

PAPER • OPEN ACCESS

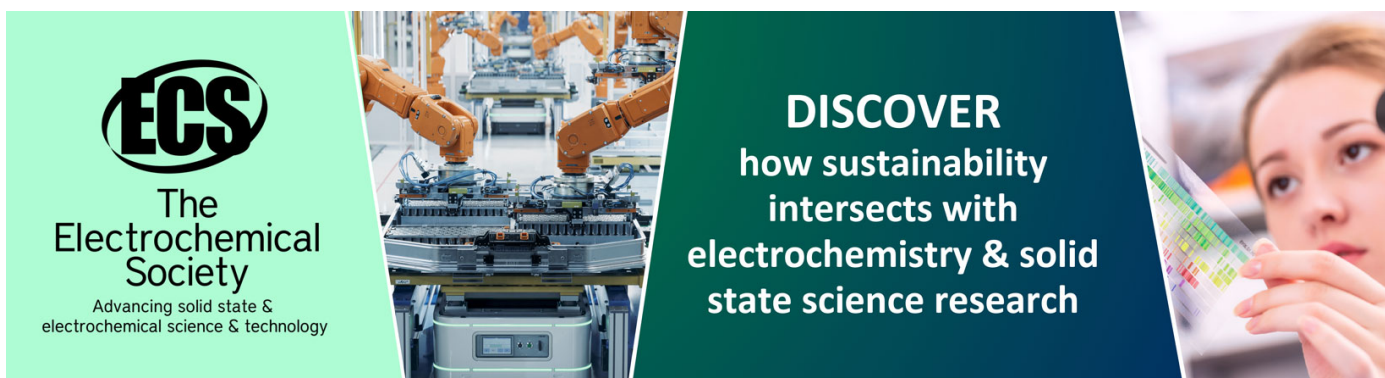
Pressure fluctuation test and vortex observation in Francis turbines draft tube

To cite this article: L Zhu *et al* 2019 *IOP Conf. Ser.: Earth Environ. Sci.* **240** 022029

View the [article online](#) for updates and enhancements.

You may also like

- [Study of the effects of modified draft tube with inclined conical diffuser on draft tube and upstream region](#)
Xing Zhou, Quanshui Huang, Xinyi Hu et al.
- [Numerical Investigation of Pressure Fluctuation and Cavitation inside a Francis Turbine Draft Tube with Air Admission through a Fin](#)
Ruizhi Zhang, An Yu, Michihiro Nishi et al.
- [Investigation on pressure fluctuation in a Francis turbine with improvement measures](#)
J J Feng, W F Li, H Wu et al.



ECS
The
Electrochemical
Society
Advancing solid state &
electrochemical science & technology

DISCOVER
how sustainability
intersects with
electrochemistry & solid
state science research

Pressure fluctuation test and vortex observation in Francis turbines draft tube

L Zhu¹, X C Meng¹, J G Zhang¹, Y Chen¹, H P Zhang¹, WP Wang¹ and L Lu¹

1. Laboratory of Hydraulic Machinery, China Institute of Water Resources and Hydropower Research, Beijing (100038), China

zhulei@iwhr.com

Abstract. The pressure fluctuations in draft tube causes serious vibration and instability, which may affect the safe operation of turbines. In this paper, the pressure fluctuation of hydraulic turbine is studied by means of model test. Two model turbines with specific speed N_{QE0} of 0.1221 and 0.1178 are selected for experimental research, and pressure fluctuation measurement and vortex observation are carried out on the universal hydraulic machinery test stand under typical part-load and overload operating conditions. The influence of Reynolds number and Thoma number on the draft tube vortex was verified by adjusting the test head H_M and tail water pressure p_{DT} on the model test stand. The high-speed photography technique was applied to observe the vortex in the draft tube, meanwhile, the pressure fluctuation signal of the vortex was acquired and analysed. The peak-to-peak amplitude and the frequency of the pressure fluctuation are analysed and compared with the vortex zone morphology. The results show that, 1) the influence of Reynolds number on pressure fluctuation can generally be neglected in the range of Reynolds number $Re > 4 \times 10^6$; 2) pressure fluctuation may decrease with the Thoma number before critical cavitation in special overload conditions, indicating that the impact of cavitation on fluctuation is related to the shape of vortex rope; 3) resonance frequency is obtained at special part-load point, the resonant condition should be confirmed accurately due to the risk of system resonance.

1. Introduction

Pressure fluctuation is one of the main factors that influence the safe operation of hydraulic machinery, because it may cause strong vibration and instability. For special conditions, the turbine needs to operate under off-design conditions, under which the internal water cannot flow smoothly along the flow passage. The flow impacts the hydraulic components, as a result creates unfavorable vortex.

Many studies have shown that the pressure fluctuation in the vaneless area has great relationship with the rotor-stator interaction (RSI) [1], and the pressure fluctuation in the draft tube originates from the rotating vortex rope [2]. The shape of the vortex is determined by the output (or discharge) of the turbine, which affects the characteristics of the pressure fluctuation. Magnoli M V divides the hydro-turbine vortex into 4 types according to turbine output, that is channel vortex, vortex rope, rope free zone and torch zone, see reference 3.

The most typical case is helical vortex rope at runner outlet in draft tube when the turbine operates at upper part load ($Q < Q_{BEP}$). Nishi M, Nicolet C, et al, have pointed out the vortex rope rotates with a precession frequency of 0.2~0.4 times the runner frequency f_n [4-6]. Gao Z obtained the frequency of 1/3.75~1/3.51 the f_n in simulations for vortex rope induced pressure pulsations [7].

Conversely, when the turbine runs at overload operating point ($Q > Q_{BEP}$), a vortex rope rotating opposite the runner is developed in draft tube. For a reversible pumped-turbine, this type of rope



produces self-excited pressure oscillations and brings power swings [8-9]. Dörfler P K studied the full-load surge mechanism through 2-phase CFD simulation [10], Müller A adopted PIV, LDV and high-speed visualizations to observe full load pressure surge in Francis turbines, capturing characteristic breathing motion of the cavitation vortex rope [11].

In this paper, the draft tube vortex and pressure fluctuation of two Francis turbine is studied by means of model test. Two model turbines with different specific speed were selected to reveal the factors that influence draft tube pressure fluctuation and vortex.

2. Methodology

2.1. Hydraulic circuit

The experimental research is carried out on the Universal Test Rig for Hydraulic Machinery (UTR-HM). The UTR-HM is a closed-circuit system, with the capacity of the test head $>100\text{m}$ and the discharge $>2.0\text{m}^3/\text{s}$. As shown in figure 1, the test system consists of low-pressure tank, high-pressure tank, main pumps, dynamometer, electromagnetic flowmeter, and model turbine. The model turbine is installed between the high-pressure and low-pressure tanks, coupled with the speed regulating dynamometer. In order to adjust the system pressure, a pressure regulating system which provides pressurized air, vacuumed air and ambient air is connected to the low-pressure tank. High precision transducers and systems are adopted to acquire data, and the comprehensive uncertainty of the efficiency test is less than 0.2%.

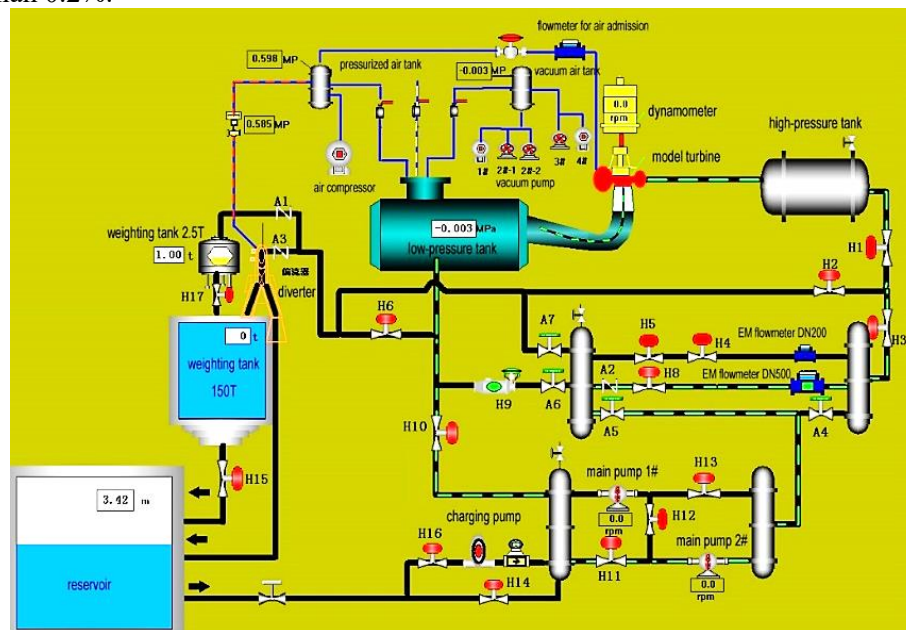


Figure 1. Layout of the Universal Test Rig for Hydraulic Machinery (UTR-HM).

2.2. Pressure fluctuation test

2.2.1. Location of pressure fluctuation measuring taps. Seven dynamic pressure transducers are used to acquire pressure fluctuation signals. The location of pressure fluctuation measuring taps are as following: one tap at the inlet of spiral casing (called HC), two taps at the up-stream and down-stream of the vaneless area between GV and Runner respectively (HSV1 and HSV2), two taps at the up-stream and down-stream of the cone section of draft tube (HD1 and HD2), and two taps at the up-stream and down-stream of the elbow of draft tube (HD3 and HD4), as shown in figure 2(a).

2.2.2. Installation of dynamic transducer. The pressure fluctuation transducer is mounted to the model turbine with insulating transducer base. During the installing of the sensor, the contact surface of the transducer is flush with the flow passage wall without convex or concave to avoid the disturbing phenomenon, as shown in figure 2(b). Before the test, transducers are calibrated.

2.2.3. Data acquisition and processing. The pressure fluctuation data acquisition system is composed of transducer, signal conditioner, high-speed data acquisition module and data analysis system, see figure 2(c). The sampling time of pressure fluctuation test is not less than 10 seconds and the sampling frequency is 2560 Hz. The pulse amplitude and frequency are obtained by analysing the pressure fluctuation signals, in which the amplitude of the pressure fluctuation ΔH is the peak-to-peak value of the mixing signal with a 97% confidence intervals, and the frequency f_1 is the first-order frequency component obtained by the Fast Fourier Transform (FFT) analysis.

2.3. High-speed camera

The cone section of draft tube is made of transparent plexiglass on the purpose of providing an optical access to the flow. A Phantom V641 series high speed camera, with a 4 megapixel (2560×1600 pixel) CMOS sensor, full-resolution frame rates of 1450 frames-per-second (fps) and $1 \mu\text{s}$ minimum exposure is employed to record the vortex rope behaviors. The high-speed camera is arranged on the side of the cone tube, with a 30-degree angle to the horizontal direction, lighting with 2 xenon lamps.

The filming speed of imaging is 1200~1800fps, and the exposure time is $100\mu\text{s}$ ~ $200\mu\text{s}$. No less than 5 rounds are taken at each working point. The typical period of the vortex rope is selected and analyzed after the film is finished.

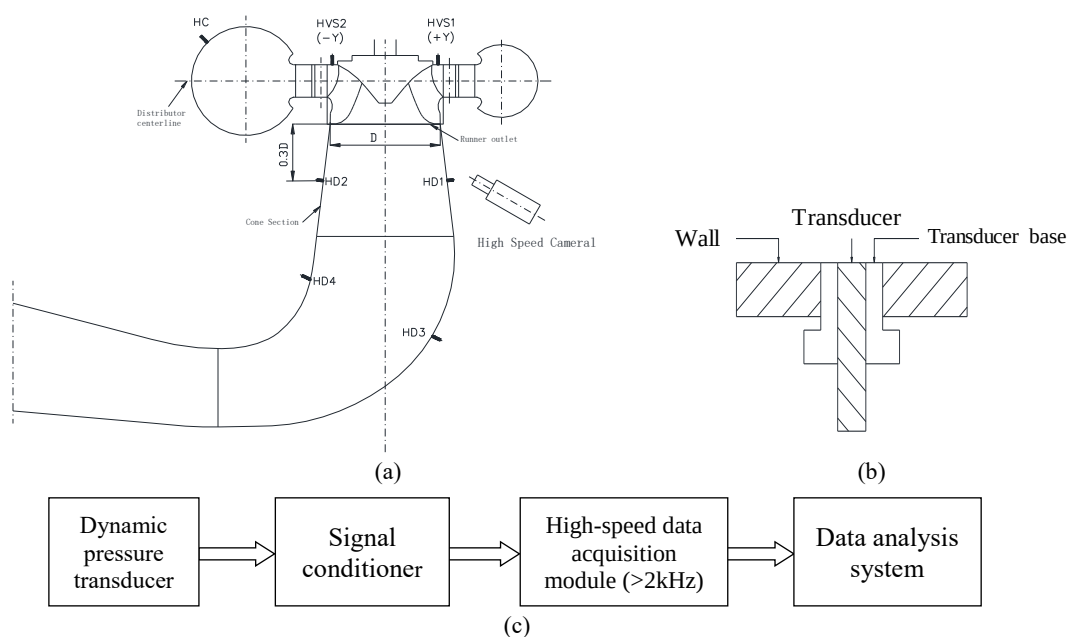


Figure 2. Schematic diagram of dynamic pressure transducer installation and data acquisition.

3. Model turbines and operating condition

3.1. Model turbines

Two Francis turbines with specific speed N_{QE0} of 0.1221 and 0.1178 respectively were experimentally studied. Both of the two turbine runners have 15 blades with 15 splitters. The diameter of runner outlet is larger than 350mm, satisfying the IEC 60193 requirement that the Reynolds number of model tests

Re should be greater than 4.0×10^6 . The parameters of the tested model turbines are shown in table 1, the definition of speed factor n_{ED} , discharge factor Q_{ED} and specific speed N_{QE0} are as following:

$$n_{ED} = nD / (gH)^{0.5} \quad (1)$$

$$Q_{ED} = Q / [D^2 (gH)^{0.5}] \quad (2)$$

$$N_{QE} = n_{ED} \cdot Q_{ED}^{0.5} \quad (3)$$

3.2. Operating condtions

One part-load and one overload operating conditions closed to the rated head were selected to study the draft tube vortex and pressure fluctuation, and the test conditions are shown in table 2, where the n_{ED0} , Q_{ED0} and P_0 are the corresponding values of the design conditions.

The Reynolds number **Re** and Thoma number σ are regulated by controlling the test head and pressure of the vacuum tank during the test. The Reynolds number **Re** is between $3.5 \times 10^6 \sim 5.5 \times 10^6$ and the Thoma number σ in the range of 0.06 and 0.26. In each group of experiments, the operating points are tested at 3~4 points within the above Reynolds number and the Thoma number range respectively.

The system resonance phenomena between model turbine and test system is likely to appear when the hydro-turbine is operating under special part-load conditions. In order to study the impact of resonant condition on fluctuation, some potential resonant condition, namely the **PNT. B1** and **B2** test points of **Turbine B**, is choosen to research the resonant draft tube vortex.

Table1. Parameters of tested model turbines.

Parameter	Turbine A	Turbine B
Type	Francis	Francis
Specific speed N_{QE0}	0.1221	0.1178
Runner blade number Z_R	15+15	15+15
Guide Vane number Z_{GV}	24	23

Table2. Model test operating conditions.

Turbine A				Turbine B			
PNT.	n_{ED}/n_{ED0}	Q_{ED}/Q_{ED0}	P/P_0	PNT	n_{ED}/n_{ED0}	Q_{ED}/Q_{ED0}	P/P_0
A1	0.98	0.82	0.85	B1	1.08	0.87	0.68
A2	1.01	1.28	1.22	B2	1.02	0.81	0.75

4. Result and analysis

4.1. Repeatability of test results

First, the repeatability verification of pressure fluctuation test is carried out. The results of pressure fluctuation test at A1 and A2 operating point under different Reynolds number at $\sigma = 0.13$ were selected and the test curves are shown in figure 3. The longitudinal axis of the coordinate is the relative amplitude of pressure fluctuation $\Delta H/H$ with 97% confidence interval, and the horizontal axis is the Reynolds number.

It can be seen from the graph that the results of the independent 2 pressure fluctuation tests at the HD location in the cone tube are well agreed. At the corresponding Reynolds number, the maximum difference of $\Delta H/H$ in the partial-load condition (**PNT. A1**) is less than 0.3%, and that in overload condition (**PNT. A2**) is about 0.2%, both of them are within acceptable range.

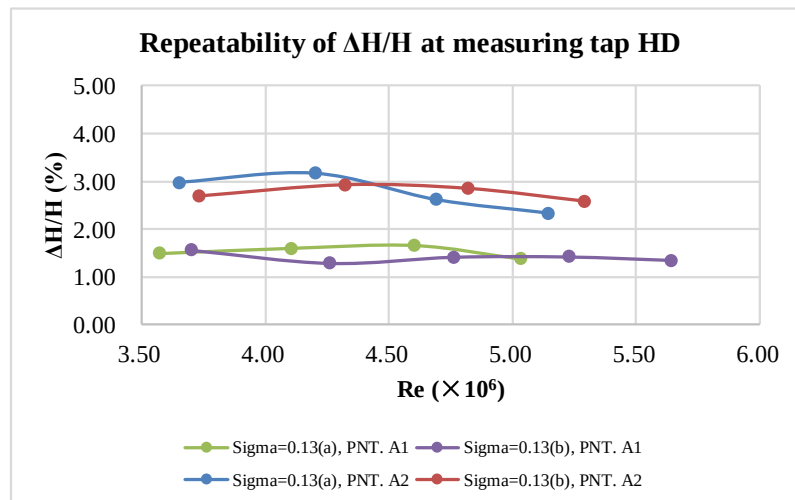
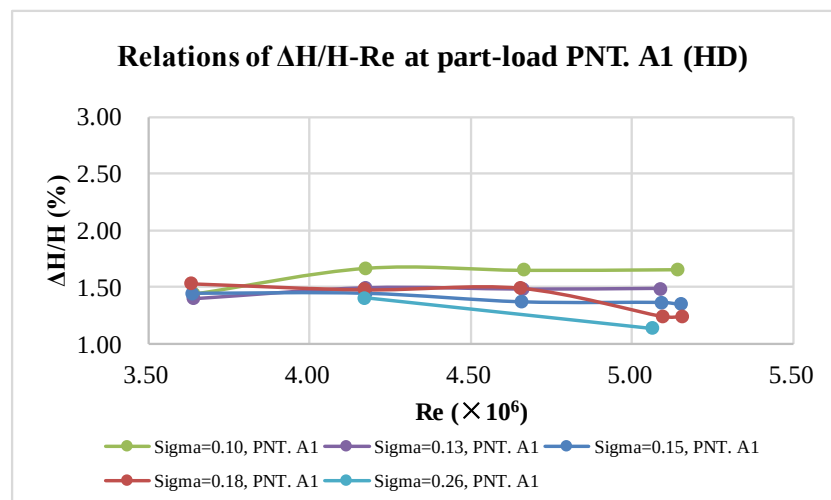


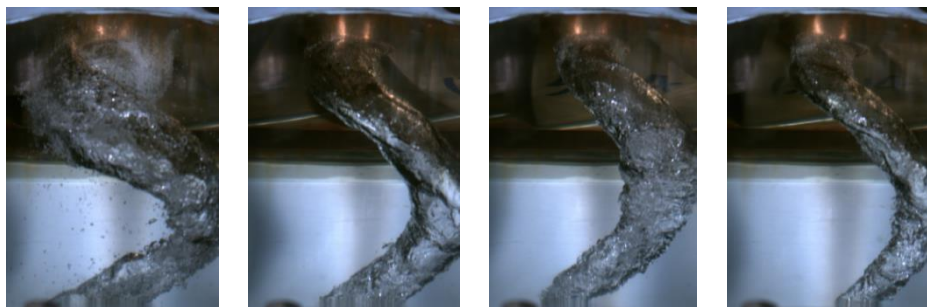
Figure 3. Repeatability check of pressure fluctuation test.

4.2. Part-load operating condition

The relationship between the relative amplitude of pressure fluctuation and Reynolds number in different Thoma numbers under part-load conditions is shown in figure 4. When σ equals to 0.10, 0.13, 0.15, 0.18 and 0.26, the pressure fluctuation of PNT. A1 is tested at 4 Reynolds number respectively, and the result of the measuring tap HD on the cone tube is given.



(a)



PNT. A1($\sigma = 0.13$), from left to right $Re = 3.6, 4.2, 4.7, 5.0 \times 10^6$

(b)

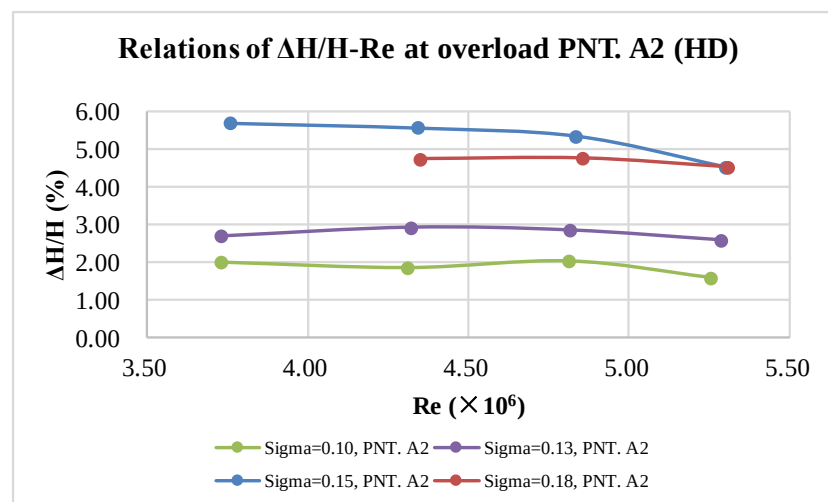
Figure 4. Pressure fluctuation and draft tube vortex at part-load point.

4.2.1. The influence of Reynold number. It can be seen from figure 4 (a) that, under the given Thoma number within the range of the test Reynolds number $(3.5\sim6.0) \times 10^6$, the maximum difference is not more than 0.5%, that means the Reynolds number has little effect on the pressure fluctuation. Compare the draft tube vortex of PNT. A1 at different Reynold number in condition of $\sigma = 0.13$, as is shown in figure 4 (b). It can be seen from the photos that the shape of the vortex rope in the above Reynolds number range are basically the same. From the comparison of pressure fluctuation and vortex rope shape, it can be seen that the amplitude of pressure fluctuation in the normal par-load conditions is nearly unaffected by Reynolds number when Re satisfy the conditions specified in IEC 60193 ($Re > 4.0 \times 10^6$).

4.2.2. The influence of Thoma number. The pressure fluctuation amplitude of different Thoma numbers is compared at the same Reynolds number, as shown in figure 4 (a). As can be seen from the graph, amplitude of the pressure fluctuation increases slightly when σ value has a decreasing tendency. This is because the cavitation may occur at the runner outlet if Thoma number becomes smaller, which may disturb the flow in draft tube. The flow pattern in the turbine is deteriorating before the critical cavitation is reached, so the amplitude of pressure fluctuation may increase, but it is not obvious.

4.3. Overload operating condition

The relationship between the relative amplitude of pressure fluctuation and Reynolds number in different Thoma numbers under overload conditions is shown in figure 5. When σ equals to 0.10, 0.13, 0.15 and 0.18, the pressure fluctuation of PNT. A2 is tested at 4 Reynolds number respectively, and the result of the measuring tap HD on the cone tube is given.



(a)

PNT. A2 ($Re=4.3$), from left to right $\sigma=0.10, 0.13, 0.15, 0.18$

(b)

Figure 5. Pressure fluctuation and draft tube vortex at overload point.

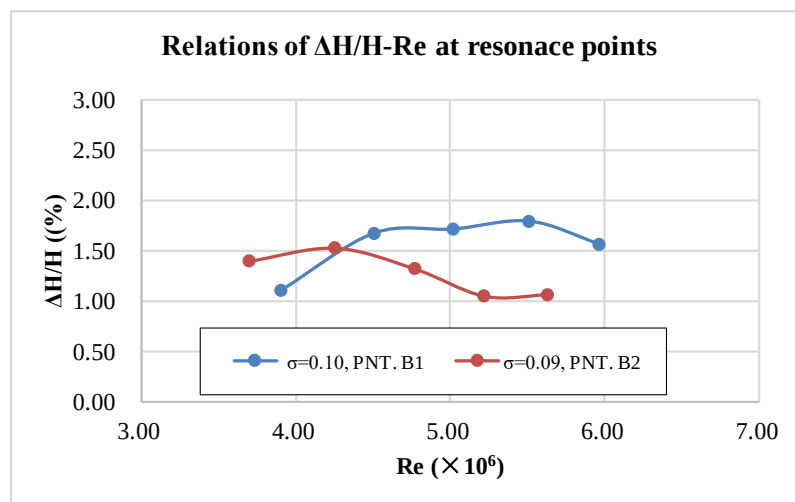
4.3.1. The influence of Reynold number. It can be seen from figure 5 (a) that, the Reynolds number has little effect on the pressure fluctuation under the given Thoma number, which is similar to part-load operating conditions. The maximum difference of pressure fluctuation amplitude is less than 0.4% except $\sigma=0.15$ (up to 1.0%).

4.3.2. The influence of Thoma number. But the σ has a great influence on the pressure fluctuation in the overload condition. As shown in figure 5 (a), comparing the amplitude of pressure fluctuation of different Thoma numbers under the same Reynolds number, we can see that, a) the amplitude difference between the pressure fluctuation of $\sigma=0.10$ and $\sigma=0.15$ at the same Reynolds number is obvious, the maximum difference reaches 3.65%; b) when the value of σ increases, the amplitude of pressure fluctuation increases, which is in contrast to partial load conditions.

Comparison of the appearance of vortex rope under different Thoma number of PNT. A2 at $Re=4.3 \times 10^6$ is shown in figure 5 (b). As can be seen from the diagram, in the case $\sigma=0.10$ and 0.13, the regular-shape vortex rope occupies large volume of cavities, resulting in overload surge, but the rope is basically not contact with the wall of the draft tube. Conversely, in the case that $\sigma=0.15$ and 0.18, the cavities of vortex rope are not fully formed, and the dispersed vortex in the downstream sweeps to the wall of draft tube. Therefore, the large amplitude of the pressure fluctuation is measured when σ value is large. The characteristics of the pressure fluctuation are highly related to the shape of the vortex rope.

4.4. Influence of resonance frequency

If the frequency of the vortex rope is close to the natural frequency of the hydraulic circuit system, the resonance can be generated when the turbine is running under part load condition. Two resonant conditions of Turbine B are selected, with $n_{ED}/n_{ED0}=1.08$ and 1.02 respectively, see figure 6.



(a)



PNT. B1 ($\sigma = 0.10$), from left to right $Re=3.9, 5.0, 6.0$

(b)

Figure 6. Pressure fluctuation and draft tube vortex at resonance point.

Figure 6 (a) gives the relation curves of pressure fluctuation amplitude with Reynolds number at B1 and B2 points. Different from the non-resonant operating conditions under which the pressure fluctuation is basically independent of the Reynolds number, the pressure fluctuation in these two conditions is obviously fluctuating with Reynolds number, and there is no uniformity. Figure 6 (b) gives photos of vortex rope in draft tube of B1 operating condition. It is very strange that the shape of vortex rope at different Reynolds numbers are quite similar to each other, and do not show much difference like the fluctuation curves.

The cause of these anomalies can be explained by analyzing the pressure fluctuation signal. The first 3-order frequency of pressure fluctuation is obtained by analyzing fluctuation signal. Firstly, define f/f_n as the ratio between pressure fluctuation frequency and rotating frequency. Table 3 gives the first-order frequency component of pressure fluctuation f_1/f_n at all measuring taps. Generally, the ratio f/f_n should be the multiple of the runner blade number under normal (non-resonant) conditions. However, the testing points basically do not satisfy this rule except No.8 point in this test. These frequencies with similar values at all measuring taps are likely to be the resonant frequencies.

The resonance frequency is an important risk for the turbine system, once the excitation energy exceeds critical value. It can cause unsteady amplitude of pressure fluctuation, which is unfavorable to measure pressure fluctuation. Unfortunately, the part-load resonance frequency is existent, and even unavoidable. This frequency of model turbine is likely to appear in prototype turbine. Therefore, the resonant condition should be confirmed accurately due to the risk of system resonance.

Table3. First order frequency of pressure fluctuation f_1/f_n .

No.	Re ($\times 10^6$)	Operating condition	HC	HVS1	HVS2	HD1	HD2	HD3	HD4
1	3.90	PNT. B1 $\sigma=0.10$	6.83	0.85	1.00	0.85	0.85	0.85	0.85
2	4.51		0.70	3.24	3.24	3.57	3.57	3.24	0.70
3	5.02		3.39	3.39	3.39	3.63	3.63	3.39	3.39
4	5.51		3.27	3.27	3.27	3.50	3.50	3.26	3.26
5	5.97		3.06	3.06	3.06	3.30	3.06	3.06	3.06
6	3.70	PNT. B2 $\sigma=0.09$	7.25	0.69	0.69	0.69	0.69	0.69	0.69
7	4.25		2.49	2.49	2.49	2.73	2.73	2.49	2.49
8	4.77		5.62	29.98	29.98	0.60	0.60	0.60	0.60
9	5.22		5.13	2.50	2.50	0.23	0.23	2.50	0.46
10	5.63		4.75	2.59	2.59	0.23	0.23	2.59	0.46

5. Conclusions

The draft tube vortex and pressure fluctuation of 2 Francis turbines is studied in this paper, which comes to the following conclusions on the previous results:

(1) In the range of Reynolds number $Re > 4 \times 10^6$, the influence of Reynolds number on pressure fluctuation can be neglected, so the model test can accurately measure the pressure fluctuation in normal conditions.

(2) The amplitude of pressure fluctuation decreases with the increase of Thoma number under normal conditions. However, in some overload conditions, before the critical cavitation is reached, the regular shape of the cavity generates at small Thoma number, resulting in smaller pressure fluctuation. Therefore, the impact of cavitation on fluctuation is related to the shape of vortex rope.

(3) The resonance frequency is unfavourable to the measurement of pressure fluctuation. However, because the resonance frequency is an important risk for the turbine system, the resonant condition should be confirmed accurately.

Acknowledgements

This research is funded by the Major National Scientific Instrument and Equipment Development Project (Project No. 2011YQ07004901), the IWHR Research & Development Support Program (HM0163A012018) and the BITC R&D Program (SY-15-ZY-01, SY-16-ZY-10 and SY-17-ZY-14). The authors would like to thank China Institute of Water Resources and Hydropower Research (IWHR) and BITC Company for their financial support to the project.

References

- [1] Rodriguez C G, Mateosprieto B and Egusquiza E 2014 *Shock and Vibration* (2014) 276796
- [2] Kirschner O, Ruprecht A and Göde E 2012 *IOP Conf. Ser.: Earth Environ. Sci.* **15** (2012) 062059
- [3] Magnoli M V and Maiwald M 2014 *IOP Conf. Ser.: Earth Environ. Sci.* **22** (2014) 032013
- [4] Nishi M, Kubota T and Matsunaga S 1980 *IAHR/AIRH 10th Symp. 10E, Hydraulic Machinery and Equipment Associated with Energy Systems in the New Decade of the 1980's* Tokyo 557-568
- [5] Nishi M and Liu S 2013 *Int. J. Fluid Machinery & Systems* IJFMS **6(1)** 033
- [6] Nicolet C, Zobeiri A and Maruzewski P 2011 *Int. J. Fluid Machinery & Systems* IJFMS **4(1)** 179
- [7] Gao Z 2009 *J. Hydraulic Engineering* **40(10)** 1162-1167 (in Chinese)
- [8] Koutnik J, Nicolet C and Schohl G A 2006 *Proc. 23rd IAHR Symp. Hydraulic Machinery and Systems* Yokohama 1-14
- [9] Rodriguez D, Rivetti A and Lucino C 2016 *IOP Conf. Ser.: Earth Environ. Sci.* **49** 082006
- [10] Dörfler P K, Keller M and Braun O 2010 *IOP Conf. Ser.: Earth Environ. Sci.* **12** 012026.
- [11] Müller A, Favrel A and Landry C. 2017 *J. Phys.: Conf. Ser.* **813** 012018