An important analysis on the aerofoil optimization for hydraulic performance improvement of pump-turbine

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Abstract. One of domestic pumped storage power station has been put into operation, installed four single capacities of 250MW single-stage mixed flow constant speed reversible pump-turbine set. After put into operation, it was found that there were an obvious "S" shaped area and hydraulic problem of large pressure pulsation in vaneless area of pump turbine, which had a great negative impact on the long-term safe and stable operation of the equipment. In order to solve this hydraulic problem, this paper is based on the actual situation of the power station, under without changing the flow passage from the volute to the draft tube condition, optimize the runner, the other components are not modified scheme A and optimize the movable guide vane and runner two flow passage parts in the same time, other parts do not make the transformation of the B scheme. Through the comparison and analysis between the A scheme, the B scheme and the existing power plant model test results, it is found that the A scheme and the B scheme have their own advantages in terms of the energy characteristics, cavitation performance, hump characteristics and other aspects of the pump-turbine, respectively. At the same time to optimize the movable guide vanes and the runner B scheme can achieve completely improve the characteristics of the S shaped region, and the unit can safely and stably connect to the grid without taking asynchronous guide vanes in the whole operating range; A and B schemes have obviously improved the pressure pulsation between the guide vanes and the runner of the turbine working conditions, while the B scheme is more prominent at this point.

1.Introduction

One of domestic pumped storage power station has been put into operation, installed four single capacities of 250MW single-stage mixed flow constant speed reversible pump-turbine set. The high pressure pipe uses a tube of two machine layout, divided into vertical and horizontal sections, two parallel arrangements. The tailrace tunnel is arranged by a single machine in a single hole. The rotational speed of pump-turbine is 333.3r/min, maximum head/hydraulic head for 346m, the smallest head/hydraulic head for 291m, the rated head is 305m, the turbine working condition rated output is 255MW, rated frequency is 50Hz, the normal range of frequency is 49.8~50.4Hz. The pump-turbine runner inlet diameter D_{p1}=4.641m, outlet diameter D_{p2}=2.55m, the number of runner blades is 9, the number of fixed guide vanes and movable guide vanes is 20, the guide vane circle diameter D_0=5.43m, the ratio of guide vane distribution guide diameter D_0 and wheel diameter of D_{p1} was 1.169.

Since put into operation, it has been found that there are two major hydraulic problems in the hydraulic performance of the pump-turbine. First, There is a clear "S" shaped region, resulting in the turbine when it operating within the scope of less than 320m head no-load connected to grid difficult, must be put into the guide vane to smoothly grid-connection, and the use of complex and conventional guide vane structure of the asynchronous guide vane structure will give the operation and maintenance of the unit to bring security risks. Second, the vibration and noise of the plant are strong. Through many tests, research and analysis, it is found that due to a large pulse caused by turbine working conditions in vaneless area. These two hydraulic problems have great adverse effects on the long-term safe and stable operation of the equipment. This topic is based on the actual situation of the power station, under the premise of without changing the flow-passage components from the volute to the draft tube, ensuring that the turbine and the pump working conditions operating range do not shift, On the basis of ensuring that the pump-turbine energy characteristics, cavitation and other hydraulic properties maintain the existing level, through change the activities of guide vanes and runner transformation methods to achieve the purpose of improving the above hydraulic problems.

On the basis of existing equipment, transformation of pump-turbine hydraulic problems, compared to the new power plant on a whole hydraulic design, there are more restrictions conditions. Must be in the premise of not changing from the volute to draft tube flow passage components, only optimize guide vane and runner to improve the shape of "S" region and the turbine working conditions vaneless area of pressure pulsation is the main difficulties faced by this topic [1-3].

2.Optimization scheme

In this paper, CFD numerical analysis technology and pump-turbine model test combination method are used to carry on the thorough research. First, the CFD numerical analysis technique is used to optimize the hydraulic design of the pump-turbine. Under the condition of achieving the optimization target, make the pump-turbine model accordingly to verify the hydraulic characteristics such as the energy characteristics, cavitation characteristics, flying characteristics and pressure pulsation characteristics of the pump-turbine.

Based on the existing problems of the power station, the main goal of hydraulic optimization is under the premise of ensuring the turbine working conditions and pump working conditions operation range do not shift and on the basis of ensuring pump-turbine energy characteristics, cavitation and other hydraulic performance to improve in two aspects: (1) pump-turbine "S" shaped area characteristics; (2) turbine working conditions vaneless pressure pulsation.

The optimization of this topic is that by increasing the ratio of the diameter of the movable guide vane distribution circle diameter D_0 to the inlet diameter D_{p1}, that is, to maintain the diameter of the movable guide vane distributed circle diameter on the basis of reducing the diameter of the inlet diameter D_{p1}, on this basis, optimize the runner on the crown, the lower ring and the airfoil. According to the optimization idea, this topic has carried out two schemes of hydraulic optimization design, respectively, for scheme A and scheme B, where scenario A is only considering the optimization of the runner, the other components are not modified. The B scheme is to consider both the operational guide vanes and the runner with two flow passage components, and the other components are not retrofitted. And two design schemes of pump-turbine are carried out model test.

3. Comparison and analysis of experimental results of optimization scheme

Here on the best efficiency and the average weighted average efficiency of pump conditions and turbine conditions, the maximum input force and the minimum flow rate and other energy characteristics of pump working condition, "S" area characteristics and turbine pressure pulsating and pump cavitation performance and other major hydraulic performance were compared and analyzed[4].

3.1. Energy characteristics

This topic adopts a two-step conversion method of efficiency，converts the efficiency of the model into the prototype efficiency, and analyzes the energy characteristics. The pump-turbine model efficiency

conversion to prototype efficiency using a two-step efficiency correction formula[错误!未定义书签。], Which is the first step in the efficiency of the model test value is corrected to the reference Reynolds number of efficiency, see equation (1) and (2); the second step is to refer to the Reynolds number of efficiency to the power plant Reynolds number of efficiency, see equation (3) and (4).

$$\Delta\eta_{hMi\to M^*} = \delta_{ref}\left[\left(\frac{\mathrm{Re}_{uref}}{\mathrm{Re}_{uMi}}\right)^{0.16} - \left(\frac{\mathrm{Re}_{uref}}{\mathrm{Re}_{uM^*}}\right)^{0.16}\right] \qquad (1)$$

$$\eta_{hMi*} = \eta_{hMi} + \Delta\eta_{hMi\to M^*} \qquad (2)$$

$$\Delta\eta_{hM^*\to P} = \delta_{ref}\left[\left(\frac{\mathrm{Re}_{uref}}{\mathrm{Re}_{uM^*}}\right)^{0.16} - \left(\frac{\mathrm{Re}_{uref}}{\mathrm{Re}_{uP}}\right)^{0.16}\right] \qquad (3)$$

$$\eta_{hP} = \eta_{hMi*} + \Delta\eta_{hM^*\to P} \qquad (4)$$

$$\delta_{ref} = \left(1-\eta_{hoptM}\right)\div\left[\left(\frac{\mathrm{Re}_{uref}}{\mathrm{Re}_{uoptM}}\right)^{0.16} + \frac{1-V_{ref}}{V_{ref}}\right] \qquad (5)$$

Where：δ_{ref}—conversion loss rate；Re_{uref}—Reynolds number；Re_{uMi}—test point Reynolds number；Re_{uM^*}—the reference Reynolds number；Re_{uP}—Plant station Reynolds number；η_{hMi}—model efficiency；η_{hP}—prototype efficiency；η_{hoptM}—optimal efficiency of model.

3.1.1. Optimal efficiency of pump - turbine working condition. According to the model test, the optimal efficiency of pump working conditions of existing pump-turbine was 91.33%, according to the equation (1), (2), (3), (4), and (5) the two step conversion equation of efficiency, converted to prototype the optimal efficiency of 93.48%; the optimal efficiency of A scheme model is 91.44%, and the prototype model efficiency is 93.58%; the optimal efficiency of B scheme model is 91.37%, and the optimal prototype efficiency is 93.53. According to the model test data, the optimization of scheme A and scheme B can improve the prototype efficiency of pump working conditions by 0.1% and 0.05% respectively, as showed in figure 1.

The optimal efficiency of turbine working condition of existing pump-turbine was 92.54%, converted into optimal efficiency of prototype is 94.86%; the optimal efficiency of A scheme model is 92.62%, and the prototype optimal efficiency is 94.74%; the optimal efficiency of B scheme model is 91.39%, and the optimal prototype efficiency is 93.88. According to the model test data, the optimization of scheme A and scheme B reduced the prototype efficiency of turbine working conditions by 0.12% and 0.98% respectively, as shown in figure 2. But only by virtue of the data and cannot make the turbine working conditions have reduced the efficiency of the conclusion, because the turbine working conditions are the best efficiency point deviation from the turbine operating range.
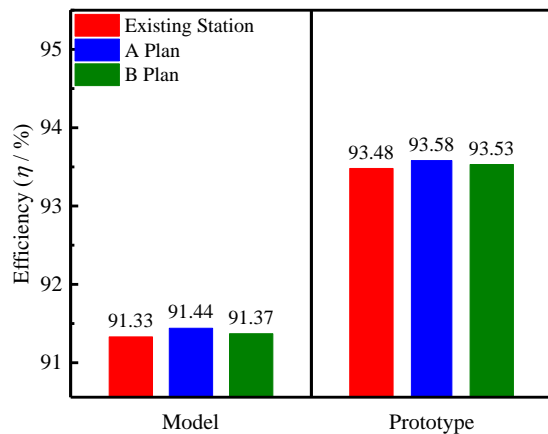
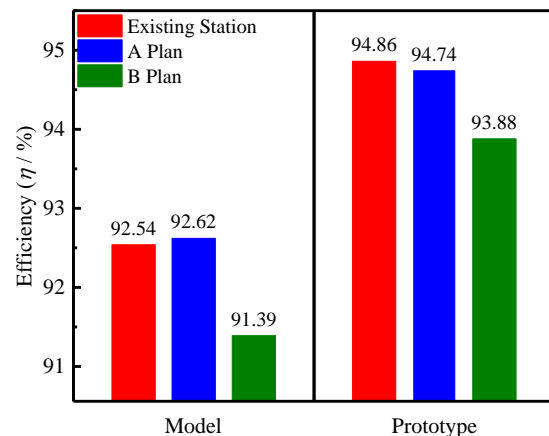**Figure 1.** Comparison of Optimum Efficiency of pump operation condition

Figure 2. Comparison of Optimum Efficiency of turbine operation condition

3.1.2. Weighted average efficiency of pump working condition and turbine working condition. The weighted average efficiency of the pump working conditions and the turbine working conditions is a comprehensive indicator of the efficiency of the whole operating range of the pump operating conditions and the full range of operating conditions of the turbine. It is one of the important indexes in the energy characteristics of the unit.

The weighted average efficiency of pump operation condition is the efficiency of each feature head under the model test is according to the weighting factors in Table 1 and calculate by the formula 6. The weighted average efficiency of the pump model is 91.34% and the weighted average efficiency of the prototype is 93.50%. The weighted average efficiency of the pump operating condition of existing pump-turbine is 91.34%, and the prototype weighted average efficiency is 93.50%; The weighted average efficiency of A scheme is 91.44%, and the prototype weighted average efficiency is 93.58%; The weighted average efficiency of scheme B is 91.27%, and the prototype weighted average efficiency is 93.43%. It can be seen that the prototype weighted average efficiency of scheme A is improved by 0.08%, and the prototype weighted average efficiency of B scheme is reduced by 0.07%. Although the weighted average of the A scheme is slightly improved, the weighted average efficiency of the B scheme is slightly reduced, but because it is not obvious, it can be considered that the two schemes have no obvious change in the improvement of the weighted average efficiency of the pump, as shown in figure 3.

The weighted average efficiency of turbine operation condition is the efficiency of each feature head under the model test is according to the weighting factors in table 1 and calculate by the formula 6. The weighted average efficiency of the turbine operating conditions of existing pump-turbine is 89.83%, the weighted average efficiency of the prototype is 92.15%; The weighted average efficiency of scheme A is 90.09%, and the prototype weighted average efficiency is 92.24%; the weighted average efficiency of scheme B is 88.91%, and the prototype weighted average efficiency is 91.40%. It can be seen that the prototype weighted average efficiency of scheme A is improved by 0.09%, the prototype weighted average efficiency of scheme B is reduced by 0.75%, that is, the scheme A is slightly improved in the weighted average efficiency of turbine condition, but the scheme B is more obvious, as shown in figure 4.

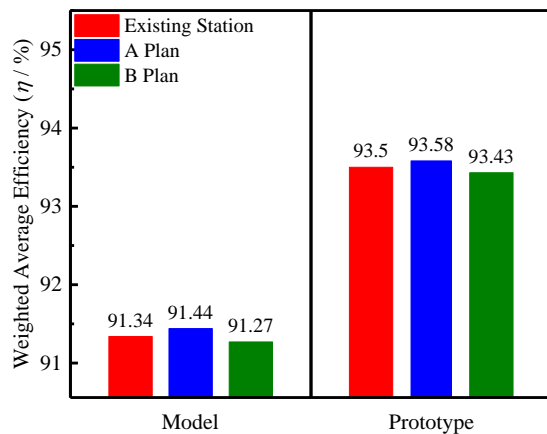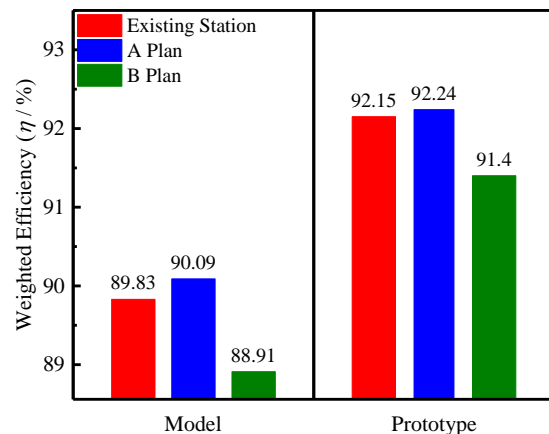**Table 1.** Weighted factor of pump operation condition

Hj(m)	Wj
345	4
335	20
325	28

315	30
305	16
295	2

Table 2. Weighted factor of turbine operation condition

Hj（m）	Wij						
	50%Pr	60%Pr	70%Pr	80%Pr	90%Pr	100%Pr	total
285	0.2	0.3	0.5	1.5	0.5	0	3
295	0.4	2	3.5	7	2.3	0	15.2
305	0.5	4	4.5	8.5	6	5.5	29
315	0.3	3	4	7	8.5	3	25.8
325	0.3	1	3.2	6.5	7	2	20
335	0.2	0.5	0.5	3.2	0.5	0.3	5.2
345	0.2	0.3	0.3	0.5	0.3	0.2	1.8
total	2.1	11.1	16.5	34.2	25.1	11	100

Where Pr is the rated output power(MW) of the turbine operation condition.

Figure 3. Comparison of Weighted Average Efficiency of Pump operation condition

Figure 4. Comparison of Weighted Average Efficiency of turbine operation condition

The weighted average efficiency equation is:

$$\eta_w = \frac{w1 \times \eta1 + w2 \times \eta2 + w3 \times \eta3 + \cdots}{w1 + w2 + w3 + \cdots} \qquad (6)$$

Where: $\eta1$， $\eta2$， $\eta3$ is the efficiency value in the specified operating point, and $w1$， $w2$， $w3$ is the corresponding agreed weighting factor.

3.1.3. Pump condition maximum force and minimum flow. The maximum input force of the prototype pump when the minimum net head of the existing pump-turbine unit is 257.1MW (50.4Hz), the maximum input force of the A scheme is 259.3MW (50.4Hz), and the maximum input force of the B scheme is 269.2MW (50.4Hz). From the results of this test, it can be seen that the maximum input force of the B scheme is 12.1MW higher than that of the existing unit, but still satisfies the guaranteed value of the maximum input power of 270.5MW of the pump condition, as shown in figure 5.
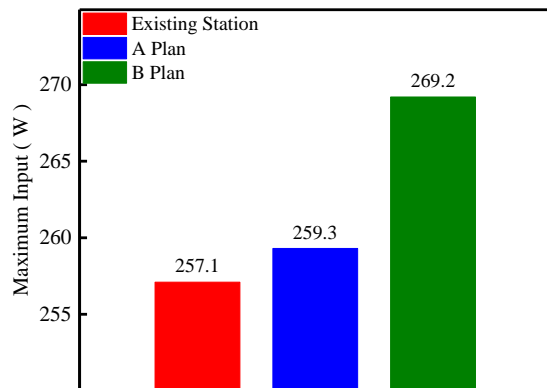

Figure 5. Comparison of Maximum Input Force of Pump operation condition
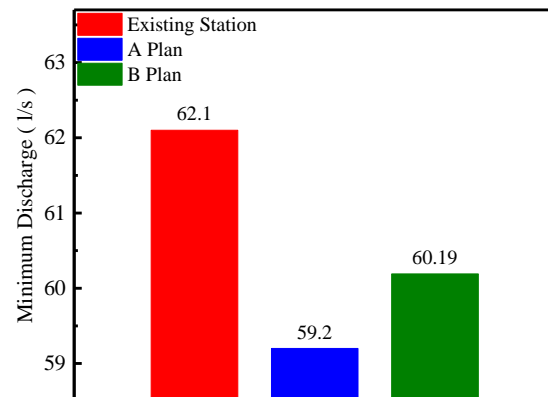

Figure 6. Comparison of minimum flow rate of pump operation condition

The minimum flow rate of the prototype water pump at the maximum net lift of the existing water pump turbine unit is 62.1m3/s, the minimum flow rate of the A scheme is 59.2m3/s, and the minimum flow rate of the B scheme is 60.19m3/s. From the results of this test, the minimum flow rates of schemes A and B are reduced to different degrees, as shown in figure 6.

3.2. "S" area characteristics

"S" characteristic test is one of the most important tests of pump -turbine. One of the important purposes of this test is to verify whether the pump-turbine can be connected smoothly in the no-load condition of the turbine operation condition. At present, the domestic engineering often uses the prototype hydraulic turbine operating range distance "S" area of the safety margin of this performance parameter requirements to determine the pump-turbine in the turbine operation conditions are smooth and connected to the hydraulic characteristics. The method of judging the critical point of the "S" characteristic zone is the four-quadrant characteristic curve of $Q_{11} \sim n_{11}$ (Q_{11} is the unit flow rate, n_{11} is the unit speed), Through the guide vane opening line and the intersection of the runaway curve to do a guide vane opening curve of the tangent, When the angle between the tangent and the N_{11} coordinate is equal to 90 degrees, it is the critical point of the "S" characteristic zone, the difference between the N_{11} value converted to the head and the minimum head that takes into account the normal frequency of the grid is the margin of safety[5-7].
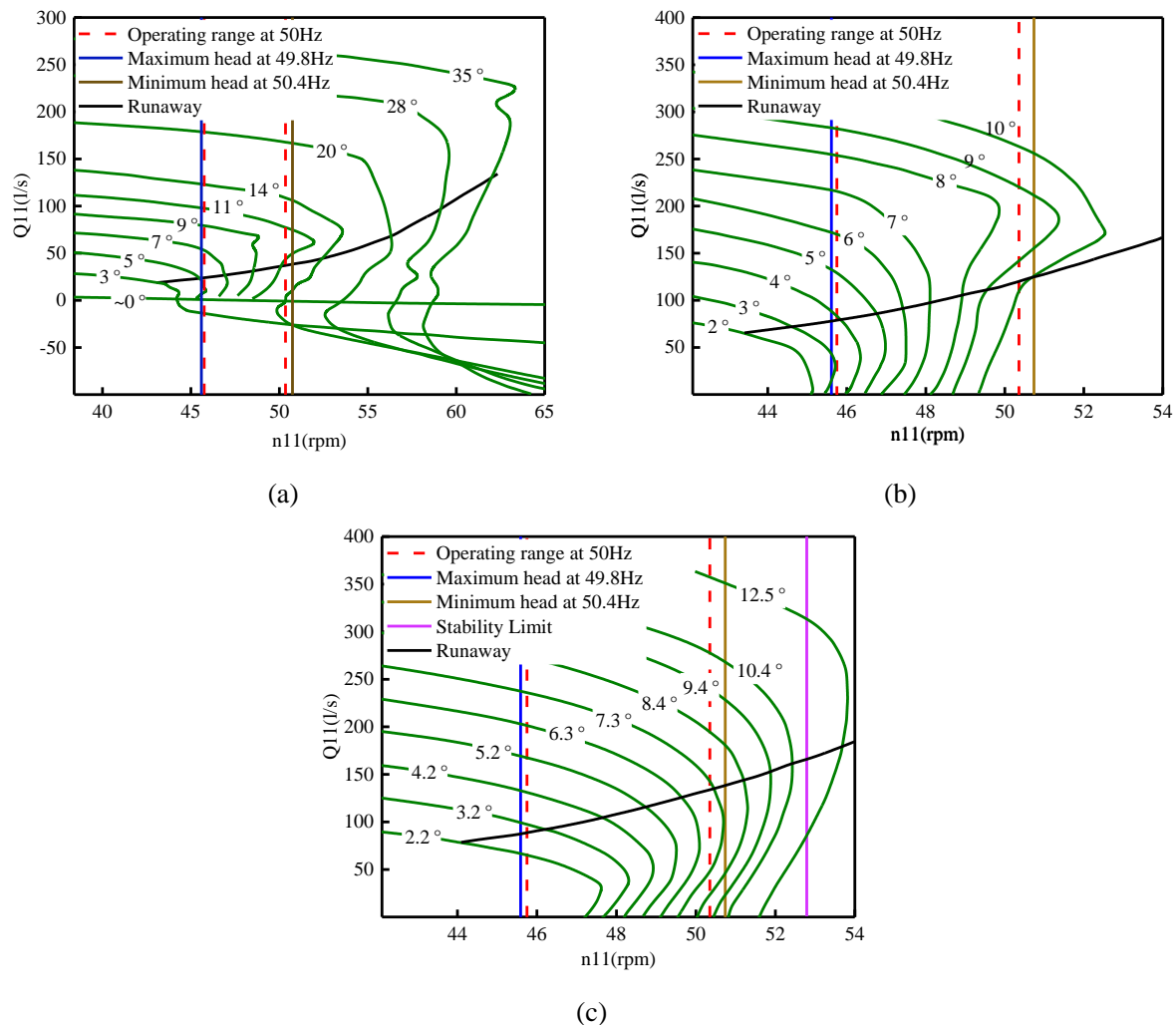
(a)

(b)

(c)

Figure 7. Characteristic curve of "S" for pump-turbine

The characteristic curve of the "S" shape of the existing water pump is shown in figure 7 (a). It can be seen clearly from the graph that there is a very obvious "S" -shaped area in the full characteristic curve of the pump-turbine. In the range of HP = 285m ~ 346m, the turbine no-load opening is a0 = 5.34 ° ~ 11.34 °, the opening range is just in the unstable "S" shaped area. In this area, turbine operation conditions in the no-load or phase transfer operation may enter the anti-pump conditions, to the turbine without load and turbine emergency shutdown and other operations difficult. Need to take appropriate measures to improve the unstable "S" shaped regional characteristics.

In order to solve this problem, two pairs of non-synchronous guide vanes are set up on the real machine to solve the unstable "S" shaped area characteristic by drawing lessons from the experience of the other pump-turbine plant. Based on the analysis of the theoretical basis, it can be seen from figure 7 (a) that there are "S" shaped regional characteristics in the whole operating range of the turbine operating condition, that is, the corresponding measures should be taken in any head, during the commissioning process, it was found that the vibration was large only when the turbine head condition was less than 320 m in the low head idling operation (before the non-synchronous guide vane was put in), and in order to improve the situation, only when the running head was less than 320 m, Input non synchronous guide vanes to reduce the vibration during no-load operation and solve the problem of smooth grid connection.

The "S" shape characteristic curve optimized by scheme A is shown in figure 7 (b). The experimental results show that n11 at the intersection of the active vane opening line of a0 = 6 ° and

the runaway curve is the safety margin limit of the "S" zone, and the point is read from n11 = 47.45rpm. The formula is transformed into head H = 320.8m (50Hz). According to the theoretical analysis, it can be seen that the operating range is only within the range of 320.8m or more, and the unit has the characteristics of smooth grid connection, and the normal operation range of less than 320.8m does not have the characteristics of smooth grid connection. In other words, the turbine operating conditions still exist in the operating range may need to use non-synchronous guide vane to solve the problem of grid connection.

The "S" shape characteristic curve optimized by B scheme is shown in figure 7 (c). The test results show that the safety margin of the "S" zone is n11 = 52.8rpm, and the safety margin is limited to H = 257m when the frequency is 49.8Hz according to the relevant calculation formula. At 50Hz, H = 259m; when the frequency is 50.4Hz, H = 263m. The data show that there is a safety margin of 25 m at the minimum head (50 Hz) within the operating range of the turbine operating conditions, Therefore, it can be predicted that the pump turbine has the characteristics of smooth grid connection without the use of asynchronous vanes in the whole operating range.

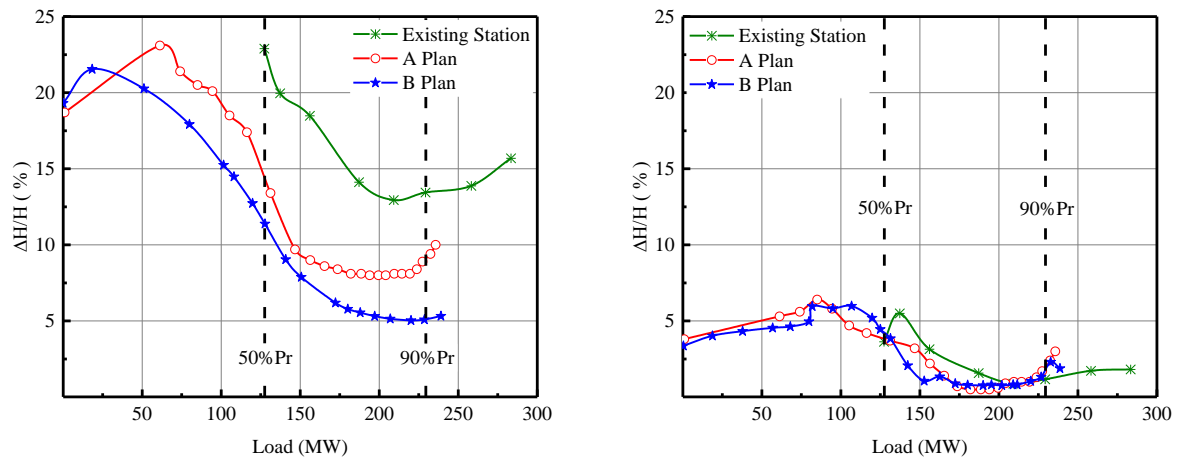*3.3. Pressure pulsation of turbine operating conditions*
Pressure pulsation is an important indicator of the stable operation of the pump turbine. The pressure pulsation test can obtain the auxiliary data from the model test and serve as a guide for predicting the operation of the prototype. The pressure pulsation is affected by the characteristics of the diversion system, the geometrical error of the model and the real machine. Taking into account the turbine operating conditions, the large pressure fluctuation in the vaneless area is the main reason of the plant vibration and noise, in this paper, the pressure pulsation test results of the turbine operation condition are compared and analyzed. (Hmin = 285m) and the rated head (Hnor = 305m), the tailrace pipe wall was compared with the pressure pulsation measurement and the results of the turbine. Where the abscissa of the results is compared with the graph of the output P, the ordinate is the pressure fluctuation amplitude of 97% confidence of the mixing peak-peak (*ΔH/H*)[8-10].

From the comparison of the pressure pulsation of the minimum head turbine operation condition in figure 8, it can be seen that the pressure fluctuation between the lower guide vane and the runner is compared with the existing power station , A scheme and B scheme in the 50% Pr ~ 90% Pr operating range have a more significant reduction, the pressure fluctuation of the 50% Pr working point of the existing power station is about 22.9%, the A scheme is reduced by about 14.4%, and the B scheme is reduced by about 11.3%. In addition, within 50% Pr ~ 90% Pr operating range, A scheme than the existing power station decreased by about 32% to 51%, B scheme than the existing power station dropped by about 50% to 62%. Moreover, although both the lack of no-load conditions and no-load to 50% Pr test values, but according to the existing test curve can also be speculated that A program and B program in the no-load conditions also decreased.

The pressure fluctuation value of the bottom pipe wall of the lower head is lower than that of the existing power station. Although the scheme and the B scheme are not 50% Pr ~ 90% Pr, the whole operating range is reduced, but the partial load is 50% Pr ~ 85% Pr is slightly lower, while the B scheme is the lowest in the partial operating range of 50% Pr ~ 65% Pr.
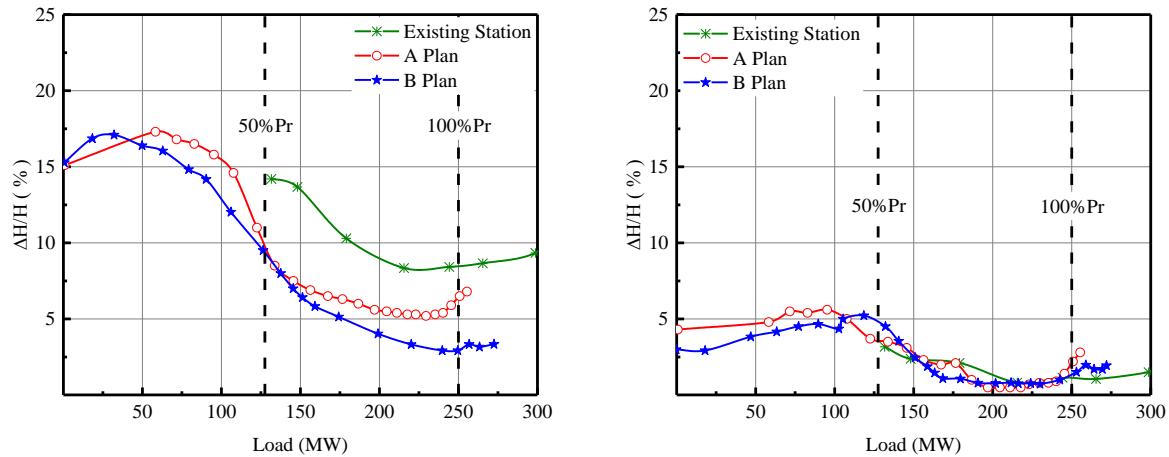
From the comparison of the pressure pulsation of the rated head turbine operation condition in figure 9, it can be seen under the rated head, the pressure pulsation *ΔH/H* between the guide vane and the runner is lower than that of the existing power station. The A scheme and the B scheme are obviously reduced in the range of 50%Pr~100%Pr operation, The maximum pressure pulsation of the 50%Pr operating point of the existing power station is about 14.2%, the A scheme is reduced by about 9.8%, and the B scheme is reduced by about 9.4%. In addition, in the 50% Pr ~ 100% Pr operating range, A scheme than the existing power station dropped by about 31% to 46%, B scheme than the existing power station dropped by about 34% to 65%.

Pressure fluctuation value of draft tube wall under rated head, compared with the existing power station A scheme and B scheme although the local slightly lower and increase, but not obvious, can be considered A scheme and B scheme are not significantly improved.

(a) Between the guide vane and the runner   (b) Draft tube

Figure 8. The pressure pulsation ratio of the minimum head turbine operation condition

(a) Between the guide vane and the runner   (b) Draft tube

Figure 9. Comparison of pressure fluctuation ratio of turbine operation condition under rated head

3.4. Other performance

The cavitation performance of the reversible pump -turbine is poor in the pump operation condition. Therefore, it is usually used in the hydraulic R & D design to judge whether the reversible pump-turbine is cavitation in the operating range. The existing power station, A scheme and B scheme can meet the requirement of no cavitation in the range of operation.

The hump feature is one of the inherent characteristics of the pump-turbine. In the high pump head, the smaller flow area, usually the hump area. In order to avoid the pump in the high-lift conditions of the start-up process and the operation of the operation of the unstable, must meet the hump margin of not less than 2% of the requirements. The existing power station, the A scheme and the B scheme meet the requirements of not less than 2%.

4.Conclusion

This topic is based on the actual situation of existing power station, only by optimizing the movable guide vanes and the runner, without changing the flow passage from the volute to the draft tube, The A scheme and the B scheme have their respective advantages in the aspects of energy characteristics, cavitation performance and hump characteristics of the pump turbine.

Only the optimization of the runner A scheme although the pump-turbine S-shaped area has been more obvious improvement, but it does not guarantee that the turbine operating conditions in the entire operating range does not take the asynchronous guide vane and can be successfully connected. While the optimization of the activities of the guide vane and runner B scheme can achieve a complete improvement in the S-shaped area of the characteristics of the unit in the entire operating range without the use of asynchronous guide vane in the case of safe and stable smooth grid.

The A scheme and the B scheme have obviously improved the pressure pulsation between the guide vanes and the runner of the turbine operation condition, while the B scheme is more prominent at this point.

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