Replies to Reviewer’s Comments for Water

Reviewer 1 Comments and Suggestions for Authors

The present article describes the numerical work of cavitation vortex rope and inter-blade vortices typically observed in a hydraulic turbine, which is suitable for the scope of the journal. Few articles have reported the interactions between inter-blade vortices and the vortex rope. Thus, the reviewer found the present article useful for several readers especially in the field of hydraulic engineering.

However, some of the analyses presented in the article rely on the fact which is not sufficiently validated. The reviewer recommends the authors to revise the manuscript to make their analyses more rigorous and convictive. Specifically, the following comments should be considered by the authors to improve the manuscript.

Major comment
(1) Figure 5: If the normalized efficiency is defined by \( \eta/\eta_{BEP} \), then this value should be 1 at \( Q/Q_{BEP}=1 \), which is not for EXP_EFF. Could you please explain why this value is lower than 1 for EXP?

Answer to reviewer’s comment: Thank you for your comments. The power, efficiency and flow rate values related to the validation were normalized by the corresponding values of the BEP from the unsteady analysis results, respectively. Thus, the explanation for the normalization has been added in the manuscript as follows.

“The power, efficiency and flow rate values related to the validation were normalized by the corresponding values of the BEP from the unsteady analysis results, respectively.”

(2) Figure 9: The authors show the calculation result of the swirl number \( S \) as a function of \( Q/Q_{BEP} \) based on the equation (5). Given that \( \eta_{ED} \) is constant, the analytical value of \( S \) is obviously in inverse proportion to \( Q/Q_{BEP} \) as shown in the figure. The equation (5) is derived based on the assumption of uniform distribution of axial velocity and rigid body rotation of the circumferential velocity. Even though this assumption is satisfied at the upper part load condition, it becomes unrealistic when the discharge is too low such as GVA=7, as clearly mentioned in reference [22]. The authors should present not only the analytical swirl number \( S \) from equation (5), but also the actual swirl number in the performed simulations, which can be calculated according to the definition of the swirl number using simulated velocity components at a given plane (for example the plane of p3 where observed line is) to validate the relation between \( S \) and \( Q/Q_{BEP} \) in Figure 9.

Answer to reviewer’s comment: Thank you for your valuable comments. The authors have got the similar comments about the swirl number from other reviewer; reflecting these both opinions, the new graph related to the local swirl number based on the simulated local averaged velocity components at the observed line marked in Fig. 4 has been added with its related sentences in Fig. 11 as follows.

“In the Fig. 11(b), the local swirl number distributions are shown with various GVAs on the observed line (marked in Fig. 4) from center to the wall in the draft tube. The difference between the maximum and minimum values of the swirl number was increased gradually and their
differences were moved from center to wall of the draft tube, as the GVAs decreased. It can be confirmed with the absolute flow angle at the runner outlet as shown in Fig. 8; as the absolute flow angle decreased (as the GVA decreased), higher swirl characteristics also were moved toward the wall of the draft tube.”

![Graph showing analytical and local swirl number distributions](image)

**Figure 11.** Calculated swirl numbers from (a) analytical and on the (b) observed line for various GVAs.

(3) Figure 12: If the authors show C2*cos(a2) in Figure 12b, then it is not the radial velocity but the circumferential velocity. If this is the circumferential velocity, it would make sense that the high circumferential velocity is observed at GVA = 12.5 where the vortex rope is most significantly developed. However, the velocity distribution at GVA=7 conflicts with the velocity triangle shown in Figure 7, since it seems to rotate in the opposite direction as the runner rotation.

**Answer to reviewer’s comment:** Thank you for your comments. Accepting to the reviewer’s comment, the radial velocity has been replaced to the circumferential velocity and then related sentences and figure have been modified and added in the manuscript as follows.
“Figure 14 compares the time-averaged axial, circumferential velocity—”

In addition, as velocity triangle at GVA=7 in Figure 7 is the ideal case with the flow averaged from hub to shroud, the authors think that these phenomena can be different compared to the local time averaged circumferential velocity distribution with the above figure.

(4) The authors provide the value of normalized power and efficiency near the best efficiency discharge for the validations of performed simulations. However, the validation of the performed numerical simulations is not sufficient since the authors mainly focus on the vortex phenomena in off-design operations far from the nominal condition. Could you please provide more information from the experiments to validate these simulation results? For example, how close the vortex precession frequency in the simulation with respect to the experiment? The authors impose the total pressure in the calculation domain inlet, then how close the simulated discharge value at a given guide vane opening with respect to the measurements?

Answer to reviewer’s comment: Thank you for your comments. The experimental results used for validating the numerical analysis results are the performance test results of actual site pump-turbine which are operating currently at the hydro power plant of Korea. Therefore, some conditions (lower flow rate) cannot be applied and performed in detail as the real site pump-turbine; so, the test results were only executed up to 50% of the test load. As the results of basis performance test, there are no detailed results with pressure pulsation. Thus, the authors performed the unsteady numerical analyses to investigate the detailed flow phenomena in the real site pump-turbine in this work. The unsteady numerical analysis was conducted according to the guide vane angle with the water level (head) at the upper and lower reservoir of performance test; and then it was confirmed the validity of the numerical analysis results with similar tendency to experimental results. In addition, the uncertainties of the power and efficiency of the experimental results have been added in the manuscript and figure as follows.

“The experimental results of the performance test were provided from the real site which is one of the hydro power plants of Korea; and the uncertainty for the power and efficiency were ±1.14% and 0.91%, respectively, as shown in Fig. 5 [22].”
Figure 5. Comparison of numerical and experimental performance curves for the pump-turbine.

(5) Line312: ‘although the swirl intensity gradually increased as the flow rate decreased at the runner outlet’

This comment is related to the one above. This is validated only based on the “assumed” swirl number given in eq.(5). The authors should provide the actual swirl number that can be calculated by the velocity distribution in the simulated domain, to confirm the relation of the swirl number.

Answer to reviewer’s comment: Thank you for your comments. As mentioned the previous answer, the local swirl number using simulated velocity components at the observed line. The related sentences and figure have been modified and added in the manuscript as follows.

“In the Fig. 11(b), the local swirl number distributions are shown with various GVAs on the observed line (marked in Fig. 4) from center to the wall in the draft tube. The difference between the maximum and minimum values of the swirl number was increased gradually and their differences were moved from center to wall of the draft tube, as the GVAs decreased. It can be confirmed with the absolute flow angle at the runner outlet as shown in Fig. 8; as the absolute flow angle decreased (as the GVA decreased), higher swirl characteristics also were moved toward the wall of the draft tube.”

(a) Analytical swirl number distribution
(b) Local swirl number distributions on the observed line

**Figure 11.** Calculated swirl numbers from (a) analytical and on the (b) observed line for various GVAs.

(6) Line310: ‘With the GVA of 7°, the flow stagnation regions developed similarly in the runner passages, with complicated flow; however, the flow at the runner outlet was relatively uniform, without the flow stagnation regions.’

This fact is hardly confirmed in Figure 15. The authors should provide an additional figure to indicate the velocity is uniformly distributed at GVA=7. According to Figure 13, the flow velocity appears to be pretty much stagnated near the runner outlet at GVA=7, since the normalized meridional velocity is close to zero.

**Answer to reviewer’s comment:** Thank you for your comments. Accepting to the reviewer’s comment, the related sentence has been modified in the manuscript as follows.

‘With the GVA of 7°, the flow stagnation regions developed similarly in the runner passages, with complicated flow; however, the flow at the runner outlet was relatively uniform.”

(7) Line313: ‘when an inter-blade vortex developed in the middle of the runner passage, within the specific range of low flow rates, the internal flow in the runner passage affected the flow at the runner outlet and draft tube inlet, resulting in the development of the vortex rope.’

This fact is not evidently confirmed in the present study. The authors imply that the development of inter-blade vortices modifies the velocity distribution inside the runner, which leads to the vortex rope development as a consequence. To validate this description, the authors should present how the velocity distribution is modified by the presence of inter-blade vortices, and how this velocity modification links with the vortex rope development in the draft tube. Some of the authors have indicated that the key for vortex rope development is the velocity distribution at the inlet of diffuser, and the vortex rope structure is present even without the runner; for example: Study on Flow instability and countermeasure in a Draft tube with Swirling flow (2015), Nakashima, T., Matsuzaka, R., Miyagawa, K., Yonezawa, K., Tsujimoto, Y.

**Answer to reviewer’s comment:** Thank you for your comments. Accepting to the reviewer’s comment, Figures 11(b) and 14 have been added newly with the related sentences in the manuscript as follows.
(b) Local swirl number distributions on the observed line

Figure 11. Calculated swirl numbers from (a) analytical and on the (b) observed line for various GVAs.

Figure 14. Distributions of (a) absolute and (b) relative flow angles along the spanwise direction at the runner outlet with GVAs of 21.5°, 12.5°, and 7°.

“when an inter-blade vortex developed in the middle of the runner passage, within the specific range of low flow rates, the internal flow in the runner passage affected the flow angle distribution and the swirl characteristics in the runner outlet and draft tube, as shown in Figs. 11 and 14, resulting in the visible development of the vortex rope.”

(8) Figure 14: At GVA11, the vortex rope structure is reduced, but it appears to have twin vortex structure. In the lower part load condition, these multiple vortex rope structures may be observed. Although the authors concluded that the vortex rope disappears “visibly” in Figure 10 based on iso-surface of pressure, there may still remain some vortical structures at GVA=7 leading to pressure fluctuations (for example, the peak of 3.2fn in Figure 18 could be the peak related to the multiple vortex rope precession, not the one from inter-blade vortices). At GVA = 11 and 8, are there no frequency peaks in FFT result related to vortex rope structures? The relevant description in these conditions would be helpful to support the fact that the vortex rope is disappeared at GVA=7.

Answer to reviewer’s comment: Thank you for your comments. The results of FFT analysis of
the guide vane outlet as shown in Fig. 18 were confirmed that the frequency peak of 3.2fn was showed with GVA of 7° which condition is developed with the inter-blade vortex near runner LE; and there was no frequency peak in lower frequency regions with GVA of 12.5° when vortex rope developed greatly. In the below figure, the FFT analysis results of GVA of 8.5° and 11° through p1 to p4 in the draft tube showed that a relatively larger magnitude occurred in the generation of the vortex rope than GVA of 7°; and the frequency peaks were about 0.6~1.8fn, which is close to 0.2~0.4fn with GVA of 12.5°. However, since 0.2~0.4fn of GVA of 12.5° where vortex rope is generated greatly was not showed at the guide vane outlet, the 3.2fn of GVA of 7° was concluded as frequency peak induced by the inter-blade vortex.

**Figure 18.** Normalized magnitude along the flow direction in the draft tube with GVAs of 21.5°, 12.5° and 7° at measuring points (a) p1, (b) p2, (c) p3, and (d) p4.

**Minor comment**

(1) Line123: ‘Water and vapor at 25 C were considered as the working fluids in a two-phase flow ...’. What kind of the mixture model is used for this two-phase calculation? Is it a homogeneous model?

**Answer to reviewer’s comment:** Thank you for your comments. Accepting to the reviewer’s comment, the related sentences have been modified and added in the manuscript as follows.

“Water and vapor at 25 °C were considered as the working fluids in a two-phase flow to account for cavitation characteristics with the Rayleigh Plesset cavitation model, which describes the growth and collapsing of vapor bubbles in a liquid as the homogeneous model [19].”
(2) Line134: ‘A numerical grid dependency test was conducted …’ The test for grid convergence is conducted at QBEP? How the authors change the number of total nodes? Change the mesh density homogeneously? The value of yplus is modified, too? Please explain it in the text.

**Answer to reviewer’s comment:** Thank you for your comments. Accepting to the reviewer’s comment, the explanation has been modified and added in the manuscript as follows.

“The different girds were applied by changing the nodes while keeping the node density ratio for each component about the total nodes and y+ on the runner blade surface.”

(3) Line145: ‘The resolution during the unsteady-state analysis was 3° per time step’ This resolution 3 degree is based on what value? Is it based on runner revolution or vortex rope rotation?

**Answer to reviewer’s comment:** Thank you for your comments. The time step was set based on the runner revolution. Accepting to the reviewer’s comment, the related sentence has been modified as follows.

“The resolution during the unsteady-state analysis was the runner revolution of 3° per time step.”

(4) Figure 6: The authors indicate the iso-surface of the normalized velocity, but which value this iso-surface indicates? Is this the region where velocity is zero?

**Answer to reviewer’s comment:** Thank you for your comments. The velocity for the iso-surface was used with about 5% of the maximum velocity. The related sentence has been modified as follows.

“The iso-surface distributions show lower-velocity regions as about 5% of the maximum velocity that can be regarded as a flow stagnation region.”

(5) The definition of the normalized velocity is not clear. Although the authors indicate normalized velocity = V/Vmax, which Vmax does this indicate? Is it the maximum velocity in the entire domain? For instance, there is no high normalized velocity region (or region close to V/Vmax = 1) in Figure 13 (same in Figure 6 and Figure 11). If Vmax indicates the maximum velocity at the selected plane shown in the figure, the region of V/Vmax=1 should exist in the figure.

**Answer to reviewer’s comment:** Thank you for your comments. Even though there is maximum velocity in Figs. 6 and 11, it is hard to see clearly from three-dimensional streamline distribution. Accepting to the reviewer’s comment, Figure 13 has been modified correctly with the related sentences as follows.
Figure 13. Meridional velocity distributions on the hub span of the runner with GVAs of 12.5° (left) and 7° (right) during one revolution of the runner: (a) 1/4 τ, (b) 2/4 τ, (c) 3/4 τ, and (d) 4/4 τ.

“The velocity in the streamline was normalized by the maximum velocity (80 m/s) of the entire domain.”

“The meridional velocity was normalized by the local maximum meridional velocity (40 m/s) of the runner domain.”

(6) Figure 13: It is not obvious to see where the inter-blade locations are in the figure. The authors could present for example the tangential streamlines on the plane to help the readers see the vortex locations on the plane.

Answer to reviewer’s comment: Thank you for your comments. Accepting to the reviewer’s comment, the streamlines on the plane have been added as follows.
Figure 13. Meridional velocity distributions on the hub span of the runner with GVAs of 12.5° (left) and 7° (right) during one revolution of the runner: (a) 1/4 τ, (b) 2/4 τ, (c) 3/4 τ, and (d) 4/4 τ.

(7) Figure 14: What pressure value of the iso-surface does this figure indicate? Does it correspond to the vapor pressure? Since the authors conducted two-phase simulations including cavitation, it would be interesting to see how the cavitation in performed simulations is developed according to the vortex rope in different discharge.

**Answer to reviewer’s comment:** Thank you for your comments. Accepting to the reviewer’s comment, the related sentence has been added in the manuscript as follows.

“The iso-surface distributions of the pressure as the flow structure were decided with the relative water saturation pressure by considering the water level of lower reservoir.”

(8) Even though the velocity triangle at the runner inlet is slightly affected by the change of discharge, the major factor to change the velocity triangle is the head (or value of nED). The
inter-blade vortex development is largely affected by the inlet velocity triangle, and thus the turbine head generally has a large impact on its developing location and intensity. These are specifically investigated in the recent article: Physical Mechanism of Interblade Vortex Development at Deep Part Load Operation of a Francis Turbine (2019), Yamamoto, K., Muller, A., Favrel, A., Avellan, F.

The reviewer suggests that the authors should provide the value of beta at each operating condition to indicate the influence of the discharge value on the flow incident angle at the turbine inlet.

**Answer to reviewer’s comment:** Thank you for your comments. Accepting to the reviewer’s comment, the sentence and figure of the normalized beta angle distributions have been added in the manuscript as follows.

“In order to indicate the influence of the flow rate on the flow incidence angle, Fig. 9 shows the beta angle distributions at the runner inlet on the runner according to the GVAs as the different flow rate conditions. The beta angles were varied with each GVA as the flow rate changed. Therefore, the flow characteristics were influenced by changing the incidence angle between the blade and flow angle according to the GVAs as compared in Fig. 8.”

![Figure 9. Beta angle distributions at runner inlet on the mid-span with GVAs.](image)

**General comment**
The reviewer would like to ask the authors to thoroughly read the manuscript again to polish English sentences for some grammatical aspects. In addition, the reviewer requests the authors to carefully revise references to include more relevant references. For example, the following articles describe fundamental characteristics of vortex ropes with respect to different flow regimes as well as inter-blade vortex development, which must give a better insight for flow distributions and vortex development in hydraulic turbines described in the present article:


Operating Problems of Pump Stations and Power Plants (1982) by Doerfler, P.

LDV survey of cavitation and resonance effect on the precessing vortex rope dynamics in the draft tube of Francis turbines (2016) by Favrel, A., Muller, A., Landry, C., Yamamoto, K., and Avellan, F.

Comparison of model measured runner blade pressure fluctuations with unsteady flow analysis predictions (2016) by Magnoli, M. V.
**Answer to reviewer’s comment:** Thank you for your comments. Accepting to the reviewer’s comment, several references suggested by the reviewer have been added with its sentences in the manuscript.

“The inter-blade vortex and vortex rope in the hydro turbines have common characteristics with the vortices characteristics in the pump-turbines. Yamamoto et al. [6] studied to reveal a physical mechanism of the inter-blade vortex development in a Francis turbine at deep part load condition by the numerical simulation. Magnoli [7] conducted numerical simulation of the Francis turbine at full load, part load and deep part load by measuring runner blade pressure fluctuations with the rotor-stator interaction, rotating vortex rope and the runner channel vortex. Fay [8] investigated the low-frequency in the Francis turbine using transient torque equation with the rotor-stator interaction, inter-blade vortices and spiraling vortex flow as the various potential pulsation sources.”
