation at chamber between guide vane and runner was investigated by FFT analysis. At the operating condition in which head is increased by backward flow, flow pattern between guide vane and runner is stable, therefore the unsteady pressure has only one peak of fluctuation by blade passing frequency. On the other hand, in advance with the increase of turbine head, the unsteady pressure has four dominant peaks at blade passing frequency and lower frequencies of it, such as three quarters, half and one quarter of blade passing frequency because of those multiple backward flows rotating.

References

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