

The flow field investigations of no load conditions in axial flow fixed-blade turbine

J Yang¹, L Gao², Z W Wang^{*1}, X Z Zhou¹, H X Xu³

¹ Department of Thermal Engineering, Tsinghua University, Beijing, 100084, China

² Yuxi xinhua water resource and hydroelectric investment co., ltd, The Zhenghe Road 18th, Luolong District, Luoyang, Henan, China

³ Zhejiang zhongshui hydropower equipment co., ltd, Huangpoling 35th, Fuchun River town, Tonglu, Zhejiang, China

Email: wzw@mail.tsinghua.edu.cn

Abstract. During the start-up process, the strong instabilities happened at no load operation in a low head axial flow fixed-blade turbine, with strong pressure pulsation and vibration. The rated speed can not reach until guide vane opening to some extent, and stable operation could not be maintained under the rated speed at some head, which had a negative impact on the grid-connected operation of the unit. In order to find the reason of this phenomenon, the unsteady flow field of the whole flow passage at no load conditions was carried out to analyze the detailed fluid field characteristics including the pressure pulsation and force imposed on the runner under three typical heads. The main hydraulic cause of no load conditions instability was described. It is recommended that the power station should try to reduce the no-load running time and go into the high load operation as soon as possible when connected to grid at the rated head. Following the recommendations, the plant operation practice proved the unstable degree of the unit was reduced greatly during start up and connect to the power grid.

1. Introduction

No load operation of the hydro turbine is a state of hydro generator rotor driven by the turbine keeps rotating at rated speed, and the voltage and excitation current also reach the rated, while the generator is off the grid and the output is zero. During the process, the turbine needs to run up to synchronous speed at a low guide vane opening, so the turbine remains rated speed at low discharge. Hydro turbines of different series and different heads correspond to their own no-load starting opening, such as 3~10% for Pelton turbines, 5~15% for Francis turbines and 20~50% for Kaplan turbines, yet it is a common law for axial turbine that the larger blade angle, the greater no-load opening.



Working at no-load condition, the inner flow of the turbine turns out to be much more complex. That is because the flow just needs to resist friction and the flow rate is low. The effect of centrifugal force on the flow becomes stronger with the speed increasing, giving rise to reversal flow, high amplitude pressure fluctuation and strong vibration, which is unfavourable to the stable operation. Furthermore, changes of head, rotate speed and discharge would in return cause the shaft torque pulsation and even worse lead to the instability of the turbine and the whole hydraulic system ^[1,2]. Based on the no-load operation study, J Li ^[3] observed that rich complicated flow structures, such as secondary flow and cross flow, present inside the runner, which will form vortex, extra inertia force, high pressure pulsation and dynamic load ^[4]. In that situation, the turbine is not able to reach the rated speed at lower guide vane opening, and neither will make grid-connect operation at rated speed.

A plant found that the axial flow fixed-blade turbine appeared strong vibration while the unit was on no-load operation and connected to grid at a medium head in the actual operation, however, no strong vibration took place at no-load starting under both low head and high head, which seriously threatens the safety and stability operation. This phenomenon is not for the first time, the axial flow fixed-blade turbine of Hongshi station also appeared extremely unstable and created serious vibration at on-load starting, and finally it is determined by means of model and prototype test that vibration can be reduced through air admission at the top cover^[5]. In Tianhuangping station, the high no-load speed swing and unable to implement grid-connect operation also come out, and analysis revealed it was the disturbance of inner flow field that led to the no-load operation instability ^[6].

To further look into the reason of unit unable to stable operate at no-load under a medium head, this paper makes an extensive analysis in the inner flow and obtains the typical characteristics of the field to find out the reason of unstable performance, the results can provide guidance for the start-up unsteady under no load condition to solve the problem.

2. Calculation model and numerical method

Taking the whole flow passage as computational domain, with respect to the specific structure of the flow passage, the domain is divided into five regions, that is, spiral casing, stay vane, guide vane, runner and draft tube respectively, as shown in Fig.1.

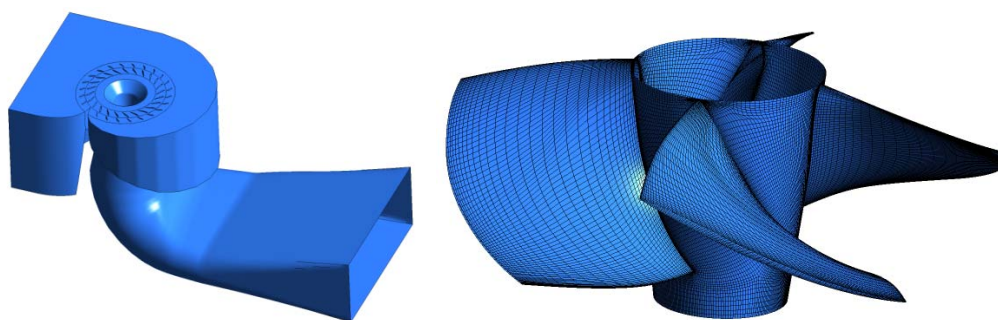


Figure 1. Flow passage model and the runner mesh

Considering the rotation property of the runner, the interfaces between rotating and stationary are set as “Frozen rotor” for steady calculation and “Transient rotor-stator” for unsteady calculation. As for the computational mesh, spiral casing and stay vane regions apply unstructured tetrahedral mesh; the other regions use structured hexahedral mesh. In order to capture the detailed results, mesh in the stay

vane and runner regions is locally refined and further improvement in the clearance. The node in runner regions is approximately 770,000. Eventually, the total number of the element and the node involved is approximately 1,729,765 and 1,336,702.

The influence of the time step on the flow field frequency was investigated on rated condition. Three time steps corresponding to 100, 200, and 300 steps per rotation cycle are compared, which correspond to time steps 0.0028s, 0.0014s and 0.000933s respectively. Table 1 shows the frequency at the guide vane outlet and draft tube cone. There is no significant difference among the results 0.0014s and 0.000933s time steps, cycle, so the time step is set to 0.0014s.

Table 1. Sensitivity of time step for frequency.

Time step	Guide vane outlet	Draft tube cone
0.0028s	14.225Hz	14.225Hz
0.0014s	14.286 Hz	14.286 Hz
0.00093s	14.284 Hz	14.284 Hz

According to the working condition, the adequate boundary conditions are set, the spiral casing inlet was pressure inlet where the total pressure is set and the direction of the velocity is perpendicular to the section, pressure outlet to the draft tube exit where pressure is relative to the level, and no-slide boundary to wall surface. The SST turbulence model was chosen to simulate the whole flow field

3. Result analysis

3.1 Steady analysis for rated head

To obtain the specific characteristics of the flow field at no-load operation and the no-load guide vane opening at rated condition, cases with different openings under rated head are calculated. The results are listed below. Fig.2 shows the power change with the opening and Fig.3 is the force acting on the runner.

As can be seen, the power rises with the increase of the opening and reaches the no-load condition point when the opening adds up to 27%, which means the unit can't implement grid-connected operation if the opening is below 27%. In this process, the hydraulics of the runner has a significant difference. Here use F_r as the radial force of the runner, $F_r = \sqrt{F_x^2 + F_y^2}$, where, F_x, F_y is respectively the force component of the blade wall in the coordinate x and y, keeps basically unchanged and so the effect can be ignored; F_z as the axial water thrust of the runner, also rises with the increase of the opening and the direction changes before and after the no-load opening, which means the runner is mostly influenced by the axial thrust during the process.

Fig.4 shows the inner flow of no-load operation at rated head. In order to understand the detailed flow, two extra sections A, B are added to reveal the serious axial and transverse vortex flow. From section A, transverse vortex is mainly concentrated near the hub region within less than 50% of the blade span, while there also exist large axial vortexes seen from section B.

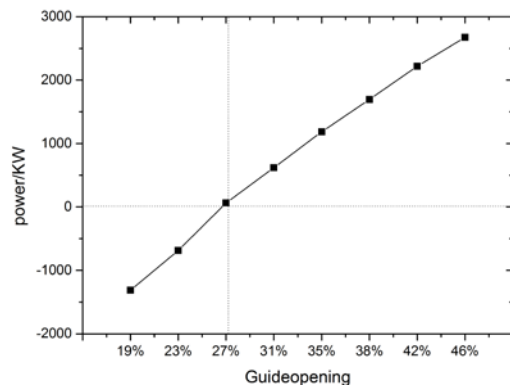


Figure 2. Power change with openings at rated head

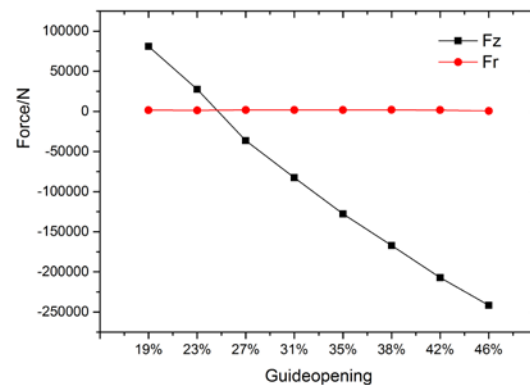


Figure 3. Runner Force with openings at rated head

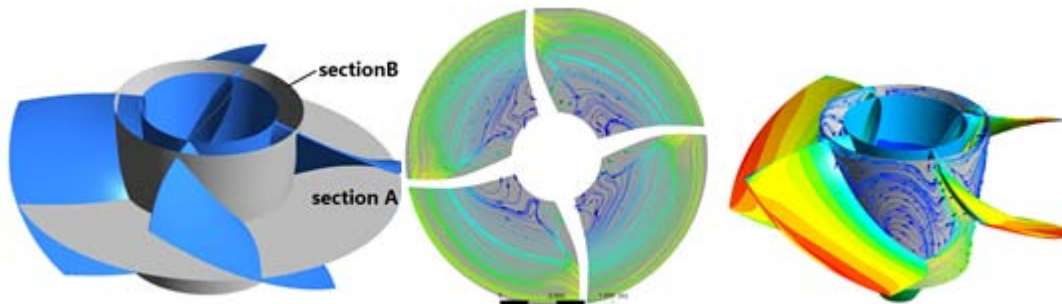


Figure 4. flow field distribution in the runner under no load condition at rated head

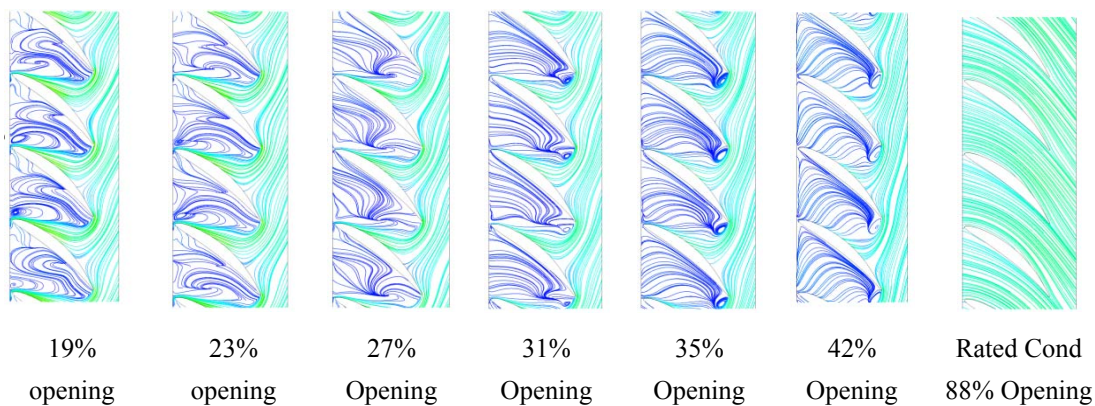


Figure 5. Streamlines of 10% blade span section at different openings

Based on the results, take 10% of the blade span section for example to compare with the flow specification among different openings, as shown in Fig.5. It is concluded that the vortex disturbance of the section flow is getting gradually weakened with the increase of the opening; at no-load point vortex almost dominates the whole channel between blades, and leads to vibration and instability, which greatly influences grid-connected operation at rated speed. However, the streamline between the inner channels becomes very smooth, without any distorted flow vortex. So the unstable problem of no-load operation can be settled by means of increasing the opening.

3.2 Flow analysis under different heads

According to the operation statistics, the unit is always required to perform grid-connected operation under different heads and then cases of different heads are taken into account as well.

Fig.6 summarizes the change tendency of no-load opening, force and discharge with the head variation. As stated above, ignoring the effect of radial force of the runner, just compare with F_z (axial thrust of the runner) under different heads. As we can see, the no-load opening as well as the amplitude of F_z gets lower with the increase of the head, which is easy to understand, the water power increases with the head, and the discharge needed to resist friction decreases, so no-load opening and axial thrust become lower.

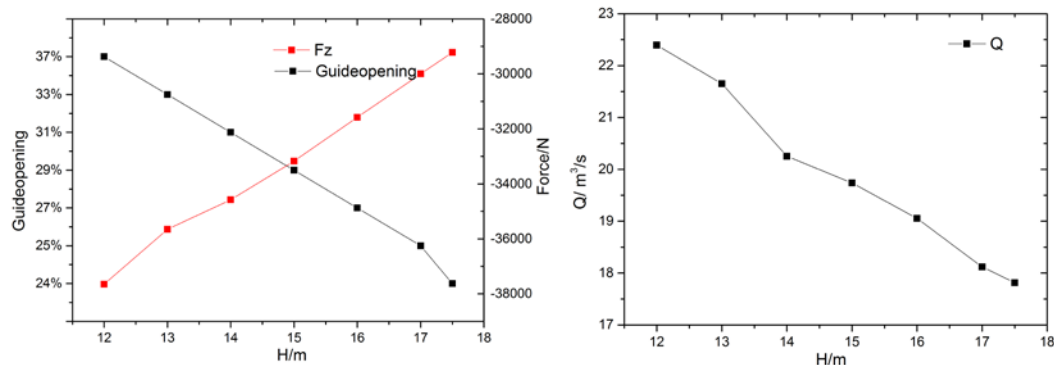


Figure 6. No-load opening, axial thrust and discharge under different heads

Different discharge at no-load condition corresponds to its own characteristics of the inner flow. The actual operation also exhibits that vibration appears particularly serious at a medium head. Therefore it is necessary to carry out detailed analysis in unstable flow characteristics under different heads to look for the reason of the increased vibration. Pick three typical heads, minimum head 12m, maximum head 17m and mean head 15m to study the flow field differences of them. The streamline changes across the span section A, B, were shown in Fig.7. It is observed that there are obvious flow vortexes between the inner channel, indicating the instability of the field and degrees of difference vary with the head. The effect of flow vortex on the blade span section gradually expands to shroud with the head increasing. That is because discharge decreases but centrifugal force strengthens as the head rises, which extends the vortex flow influence range along the spanwise.

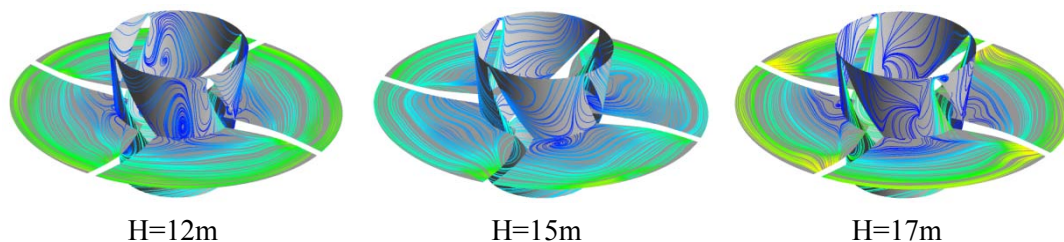


Figure 7. Flow field in section A and B under different heads

3.3 Unsteady flow analysis under different heads

For further analysis of the instability under different heads, no-load operations at rated speed of three typical heads are selected for unsteady calculation to analyse the flow characteristics, such as pulsation peak-peak value, pulsation frequency and axial force. The locations of record points for pressure pulsation are shown in Fig.8.

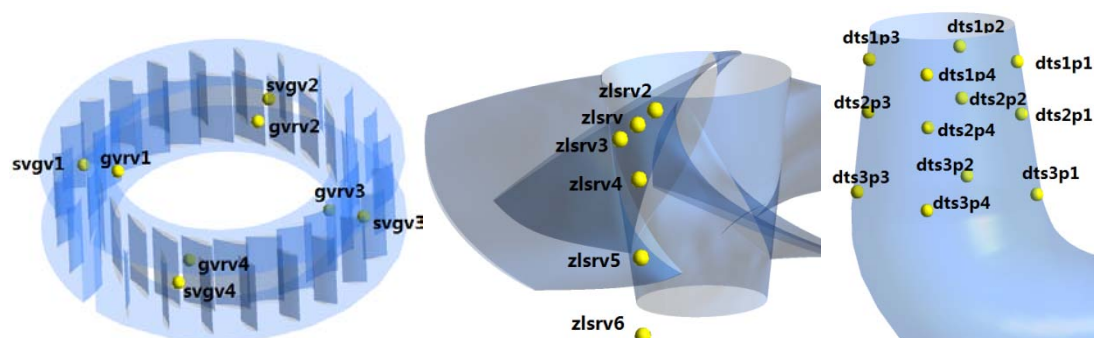


Figure 8. Record points in the flow passage

Starting with the hydraulic cause of the vibration, the peak to peak value of pressure pulsation corresponding to the record points at three heads are compared, as shown in Fig.9. As can be seen, the maximum amplitude of the pulsation peak-peak is located in the runner, secondly in the guide vane and obvious pulsation also appears in the upper of the draft tube. The pulsation peak-peak at 12m head is lower and the others increase, especially at the 15m head, meaning that the stability at both minimum and maximum heads is superior to that at 15m head. This can be interpreting as the combined action of centrifugal force and flow rate decrease. The rotating centrifugal force grows and the complex flow in the blade channel becomes worse with the head increasing from the minimum to the mean value, resulting in high amplitude pressure pulsation. As the head further increases, rotating centrifugal force also grows but discharge decreases, reducing the pulsation. As a result, the peak-peak of the pressure pulsation reaches maximum at the mean head. The same conclusion can be obtained through the detailed analysis of the runner force, as shown in Fig.10 which exhibits the change of the force component in the coordinate x, y and z.

It can be seen from Fig.10 that comparison with the three components F_x , F_y and F_z at the minimum head 12m, only the amplitude of F_z is relatively high, F_x and F_y components are very low. All of the three components display a regular periodic fluctuation, demonstrating a better stability of no-load operation at this head. With the head increasing, F_x , F_y also exits evident components, periodicity of each force component get worse, which means the operation instability at high heads enhanced. Compared with the condition at 15m and 17m heads, the fluctuation at 15m head has a higher amplitude, which strengthens the evidence that vibration becomes worse when the unit performs no-load operation at the mean head. To explain the change of the periodicity in the force component, the frequency characteristics of the pressure pulsation at record points are shown in Fig.10.

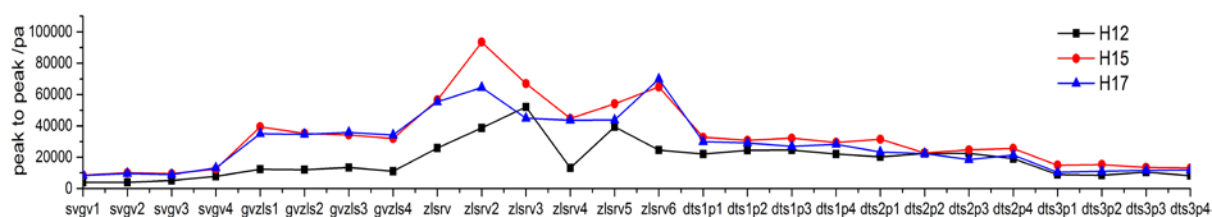


Figure 9. Peak to peak value of pressure pulsation under different heads

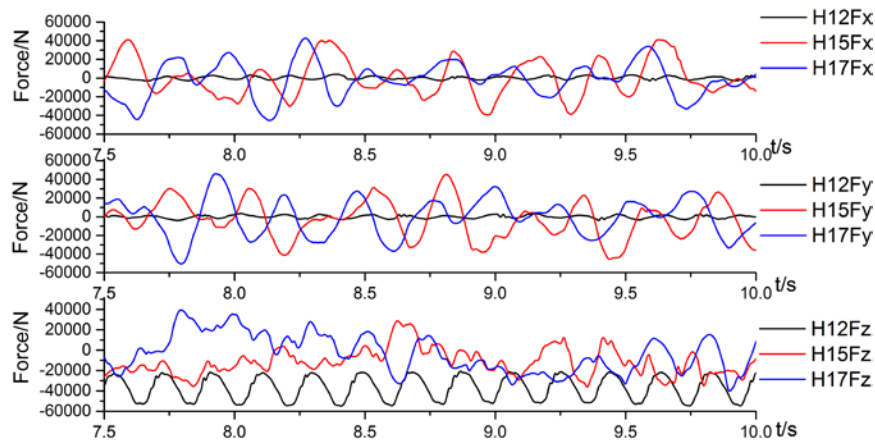


Figure 10. Component of runner force under different heads

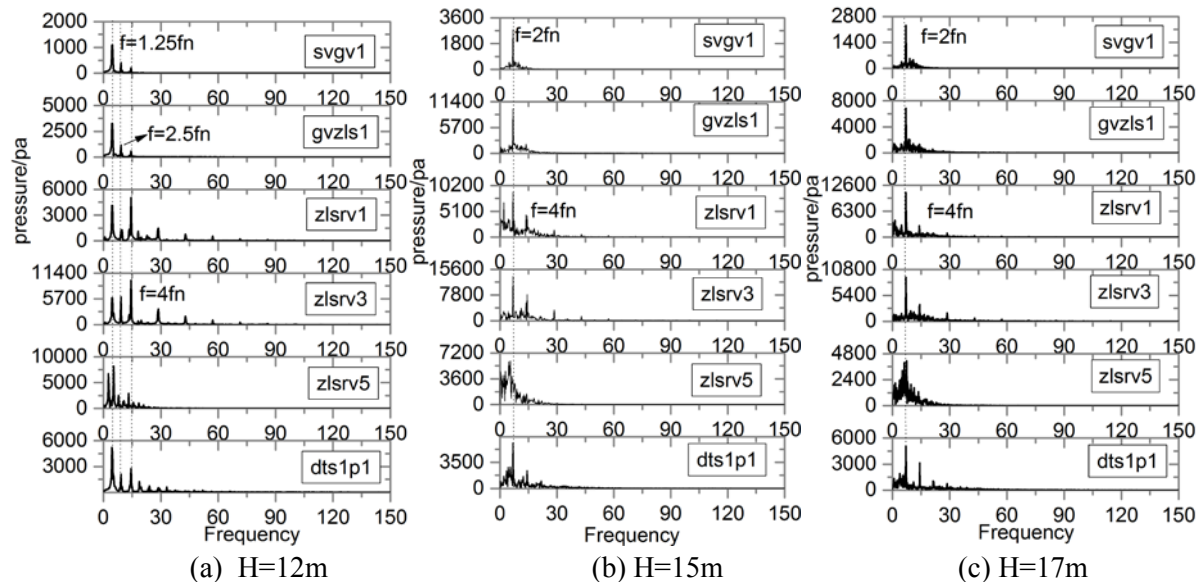


Figure 11. Frequency of pressure pulsation under different heads

According to the frequency chart, there is a significant distinction among the frequency at different locations. In the guide vane region only exist a single frequency, which is rotation frequency or double frequency. Frequency in the runner region are complicated and have multi frequency, which indicating the runner rotating effect on the unstable flow. Frequency in the draft tube appears a little more complex than that in the guide vane but the amplitude drops drastically than that in the runner. Besides the differences in the amplitude, frequency component varies at different heads. At 12m head, the main frequency is 1.25 times and 4 times rotation frequency. While the main frequency increases at the others, the frequency is 2 times or 4 times rotation frequency at 15m and 17m heads. Through the comparison, the amplitude of the frequency at 15m head is still the largest, which explains the emergence of strong vibration.

4. Conclusions

Through the numerical calculation and detailed analysis of no-load conditions at different heads, the following conclusions can be drawn:

(1) There are obvious distorted flow and vortex in the runner when the unit performs no-load operation. With the opening increasing, the stability of the flow field gradually improves at rated head. Thus during the starting up, it is feasible to reduce the operation time of no-load condition and achieve fast grid-connected in order to alleviate the vibration.

(2) With the increase of the head, the opening of no-load operation decrease, but performance instability strengthens as a result of low discharge and high centrifugal force. Comparison with the results based on the unsteady analysis, it indicates that instability of no-load operation is lowest at the minimum head, secondly at the maximum head and highest at the mean head, which is consistent well with the actual situation of power plant. And it is inferred that the instability of no-load operation at different heads may be the result of combined action of centrifugal force and flow rate decrease.

Acknowledgements

The authors thank the National High Technology Research and Open Research Fund Program of State Key Laboratory of Hydrosience and Engineering (No. sklhse-2012-E-01) for supporting present work.

References

- [1] Sallaberger M, Gentner C, Widmer C, Henggeler U, 2012, Stability of Pump Turbine-A Challenge Faced in Design, *Northwest Hydropower*, supplement 1:86-90.
- [2] Casartelli E, Widmer C, Ledergerber N et al. 2010, Simplified CFD model for flow field investigations at no-load conditions. *J. Proceedings of Hydro*, 27-29.
- [3] Li J, Yu J, Wu Y. 2010, 3D unsteady turbulent simulations of transients of the Francis turbine, *25th IAHR Symposium on Hydraulic Machinery and Systems*, Timisoara, Romania.
- [4] Liang Q W, Keller M, 2010, Behaviour of Pump Turbines Operating at Speed no Load Condition in Turbine Mode. *J. Proceedings of Hydro Vision*, 27-39.
- [5] Li Y H, 1995, A few questios of aerating and vibration reduction in axial flow turbine with fixed propeller at Hongshi power station, *Water Resources and Hydropower Engineering*, 3:19-22.
- [6] Sun H M, Zhu Y X, Han Z X, 2001, Improvement of Unit 1 low head no-load stability in Tianhuangping Pumped Storage Power Station, *Hydroelectric power*, 6:60-63.