Development of a pump-turbine runner based on multiobjective optimization

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Abstract. As a key component of reversible pump-turbine unit, pump-turbine runner rotates at pump or turbine direction according to the demand of power grid, so higher efficiencies under both operating modes have great importance for energy saving. In the present paper, a multiobjective optimization design strategy, which includes 3D inverse design method, CFD calculations, response surface method (RSM) and multi-objective genetic algorithm (MOGA), is introduced to develop a model pump-turbine runner for middle-high head pumped storage plant. Parameters that controlling blade shape, such as blade loading and blade lean angle at high pressure side are chosen as input parameters, while runner efficiencies under both pump and turbine modes are selected as objective functions. In order to validate the availability of the optimization design system, one runner configuration from Pareto front is manufactured for experimental research. Test results show that the highest unit efficiency is 91.0% under turbine mode and 90.8% under pump mode for the designed runner, of which prototype efficiencies are 93.88% and 93.27% respectively. Viscous CFD calculations for full passage model are also conducted, which aim at finding out the hydraulic improvement from internal flow analyses.

1. Introduction

Recently, electricity grid increases rapidly in China with the development of its economy. The requirements for secure, stable and efficient operation are urgent [1]. Therefore, more and more attentions are paid to pumped storage power plants. As key component of the reversible pump-turbines, the runners are needed to rotate at two different directions. Frequent changes between pump mode and turbine mode propose real challenge for the hydraulic design of pump-turbine runners. Hydraulic efficiencies under both pump and turbine modes have to be improved for energy saving. On the other side, hydraulic efficiencies under pump and turbine modes influence each other and sometimes are conflicting [2].

Since the requirements for pump operations are a bit difficult to meet, pump-turbine runners are usually designed from pump mode, and verified with turbine operation [3]. With development of design theory and computing technology, 3D design methods become more and more popular nowadays. Tan et al. proposed a direct and inverse iterative design method by solving the meridional velocity in which effects of the blade shape on the flow were taken into account [4]. Zangeneh et al.
proposed a 3D inverse design method, in which blade shapes are controlled by blade loading distributions on hub and shroud [5-7]. Both geometric and hydrodynamic parameters can be used in description of the runner shapes, but usually the former needs larger number of parameters than the latter. Furthermore, there is no direct relationship between geometric parameters and runner performances, which accordingly needs longer time to find out an optimal runner shape [8]. CFD-based optimization is commonly employed recently in the design of pump-turbine runners, in which runners’ profile is modified by changing the design parameters based on the internal flow analyses [9-11]. Although the pump-turbine runners with rather excellent hydraulic performances can be developed based on CFD, the works are really time-consuming and experience-needing. In order to reduce the computational costs and dependency on experiences, an optimization system based on 3D inverse design, CFD, and optimization algorithm was proposed [8, 12, 13]. In this optimization system, blade loading parameters were set as input, and runner performances obtained from CFD estimations were chosen as objective functions. The optimization system could optimize runner profile automatically and reduce the development time. However, the optimization processes mainly depend on performance estimations conducted by CFD calculations, and one time calculation of runner performance is needed in one time optimization. In this paper, an optimization design strategy combined by 3D inverse design method, design of experiment (DoE), computational fluid dynamics (CFD), response surface methodology (RSM), and multi objective genetic algorithm (MOGA) is introduced. The strategy can solve the multiobjective problems in the design of pump-turbine runners and reduce the computational cost. A brief introduction of the design strategy is firstly given. Then, the optimization strategy is applied to the optimization design of a scaled pump-turbine runner. Finally, model tests and numerical simulations are conducted for verification of the optimal design strategy and analization of the detailed internal flow.

2. Multiobjective Optimization Design System
Optimization design of turbomachinery includes blade configurations, performance estimations and optimization calculations. Figure 1 presents the flow chart of the optimization strategy introduced in this paper. Blade parameterization is conducted by a 3D inverse design method, TURBODesign-1, in which hydrodynamic parameters, such as blade loadings, are set as main inputs. Commercial software ANSYS 13.0 is used to calculate hydraulic performances under different operating conditions. RSM plays the role to build approximate model between input parameters and objective functions by using sample points, on which the optimization algorithm operates. In order to obtain more approximate relationships with less computation, the distributions of sample points in design space is decided by an effective DoE method, a latin hypercube sampling method [14]. The optimization system is built with the help of commercial software iSIGHT.

2.1. 3D inverse design
3D design software, TURBODesign-1 is employed to parametrically design the runner shapes. In this approach, blades are represented by sheets of vorticity, whose strength is determined by a specific distribution of circumferentially averaged swirl velocity $\bar{\vartheta}rV$, defined as

$$\bar{\vartheta}rV = \frac{B}{2\pi} \frac{\vartheta}{\theta_0} rV_\theta d\theta$$

(1)

$rV_\theta$ can also be referred as blade loading, whose meridional derivative is directly related to the pressure distribution on the blade surface in incompressible potential flows

$$p^* - p^- = \frac{2\pi}{B} \rho W_{mbl} \frac{\partial (rV_\theta)}{\partial m}$$

(2)

where $p^*$ and $p^-$ are pressure on the upper and lower side of blade, $B$ is the number of blades and $W_{mbl}$ is the meridional velocity on the blade, and $m$ is meridional location. Blade loading distributions, as shown in Fig.2, are usually specified along the hub and shroud streamlines. Along each streamline,
three-segment distribution is adopted, and four parameters, connection point location NC and ND, slope of the linear line SLOPE, loading at leading edge DRVT, are used to control the distribution curve. The blade loading inside the blade channel is obtained by linear interpolation between the hub and the shroud.

\[
\left( \vec{V}_z + V_{z0} \right) \frac{\partial f}{\partial z} + \left( \vec{V}_r + V_{r0} \right) \frac{\partial f}{\partial r} = \frac{r \vec{V}_{\phi}}{r^2} + \frac{V_{\phi0}}{r} - \omega
\]  
(3)

**Figure 1** Procedure for multiobjectives optimization.

**Figure 2** Blade loading distributions.

Blade loading is directly related to the flow field, and therefore can control the blade shape according to blade shape Eq.(3)
In this first-order hyperbolic partial differential equation, $f$ is the wrap angle. In order to calculate the blade shape, the equation needs to be integrated along the meridional projections of streamlines on the blade surface. The initial integration position of the wrap angle $f$ is determined by a lean angle of high pressure side of the blade $\theta$, as shown in Fig. 3. $\beta$ is the rake angle, which is referred as the stacking condition.

2.2. CFD analysis

CFD plays two roles here, one is to estimate the objective functions, and the other is to validate the optimized results in numerical way. The accuracy of objective function is crucial for the optimization process, so steady turbulent flow simulations are carried for the full-passage pump-turbine. The commercial software ANSYS CFX 13.0 is used to perform the simulations.

2.3. Response surface methodology (RSM)

Optimization processes are based on the relationships between input design parameters and output targets. In Eq.(4), $F_j$ represents the output target, and $X_i$ represents the input parameter.

$$F_j = F_j(X_i) \quad i = 1,N; \quad j = 1,M$$

In engineering design, it is almost impossible to exactly describe the functions between design parameters and output targets. In this paper, response surface methodology (RSM) is employed to express the approximate relationships by using polynomials, as is shown in Eq.(5). The undetermined coefficients can be found out by principle of least square regression with the help of a set of sample points distributing in design space.

$$F_j = \beta_0' + \sum_{i=0}^{n} \beta_i' X_i + \sum_{i=0}^{n} \beta_i'^2 X_i^2 + \sum_{i=0}^{n} \beta_i' X_i X_k$$

2.4. Design of experiment (DoE)

The distribution of sample points in design space has significant influence in the accuracy of approximate model, so it is necessary to choose an effective method for DoE. The latin hypercube
sampling method is selected, because its principle for the distributions of sample points in design space is equiprobable, random and orthogonal, thus higher accuracy of approximate model can be obtained with as few as sample points. The other advantage of latin hypercube sampling method is that the number of sample points can be prescribed manually according to the adopted approximate model. The least number of sample points is determined by the following equation:

\[ S_{\text{min}} = N + 1 \]  \hspace{1cm} (6)

\[ S_{\text{min}} = \frac{(N + 1)(N + 2)}{2} \]  \hspace{1cm} (7)

Eq.(6) is used for linear approximation model and Eq.(7) is used for quadratic approximation model, in which \( N \) is the number of input variables selected. By using latin hypercube sampling method, the effects of changing selected input parameters on any number of objective functions can be examined, and also the interactions between input parameters can be considered, which means the changing of one parameter influences the response caused by changing another.

2.5. Multiobjective optimization strategy

The modified non-dominated sorting genetic algorithm (NSGA-II) is selected to conduct the multiobjective optimization [17, 18] as the RSM between the input parameters and the output targets is built. NSGA-II adopted fast non-dominated sorting and crowding mechanism, and it not only reduced the computation complexity, but also improved the elitist strategy. Therefore, it is suitable for the optimization design of the pump-turbine runners.

So, optimization design starts from the selection of input parameters. Once the variation range of input parameters are given, different combinations decided by DoE are input to TURBODesign-1, and a number of runner configurations are generated. By using ANSYS CFX to calculate runner performances under different operating conditions, the RSM model between input parameters and runner performances is built. Then, NSGA-II operates on the RSM model, and optimal solutions are determined. There is no need to regenerate the runner model and estimate its performances in the optimization process, so Pareto front of the optimal solutions can be obtained in a few minutes.

3. Design of a Scaled Pump-Turbine Runner

3.1. Design specifications

Specific design parameters of a pump-turbine are as follows: in the pump mode the maximum and minimum head are \( H_{\text{max}} = 298 \text{m} \) and \( H_{\text{min}} = 239 \text{m} \), respectively. The minimum discharge is \( 70 \text{m}^3/\text{s} \), and the maximum input power should not exceed 270MW. In the turbine mode, the rated head is \( H_r = 259 \text{m} \), and the rated output power is \( P_r = 255 \text{MW} \). The rotational speed for both pump and turbine mode is \( n_r = 300 \text{rpm} \). In order to carry out the measurements on a standard test rig, a scaled pump-turbine runner is designed, and the design parameters are shown in Tab.1, where \( H_m, Q_m, N_m \) are head, discharge and rotational speed for model runner, B is blade number.

Table. 1 Design parameters of a scaled pump-turbine.

| Mode   | \( H_m \) (m) | \( Q_m \) (m\(^3\)/s) | \( n_m \) (rpm) | B |
|--------|---------------|------------------------|-----------------|---|
| Pump   | 34.79         | 0.337                  | 1200            | 7 |
| Turbine| 37.68         | 0.384                  | 1200            | 7 |

The meridional shape of model pump-turbine runner is given referring to pump-turbine with similar specific speed and superior performance, as is shown in Fig.4. The diameter at high pressure side (HP) is \( D_d = 448.2 \text{mm} \), the diameter at low pressure side (LP) is \( D_{1s} = 270.0 \text{mm} \), the diameter at low pressure side on hub is \( D_{1h} = 143.5 \text{mm} \), the exit width at high pressure side is \( b_2 = 51.2 \text{mm} \).
3.2. Optimization settings

Blade loading and blade lean at high pressure side are selected as the main input parameters. Among blade loading parameters, NC and ND for hub, and ND for shroud are fixed in order to reduce the number of inputs. The variation range of input parameters is shown in Table 2. The runner’s efficiency at pump design point and runner efficiency at turbine rated point is set as an optimization target. Head at pump design point is set as a constraint.

Table 2. Variation range of input parameter.

| Input Parameter | Minimum | Maximum |
|-----------------|---------|---------|
| NCS             | 0.1     | 0.3     |
| SLOPEH          | -2      | 2       |
| SLOPES          | -2      | 2       |
| DRVTH           | -0.05   | 0.05    |
| DRVTS           | -0.05   | 0.05    |
| Blade lean      | -30°    | 30°     |

(Note: NCS is fore-part connection point on shroud, SLOPEH and SLOPES are slope of linear line on hub and shroud, DRVTH and DRVTS are loading at leading edge on hub and shroud, Blade lean is lean angle of high pressure side of the blade $\theta$.)

Forty different runner geometries were provided by TURBODesign-1 to build the quadratic response surface model in DoE. Eighty times CFD calculations were carried for RSM in order to obtain the efficiency in both pump and turbine mode for forty runners. The computation domain is shown in Fig.5, including a spiral casing, stay vanes, guide vanes, a runner and a draft tube. Hexahedral meshes were used except in the volute tongue with tetrahedral meshes for the complicated structure. Total mesh number was about $3,500,000$, and mesh number in the runner domain was about $700,000$. The widely validated RNG k-\(\varepsilon\) turbulence model was used for the closure of the Navier-Stokes equation and the frozen rotor model was used for interfaces between stationary and rotational components. For the pump mode, static pressure was set at inlet and mass flow rate was set at outlet; and for the turbine mode, mass flow rate was set at inlet while static pressure was set at outlet. All the calculations were carried out in a blade server with eight CPUs of Intel 5645 with 2.4GHz processor, 96GB memory and 2TB hard drive.
Once runner performances were numerically calculated, the RSM between input parameters and optimization targets was easily established, and then the optimization algorithm NSGA-II could be operated. The parameter settings for NSGA-II were shown in Tab.3. By using the parameters in Tab.3, 10,000 different optimized runners can be obtained by the optimization algorithm.

Table 3 Parameters setting for MOGA.

| Parameters                | Value |
|---------------------------|-------|
| Population size           | 100   |
| Number of generations     | 100   |
| Crossover probability     | 0.9   |
| Crossover distribution index | 10   |
| Mutation distribution index | 20   |
| Initialization mode       | random |

3.3. Optimization results

The relation between the pump efficiency and the turbine efficiency is shown in Fig.6. There are 10,000 different runners. For each runner, its design parameter is decided by NSGA-II, and its performance is estimated based on RSM. The blue points on the upper right form the Pareto front. It is clear that efficiency under pump and turbine mode are conflicting, the increase in the pump efficiency will decrease the turbine efficiency. Since different choices on the Pareto front would lead to different optimization results, four runners along Pareto front, marked by B1, B2, B3 and B4, are chosen for further detailed investigation.

Table 4 shows the comparisons of efficiency calculated by CFD and estimated by RSM for four optimized runners along Pareto front in Fig.6, the initial base line runner is also reported. CFD results given in Tab.4 are obtained from the redesigned runners using the optimized blade loading and blade lean. The baseline runner was designed by TURBODEsign-1 with an initial blade loading, shown in Fig.7, and zero blade lean at high pressure side. According to Tab.4, the runner’s efficiency in turbine mode can be increased by about 2% with optimization, while efficiency in pump mode remaining almost unchanged.
Figure 6. Pareto front for optimization results.

Figure 7. Comparison of blade loading between baseline and optimized design (B2).

Figure 8. Comparison of runner shape: (a) baseline design; (b) optimized design.
The comparisons of blade loading between initial and optimized runner B2 is shown in Fig.7. The optimized blade loading distributions is fore-loaded on both hub and shroud, and maximum loading difference between hub and shroud appear on the fore part of the blade, which is beneficial for controlling the secondary flow [17].

Figure 8 gives the comparison between the initial and the optimized runner shapes. The optimized runner has a large positive blade lean at high pressure side of runner. Figure 9 shows the pressure distribution comparisons on different spanwise locations between the initial and the optimized runners. Pressure difference between upper and lower sides increases at aft part (high pressure side m=1) of blade and decreases at fore part (low pressure side m=0) near hub, pressure difference between upper and lower sides decreases at aft part of blade and increases at fore part near shroud, while the pressure...
difference at middle span almost the same. Meanwhile pressure variations at runner inlet of turbine operation \((m=1)\) is smaller for optimized runner compared with the initial design, which means inflow states were more uniform.

4. Model tests
Model tests were carried out in a stand high-head test rig in Harbin Institute of Large Electric Machinery of China, as shown in Fig.10, which has about ±0.2% of composite error for efficiency measurement.

Figs.11 and 12 show the measured characteristic curves of pump and turbine modes. The highest unit efficiency of the turbine and pump mode is 91.0% and 90.8%, respectively. Based on the IEC60193 standard, the corresponding unit efficiency for prototype turbine and pump is 93.88% and 93.27%, respectively. Since the operation zone of turbine mode is usually far from its highest efficiency point, the averaged efficiency in operation zone is useful. For the optimized runner, this efficiency is 87.56% and 90.31% for model and prototype runner.

![Figure 10](image)

**Figure 10** Test rig and model pump-turbine runner.

![Figure 11](image)

**Figure 11.** Efficiency curve of the turbine mode.
5. Conclusions
A multiobjective optimization design system for the pump-turbine runner is present in this paper. The system consists of 3D inverse design, DoE, CFD, RSM and MOGA, and can be used to carry out the optimization designs with multiobjectives at multipoints. Therefore, this system is very suitable for development of pump-turbine runner.

A middle-head pump-turbine runner was designed by using the proposed system. Optimization results show that, with fixed meridional shape, big positive blade lean at high pressure side and fore-loaded blade loading distributions on both hub and shroud are good for the improvement of runner performances, especially for turbine operation. The model test shows that the highest unit efficiency of the turbine and pump mode is 91.0% and 90.8%, and corresponding unit efficiency for prototype turbine and pump is 93.88% and 93.27%, respectively.

6. References
[1] Zeng M, Feng J J, Xue S, et al. Development of China's pumped storage plant and related policy analysis. Energy Policy, 2013, 61: 104-113.
[2] Yang W, Xiao R F. Multiobjective optimization design of a pump–turbine impeller based on an inverse design using a combination optimization strategy. Journal of Fluids Engineering, 2014, 136(1): 014501.
[3] Mei ZY. Power generation technology of pumped storage power station. China Machine Press; 2000.p.105 [in Chinese]
[4] Tan L, Cao S, Wang Y, et al. Direct and inverse iterative design method for centrifugal pump impellers. Proceedings of the Institution of Mechanical Engineers, Part A: Journal of Power and Energy, 2012, 226(6): 764-775.
[5] Zangeneh M. A compressible three-dimensional design method for radial and mixed flow turbomachinery blades. International Journal for Numerical Methods in Fluids, 1991, 13(5): 599-624.

[6] Zangeneh M. Inviscid-viscous interaction method for three-dimensional inverse design of centrifugal impellers. Journal of Turbomachinery, 1994, 116: 280.

[7] Zangeneh M, Goto A, Takemura T. Suppression of secondary flows in a mixed-flow pump impeller by application of three-dimensional inverse design method: part 1—design and numerical validation. Journal of Turbomachinery, 1996, 118: 536.

[8] Ashihara K, Goto A. Turbomachinery blade design using 3-D inverse design method, CFD and optimization algorithm. ASME paper, 2001, (2001-GT): 0358.

[9] Tani K, Okumura E H. Performance improvement of pump-turbine for large capacity pumped storage power plant in USA. Hitachi Review, 2009, 58(5): 199.

[10] Tani K, Niikura K. CFD-based design optimization for hydro turbines. Journal of Fluids Engineering, 2007, 129: 159.

[11] Sallaberger M, Bachmann P, Michaud C, et al. Modern hydraulic design of large pump-turbines. International Journal on Hydropower & Dams, 2003, 10(5): 112-118.

[12] Zangeneh M, Daneshkhah K, Dacosta B. A multi-objective automatic optimization strategy for design of waterjet pumps. 2008.

[13] Yiu K, Zangeneh M. Three-dimensional automatic optimization method for turbomachinery blade design. Journal of Propulsion and Power, 2000, 16(6): 1174-1181.

[14] Sui Y K, Yu H P. Improvement of response surface method and its application to engineering optimization. Science Press; 2011, 8-10 [in Chinese]

[15] Deb K, Agrawal S, Pratap A, et al. A fast elitist non-dominated sorting genetic algorithm for multi-objective optimization: NSGA-II. Lecture notes in computer science, 2000, 1917: 849-858.

[16] Deb K, Pratap A, Agarwal S, et al. A fast and elitist multiobjective genetic algorithm: NSGA-II. Evolutionary Computation, IEEE Transactions on, 2002, 6(2): 182-197.

[17] Zangeneh M, Goto A, Harada H. On the design criteria for suppression of secondary flows in centrifugal and mixed flow impellers. Journal of Turbomachinery, 1998, 120: 723.

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