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Force Characteristics Analysis of Yixing Pump-Turbine Head Cover Bolts

Xiang Zhou[2, a], Delou Wang[1,4, b], Peng Zhang[2, c], Liu Chen[3, d], Zheng Zhang[2, e], Chengjun Li[1,4, f], Yuwen Chen[2, g], Shunyi Chen[1,4, h] and Tianjiao Xue[2, i]

[1] HydroChina Huadong Engineering Corporation, Hangzhou 311122, China.

[2] East China Yixing Pumped Storage Company, Yixing 214000, China.

[3] China Institute of Water Resources and Hydropower Research, Beijing 100048, China.

[4] HydroChina Huadong Engineering Corporation, Hangzhou 311122, China.

[a] cr7wszx12345@qq.com, [b] Wang_dl@hdec.com, [c] 903532398@qq.com,

[d] 85550027@qq.com, [e] zheng-zhang@sgxy.sgcc.com.cn, [f] Li_cj@hdec.com,

[g] 545832962@qq.com, [h] Chen_sy@hdec.com, [i] 736221836@qq.com

Abstract. This article conducts numerical calculations on the head cover bolts of the pump turbine in Yixing Pumped Storage Power Station under typical steady-state and transient conditions and analyses the force characteristics of the coupling bolts under turbine, pump, pump zero flow and load rejection conditions. On this basis, further evaluation of head cover bolts safety and reasonable suggestions for subsequent operation are provided.

Keywords: pump-turbine; head cover bolt; force characteristic; numerical calculation.

1. Introduction

In recent years, serious accidents caused by broken head cover bolts have occurred in hydroelectric and pumped storage power plants both domestically and internationally[1-3]. The design, manufacturing, and operation of the head cover bolts have begun to receive attention. At present, the literatures mainly focus on methods for analyzing bolt strength [4-6], the effect of preload on bolt stress [7-12], and research on bolt stress characteristics [13-17]. With the rapid development and application of fluid structure coupling technology, the calculation method of flow induced stress in pump turbines(PT) has also achieved certain results [18-22]. Some literatures have analyzed the steady-state and partial transient process stresses of the head cover bolts of PT [23-29]. Based on existing research, this article obtains accurate water pressure distribution on the head cover bolts of the Yixing pump turbine by the 3D flow field calculation method firstly, and then further loads them on the overall structure of the connection between the head cover and the stay ring. At last, stress distribution of bolts is obtained through structural finite element calculation method (FEM). In order to conduct safety evaluation of bolts on the basis of more accurate stress data, the actual operating data and measured data are adopted, especially in load rejection process.

2. Parameters of unit and bolt

Yixing Pumped Storage Power Station is a daily regulated pure pumped storage power station and located in the southwestern suburbs of Tongguan Mountain in Yixing City. The power station was put into operation in 2008 and is equipped with four vertical single stage mixed flow reversible pump turbine units with a single unit capacity of 250MW. PT is manufactured by GE (Norway), with a runner diameter of 2.6m and a double flange head cover structure. The main parameters of the unit and bolts are as follows:

Table 1 Main parameters of the unit and bolts of Yixing PT

Name	Content
Unit type	mixed flow reversible pump turbine
single unit capacity	250MW
Rated speed	375r/min
Rated head(turbine)	363m
Maximum head(turbine)	410.5m
Minimum head(turbine)	338.6m
Rated output	255MW
Maximum head(pump)	420m
Minimum head(pump)	352.3m
Maximum input(pump)	275MW
runner diameter	2.6m
Material of head cover bolt	34CrNiMo6
Material yield strength	800MPa
Material tensile strength	1100MPa
Head cover bolt diameter	M64
Head cover bolt number	116

3. Flow field numerical calculation of PT

The main loads of the head cover bolt include preload and working load, and the working load mainly comes from the axial water thrust on the head cover. This article conducts a 3D flow field calculation of the entire flow channel of PT based on measured data and obtains the force distribution of the head cover under various operating conditions, providing accurate boundary conditions for bolt finite element calculation and analysis.

3.1 Model and condition

The calculation model and grid division are shown in Figure 1. Considering the complexity of the model, local simplification is applied to non-core geometric structures in actual calculations, such as ignoring some small chamfers, local sharp corners, etc.

The calculation conditions include steady-state turbine condition at the maximum head, steady-state pump condition at the maximum head, and the pump zero flow condition. Totally eight key time points in the load rejection process are selected, they are the starting point, the minimum pressure at the outlet of the draft tube, the maximum speed, the maximum and minimum pressure at the inlet of the spiral case, the maximum pressure at the outlet of the draft tube, and the local maximum and minimum pressure at the inlet of the spiral case, as shown in Figure 2. The working condition parameters are shown in Table 2.

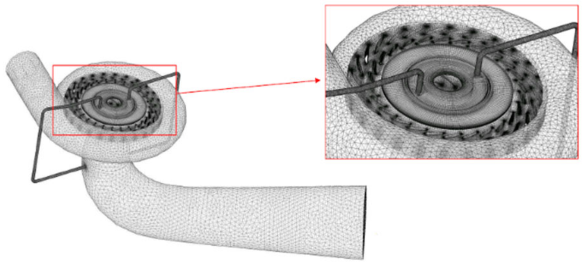


Fig. 1 3D flow field calculation model

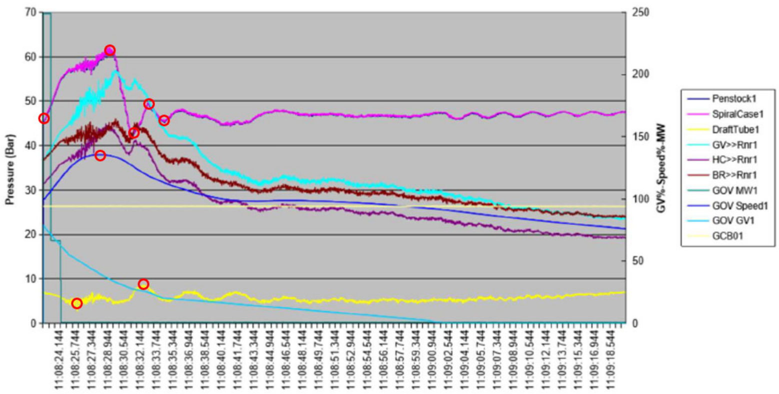


Fig. 2 Conditions in load rejection process

Table 2 Main parameters of calculation conditons

Condition			Output (MW)	Spiral case inlet/outlet pressure (m)	Draft tube outlet/inlet pressure (m)	Head (m)	Speed (r/min)	Opening (°)
Turbine(maximum head)			255	472.5	62	410.5	375	15
Pump(maximum head)			-	477.3	57.2	420.1	-375	14.4
Pump zero flow			55.7	-	57	420.1	-375	0
Load rejection	1	start	-	462.2	72.8	389.4	375	16.8
	2	Minimum pressure of draft tube outlet	-	591.1	28.8	562.4	487.9	10.9
	3	Maximum speed	-	616.5	64.5	552.0	508.6	8.4
	4	maximum pressure of spiral case inlet	-	633.0	57.6	575.4	505.4	7.6
	5	minimum pressure of spiral case inlet	-	428.6	61.9	366.7	476.5	6.0
	6	maximum pressure of draft tube outlet	-	474.7	87.7	387.0	453.3	5.6
	7	local maximum pressure of spiral case inlet	-	507.2	82.7	424.5	445.1	5.3
	8	local minimum pressure of spiral case inlet	-	463.3	54.3	409.0	422.8	4.3

3.2 Calculation method

The working medium of PT is regarded as a viscous incompressible Newtonian fluid, and the k-ω SST turbulence model based on the Reynolds time averaged equation is adopted. This model integrates the advantage of k-ωand k-ε model. It can more accurately predict near wall flow and flow separation [30].

$$\frac{\partial}{\partial x_i}(\vec{u}_i)=0 \tag{1}$$

$$\frac{\partial u_i}{\partial t}+\frac{\partial}{\partial x_j}(u_i u_j)=-\frac{\partial p}{\rho \partial x_i}+\frac{\partial}{\partial x_j}[\frac{(\mu+\mu_t)}{\rho}(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i})]+f_i \tag{2}$$

This article calculates the flow inside a rotating runner. In the rotating coordinate system, the Reynolds time averaged N-S equation is shown in equations (3) to (6):

$$\frac{\partial}{\partial x_i}(w_i)=0 \tag{3}$$

$$\frac{\partial w_i}{\partial t}+\frac{\partial}{\partial x_j}(w_i w_j)=-\frac{\partial p}{\rho \partial x_i}+\frac{\partial}{\partial x_j}[\frac{(\mu+\mu_t)}{\rho}(\frac{\partial w_i}{\partial x_j}+\frac{\partial w_j}{\partial x_i})]+f_i' \tag{4}$$

$$\vec{u}=\vec{w}+\vec{\omega}\times\vec{r} \tag{5}$$

$$\overrightarrow{f'}=-2\vec{\omega}\times\vec{w}-\vec{\omega}\times(\vec{\omega}\times\vec{r})+\vec{f} \tag{6}$$

Where p is pressure, ρ is water density, u_i and w_i are the absolute and relative velocity components, f_i is volume force component, $\vec{\omega}$ is runner rotation angular velocity, μ is viscosity coefficient and μ_t is turbulent viscosity coefficient.

3.3 Calculation results

Compare the calculation results with the pressure measured data in the unit bladeless area and between head cover and runner, as shown in Tables 3 to 4. The deviations under steady turbine and pump operating conditions are relatively small, all of which are 3% or less. The deviation at pump zero flow condition is 7.7%。During the load rejection process, the deviations are relatively large, with the deviations at four operating points exceeding 10%. There may be two reasons. Firstly, the load rejection process is a transient process, where the pressure at each measuring point changes rapidly with time. The delay of the test signal results in differences between the calculated boundary conditions and the actual settings. Secondly is that the calculation model is established based on the design drawings, which may differ slightly from the actual machine.

Table 3 Comparison between the calculated and measured pressure results under normal conditions

Condition	Test point	Measured results MPa	Calculated results MPa	Deviation %
Turbine	GV-RU(bladeless area)	3.80	3.70	2.63
	HC-RU(between head cover and runner)	3.20	3.30	3.13
Pump	GV-RU	3.99	4.0	0.25
	HC-RU	3.20	3.3	3.13

Table 4 Comparison between the calculated and measured pressure results under transitional conditions

Condition		GV-RU MPa		Deviation %	HC-RU MPa		deviation %
		measured results	calculated results		measured results	calculated results	
Pump Zero flow		5.28	5.20	1.52	4.44	4.10	7.70
Load rejection condition	1	3.70	3.60	2.70	3.10	3.30	6.50
	2	4.70	4.80	2.10	3.90	3.90	0
	3	5.10	5.00	2.00	4.40	3.90	11.40
	4	5.40	5.00	7.40	4.40	3.90	11.40
	5	5.30	4.60	13.20	3.90	3.80	2.60
	6	5.20	4.20	19.20	4.00	3.30	17.50
	7	5.10	3.90	23.50	3.90	3.10	20.50
	8	4.30	3.90	9.30	3.30	3.30	0

According to the pressure distribution from the flow field calculation and geometric shape of the head cover, the water thrust of the head cover under various working conditions can be obtained, as shown in Tables 5-6.

Table 5 Water thrust of the head cover under normal conditions

	Turbine	Pump
Water thrust(kN)	68681.3	70290.6

Table 6 water thrust of the head cover under transitional conditions

	Pump Zero flow	Load rejection condition							
		1	2	3	4	5	6	7	8
Water thrust(kN)	79130.1	68050.0	77840.0	75345.6	75643.8	74081.4	69440.5	60794.6	68358.3

4. Finite element calculation of head cover bolt

4.1 Model and condition

The FEM model and grid of bolts is shown in Figure 3, including the the head cover, seat ring and bolts. Similarly, considering the complexity of the model, in actual calculations, each component is assembled into a linear elastic metal structure under ideal conditions for analysis, ignoring and simplifying details such as welds. Do not consider the differences between material properties and mechanical properties and theoretical material data caused by material defects, process production levels, or installation errors. The calculation conditions are consistent with the flow field calculation, and also include the initial state conditions of the unit with no flow.

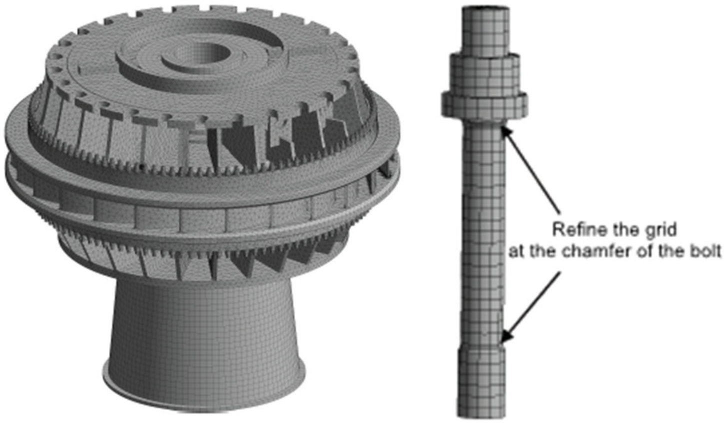


Fig. 3 FEM model

4.2 Stress analysis of bolt

According to the requirements of GB/T 15468-2020 Fundamental technical requirements for hydraulic turbines: "4.2.2.7... The comprehensive working stress of bolts shall not exceed 2/3 of the yield strength of the bolt material under normal and transitional conditions, and shall not exceed 4/5 of the yield strength of the bolt material under special conditions……". The transitional conditions include the start-up process, shutdown process, and load rejection; Special working conditions refer to pressure tests, runaways, guide vane jamming, damage to guide vane protection devices, earthquakes, etc.

This article conducts classical formula calculation and FEM on the head cover, seat ring, and connecting bolt structure under the above working conditions. The stress results of the bolts are shown in Tables 7 to 10, respectively. The design preload stress of the bolt is 560MPa. The average stress calculation result of the middle section of the bolt in its initial state is 560MPa, which is consistent with the design value. The standard yield strength of the bolt material is 800MPa. Through sampling some bolts for actual measurement, the data shows that the actual yield strength is above 970MPa. The actual yield strength of the newly purchased spare bolts is 950MPa. Therefore, this article compares the calculated bolt stress results with the standard yield strength (800MPa) and the actual yield strength (considering 950MPa).

Using classic formulas to calculate the water thrust under various working conditions, evenly distribute it to each bolt and obtain the working load of the bolt. Considering the relative stiffness design value of 0.15 between the bolt and the connecting structure, the additional force of the bolt in the working state can be obtained. The total load of the bolt is composed of the additional force and preload, and the average stress can be obtained based on the size of the middle section. It can be seen that the ratio of the average stress to the standard yield strength of the bolt calculated by the classical formula under normal operating conditions and transitional conditions is 0.74~0.75, and the ratio to the actual yield strength is 0.62~0.63.

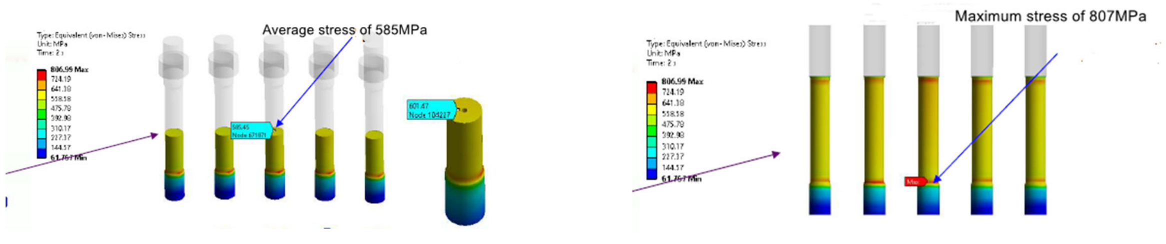
Table 7 Bolt stress results by classic formula under normal conditions

Conditon	Preload	Working load	Additional force	Total load	Diameter of middle section	Average stress of middle section	Average stress/ standard yield strength	Average stress/ actual yield strength
	kN	kN	kN	kN	mm	MPa		
Turbine	1480	592.08	88.81	1568.81	58	593.78	0.74	0.63
Pump	1480	605.95	90.89	1570.89	58	594.57	0.74	0.63

Table 8 Bolt stress results by classic formula under transitional conditions

Conditon		Preload	Working load	Additional force	Total load	Diameter of middle section	Average stress of middle section	Average stress/ standard yield strength	Average stress/ actual yield strength
		kN	kN	kN	kN	mm	MPa		
Pump Zero flow		1480	682.16	102.32	1582.32	58	598.89	0.75	0.63
Load rejection condition	1	1480	586.64	88.00	1568.00	58	583.47	0.74	0.63
	2	1480	671.03	100.66	1580.66	58	598.26	0.75	0.63
	3	1480	649.53	97.43	1577.43	58	597.04	0.75	0.63
	4	1480	652.10	97.82	1577.82	58	597.19	0.75	0.63
	5	1480	638.63	95.79	1575.80	58	596.42	0.75	0.63
	6	1480	598.63	89.79	1569.79	58	594.15	0.74	0.63
	7	1480	524.09	78.61	1558.61	58	589.92	0.74	0.62
	8	1480	589.30	88.39	1568.39	58	593.62	0.74	0.62

Meanwhile, the standard stipulates that "4.2.2.8 Stress analysis obtained by finite element method should provide stress distribution, point out the cloud map of local stress locations, and extract the average stress and local stress of components. The average stress at non-stress concentration points under normal and transitional conditions should not exceed the allowable stress specified in Table 2. Local stress (calculation results of finite element considering transitional fillets) shall not exceed 2/3 of the yield strength of the material, and the maximum local stress under special working conditions shall not exceed the yield strength of the material. " The maximum local stress of the bolts calculated by FEM in this article is located near the chamfer of the stay ring, as shown in Figure 4. The ratio of the maximum stress to the standard yield strength of the bolt under local normal and transitional conditions has exceeded 2/3, and the ratio to the actual yield strength is also between 0.83 and 0.94, which does not meet the standard requirements.



(a)Average stress of middle section (b)maximum stress

Fig. 4 Stress distribution of bolts under turbine condition with maximum head

Table 9 Bolt stress results by FEM under normal conditions

Conditon	Average stress of middle section	Maximum stress	Average stress/ standard yield strength	Average stress/ actual yield strength	Maximum stress / standard yield strength	Maximum stress / actual yield strength
	MPa	MPa				
Turbine	585	807	0.73	0.62	1.01	0.85
Pump	565	772	0.71	0.59	0.98	0.83

Table 10 Bolt stress results by FEM under transitional conditions

Conditon		Average stress of middle section MPa	Maximum stress MPa	Average stress/ standard yield strength	Average stress/ actual yield strength	Maximum stress / standard yield strength	Maximum stress / actual yield strength
Pump Zero flow		600	830	0.75	0.63	1.04	0.87
Load rejection condition	1	587	806	0.73	0.62	1.01	0.85
	2	591	838	0.74	0.62	1.05	0.88
	3	591	853	0.74	0.62	1.07	0.90
	4	591	869	0.74	0.62	1.09	0.91
	5	591	879	0.74	0.62	1.10	0.93
	6	590	882	0.74	0.62	1.10	0.93
	7	587	879	0.73	0.62	1.10	0.93
	8	589	890	0.74	0.62	1.11	0.94

4.3 Safety cofficient analysis of bolt

For connecting bolts, in addition to focus on stress levels, the preload coefficient and clamping force coefficient are also the main assessment indicators. The preload coefficient is the ratio of preload to working load; The clamping force is the ratio of the clamping force to the working load, which is the remaining pre-tightening force between the head cover and the seat ring under working conditions. The preload coefficient and clamping force coefficient obtained through FEM in this article are shown in Table 11. According to the standard "4.2.2.7... The pre tightening force value of the head cover and seat ring connecting bolts should not be less than Table 3 under normal and transitional conditions, divided by the unit head range." The above mentioned Table 3 is detailed in Table 12 of this article. The maximum gross head of Yixing Pumped Storage Power Station is 410.7m, and the corresponding values in the table should not be less than 2.3 for the preload coefficient. It can be seen that the preload coefficient of Yixing meets the standard requirements under normal working conditions, while preload coefficient of the pump zero flow condition and some transitional conditions are slightly less than the requirement.

For the clamping force coefficient, it is usually recommended to take a value of 0.6~1.0. The results are greater than 1.0 under all working conditions and has a certain margin, indicating that there is sufficient margin for the clamping force between the head cover and the stay ring during the working process.

Table 11 Preload and clamping force coefficient results

Condition		Preload kN	Working load kN	Clamping force kN	Preload coefficient	Clamping force coefficient
Turbine		1480	592.08	953.54	2.50	1.61
Pump		1480	605.95	886.82	2.44	1.46
Pump Zero flow		1480	682.16	903.09	2.17	1.32
Load rejection	1	1480	1480	964.26	2.52	1.64
	2	1480	1480	890.43	2.21	1.33
	3	1480	1480	911.94	2.28	1.40
	4	1480	1480	909.37	2.27	1.39
	5	1480	1480	922.84	2.32	1.45
	6	1480	1480	960.20	2.47	1.60
	7	1480	1480	1026.81	2.82	1.96
	8	1480	1480	966.89	2.51	1.64

Table 12 Value of preload coefficient in GB/T 15468-2020

Maximum head(Hmax) m	Hmax < 50	50≤Hmax < 100	100≤Hmax < 300	300≤Hmax < 500	500≤Hmax < 700
Preload/maximum working load	3.8	3.3	2.8	2.3	2.0

5. Conclusions

This article conducts full channel flow field calculation and finite element calculation of the head cover bolt for the pump turbine unit of Yixing Pumped Storage Power Station, obtains the average stress and local maximum stress of the bolt under normal and transitional conditions, analyzes the preload coefficient and clamping force coefficient of the bolt, and evaluates the Yixing head cover bolt. The conclusion is as follows:

The ratio of the average stress of the bolt to the standard yield strength of the material under normal and transitional working conditions exceeds 2/3, and the ratio to the actual yield strength of the material is also close to 2/3; The maximum local stress also significantly exceeds 2/3 of the material's yield strength, which does not meet the standard requirements. Overall, the stress level of bolts under working conditions is relatively high. It is recommended to strengthen monitoring and replace bolts with longer service life in a timely manner.

The preload coefficient of the bolt under normal working conditions meets the standard requirements, but is slightly lower under pump zero flow condition and some conditions during load rejection processes. At the same time, considering that there is sufficient margin for the clamping force between the head cover and the seat ring under various working conditions, the connection effectiveness of the overall structure of the Yixing head cover and seat ring can meet the operational requirements.

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References

[1] WANG Guobing. Some Thoughts about the Sayano-Shushenskaya Hydropower Plant Incident in Russia.SHANXI HYDROTECHNICS. Vol. 176 (2010), No.2, p. 1-5, 15.

[2] YANG Jiandong, ZHAO Kun, LI Ling, et al. Analysis on the causes of units 7 and accidents at Sayano-Shushenskaya hydropower station. JOURNAL OF HYDROELECTRIC ENGINEERING. Vol. 30 (2011) No. 4, p. 226-234.

[3] ZHAO Qiang, CHAI Jian-feng, MA Chuan-bao, et al. Causes of Header Bolt Fracture of a Pumped Storage Power Station. Journal of Yangtze River Scientific Research Institute. Vol. 36 (2019) No. 1, p.134-138.

[4] XIONG Xin, LI Haoliang. Analysis of bolt strength of top cover seat ring connection of energy storage unit. China Plant Engineering.Vol. 5 (2018), p. 137-139.

[5] LI Xiaofeng, SUN Rui, YAN Kaichen. Study on Strength Analysis Method of Bolts in Standardized EMU Equipment Bay. JOURNAL OF DALIAN JIAOTONG UNIVERSITY. Vol. 41 (2020) No. 1, p. 29-34.

[6] ZHAO Jiufeng, NIU Miao-yu, ZHAO Zhen-ci, et al. Strength Check and Test Analysis of High-Strength Bolts in Amusement Facilities. Mechanical Research & Application. Vol. 33 (2020) No. 2, p. 32-34.

[7] HE Shaorun, CHEN Hongyu, YANG Zhao, et al. Analysis of the headcover bolt’s pretightening force of the large pumped storage unit. Hydropower and Pumped Storage. Vol 4(2018) No.1, p. 11-14, 31.

[8] WANG Lu, XUE Jilin. Stress analysis of bolted flange joints based on nonlinear finite element method. China Plant Engineering. (2019) No. 416, p. 137-139.

[9] WEN Shujie, CHANG Xibing, LI Haoliang, et al. Discussion on the preload of bolts on the top cover of large pumped storage units. Pumped Storage Power Station Engineering Construction Collection 2019. (2019). p. 162-168.

[10] QIN Jun-wei, LIU Xiao-hua, CHEN Zhao. Analysis on Pre-tightening Stress Based on Nonlinear Model for Contact between Bolt and Threaded Hole. Pressure Vessel Technology. Vol. 35 (2018) No. 9, p. 30-34.

[11] QIAN Julin, GE Xinfeng, ZHU Shuangju, et al. Static stress analysis of top cover and connecting bolts on Kaplan turbine under different bolts preload. Machinery & Electronics. Vol. 39 (2021) No. 2, P. 12-16.

[12] XIAO Liangyu, LI Yongheng. Study on combined effect of pretansioned head cover, stay ring for hydraulic turbine. Dongfang Electric Review.Vol. 25 (2011) No. 2, p. 31-33.

[13] WANG Ningning, GE Xinfeng, ZUO Li, et al. Stress characteristics of top cover bolts of pumped storage power station based on finite element method. Hydropower and Pumped Storage. Vol. 7 (2021) No. 3, p. 1-6.

[14] GE Xinfeng, XU Xu, Shen Minghui, et al. Strength analysis of remaining bolts as several connection bolts of hydro-turbine head cover in failure. Journal of Drainage and Irrigation Machinery Engineering. Vol. 37 (2019) No. 7, p. 600-605.

[15] CHEN Liu, YU Jixing, LUO Yongyao. Preliminary analysis and study on the stress characteristics of the connecting bolts of the top cover seat ring of xianju pumped storage power station. Proceedings of the 2018 Hydraulic Machinery Information Network Technology Exchange Conference. Hangzhou, 2018, p. 163-167.

[16] XU Jing, XUE Weixin, LU Jian. Stress and Fatigue Analysis of Bolted Flange Connection Structures Based on ANSYS Software. Machinery. Vol. 56 (2018) No. 652, p. 17-20.

[17] GUO Chunli, HUANG Lingyan, TANG Qiang, et al. Stress numerical simulation based on Ansys and failure mechanism analysis of bolt. Cryo. & Supercond. Vol. 45 (2017) No. 2, p. 6-10.

[18] HUANG, X., Oram, C., Sick, M. Static and dynamic stress analyses of the prototype high head Francis runner based on site measurement. IOP Conf. Ser. Earth Environ. Vol. 22 (2014), p. 32052.

[19] JIANG, Y., Yoshimura, S., Imai, R.; Katsura, H., et al. Quantitative evaluation of flow-induced structural vibration and noise in turbomachinery by full-scale weakly coupled simulation. J. Fluids Struct. Vol 23 (2007), p. 531-544.

[20] Egusquiza, E., Valero, M., Presas, A., et al. Analysis of the dynamic response of pump-turbine impellers. Influence of the rotor. Mech. Syst. Signal Process. (2016), p. 68-69, 330-341.

[21] HE, Q., HUANG, X., YANG, M., et al. Fluid-Structure Coupling Analysis of the Stationary Structures of a Prototype Pump Turbine during Load Rejection. Energies Vol. 15 (2022), p. 3764.

[22] TANG Maojia, TAI Rong, CHENG Yongguang, et al. Analysis of flow-induced vibrations of pump-turbines during load rejection by CFD-FEM simulations. Journal of Hydroelectric Engineering, Vol. 41 (2022). No. 7, p. 140-148.

[23] Xie, J., HUANG, B., FU, L. Reinforcement and Resonance Control of Head Cover of Francis Turbine by Finite Element Analysis and Modal Testing. IOP Conf. Ser. Earth Environ. Sci. Vol. 560 (2020), p. 12051.

[24] WANG, Z., YANG, J., WANG, W., et al. Research on the Flow-Induced Stress Characteristics of Head-Cover Bolts of a Pump-Turbine during Turbine Start-Up. Energies Vol. 15 (2022), p. 1832.

[25] LUO, Y., CHEN, F., CHEN, L., et al. Study on Stresses of head cover Bolts in a pump-turbine Based on FSI. IOP Conf. Ser. Earth Environ. Sci. Vol. 804 (2021), p. 42062.

[26] LUO, Y., CHEN, F., CHEN, L., et al. Stresses and Relative Stiffness of the head cover Bolts in a pump-turbine. IOP Conf. Ser. Mater. Sci. Eng. Vol. 493 (2019), p. 12113.

[27] CHEN, F., CHEN, L., WANG, Z., et al. Computation of Static Stresses of the head cover Bolts in a pump-turbine. IOP Conf. Ser. Mater. Sci. Eng. Vol. 493 (2019), p. 12143.

[28] CHEN, L., LI, H., YU, J., et al. Stress Analysis of Screw Connection of Key Structural Components in pump-turbine. IOP Conf. Ser. Earth Environ. Sci. Vol. 804 (2021), p. 32037.

[29] ZHAO W., HUANG, X., YANG, M., et al. Flow-Induced Dynamic Behavior of Head-Cover Bolts in a Prototype Pump-Turbine during Load Rejection. Machines Vol. 10 (2022), p. 1130.

[30] Menter, F.R. Two-equation eddy-viscosity turbulence models for engineering applications. AIAA. 14, 32,1598-1605