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Numerical and experimental investigations on inflow loss in the energy recovery turbines with back-curved and front-curved impeller based on the entropy generation theory

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ABSTRACT

Pump as turbine (PAT) is one of the economical and effective energy recovery devices in small hydropower stations. A back-curved PAT and a front-curved PAT were designed, and performance characteristics were studied, the accuracy of the numerical calculation was verified by comparing with the experimental results. The entropy generation theory was used to compare performance and energy loss of PATs. The results show that the high efficiency range of front-curved PAT is significantly wider than that of back-curved PAT. Under part-load condition ($0.8Q_d$), design flow condition ($1.0Q_d$) and over-load condition ($1.2Q_d$), the efficiency of the front-curved PAT is 0.6%, 5.9% and 7.9% higher than that of the back-curved PAT, respectively. The energy loss in the PAT impeller mainly comes from the turbulent entropy generation rate which is mainly concentrated on the blade leading edge and trailing edge. Flow separation and flow impact caused by the mismatch between the relative flow angle and the blade setting angle are the main mechanisms of energy loss in impeller. In addition, the loss caused by the wall friction in the front-curved impeller is less than that in the back-curved impeller. Therefore, the entropy generation theory can provide guidance for the performance optimization of PAT.

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1. Introduction

Pump as turbine (PAT) can be considered an effective way to mini-hydropower development in remote areas and pressure energy recovery in the industry. And rural electricity supply involved institutions have recently recommended the pump as turbine (PAT) based micro hydropower plant schemes for remote off-grid electrification, mostly from their economic advantages [1]. Nowadays, PAT is widely used as storage power plant [2,3], water distribution networks [4] and pressure reducing valves sites with small power output capacities owing to their cost effectiveness, high efficiency, effective storage capability and high flexibility [5]. Because of the increasing importance of PAT in the hydropower industry and energy policy, it is of great significance to study how to improve the performance and operational stability of a PAT.

In order to improve the performance of the PAT, a large number of investigations and studies had been carried out in recent years.

This part of research mainly focused on the parameter relationships between the pump mode and the turbine mode, the hydraulic design method of PAT, the influence mechanism of the geometric parameters of the turbine components on the hydraulic performance, the relationship between the hydraulic performance, energy loss mechanism and internal flow characteristics of the PAT. In terms of the relationships of performance between the pump mode and turbine mode, different methods have been used to disclose the relationships between the flow ratio, the head ratio and the specific speed. Singh et al. [6] found that under the same rotation speed, the flow rate and head of turbine mode are larger than that of pump mode at best efficiency point. Qian et al. [7] designed adjustable guide, which provides a cost-effective solution to considerably improve the efficiency of the axial-flow PAT under partial loading. Xu Tan [8], Shahram [9], Fernández, J [10] and Frosina, Emma et al. [11] discovered some relations based on experimental data, and the relations were used to predict the best efficiency point of a pump working as a turbine based on pump hydraulic characteristics. Yang et al. [12] proposed a theoretical conversion equations of hydraulic parameters between pump mode and turbine mode by analyzing a large amount of

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Nomenclature			
S_T	Stress tensor (—)	$\Delta\eta_{abs}$	Absolute error of efficiency (%)
μ	Dynamic viscosity (Pa·s)	$\Delta\eta_{rel}$	Relative error of efficiency (%)
W	Entropy generation rate ($W \cdot m^{-3}$)	I	Unit tensor
W_{vis}	Viscous entropy generation rate ($W \cdot m^{-3}$)	φ	Viscous dissipation function
W_{tur}	Turbulent entropy generation rate ($W \cdot m^{-3}$)	$\bar{\omega}$	Vorticity (s^{-1})
W_{wall}	Wall entropy generation rate ($W \cdot m^{-3}$)	H	Head (m)
P_{egr}	Entropy generation power (W)	g	Gravitational constant ($m \cdot s^{-2}$)
P_{vis}	Viscous entropy generation power (W)	ρ	Density ($kg \cdot m^{-3}$)
P_{tur}	Turbulent entropy generation power (W)	H_d	Designed head (m)
P_{vol}	Volume entropy generation power (W)	Q_d	Designed flow rate ($m^3 \cdot h^{-1}$)
P_{wall}	Wall entropy generation power (W)	N_{sd}	Designed specific speed
Y^+	y plus value (—)	M	Torque of the PAT ($N \cdot m$)
σ	Wall stress (Pa)	n_d	Rotating speed (rpm)
p_{in}	Inlet pressure (Pa)	η	Efficiency (—)
p_{out}	Outlet pressure (Pa)	P	Shaft power (kW)
p	Static pressure (Pa)	ε	Turbulent eddy dissipation ($m^2 s^{-3}$)
f	Body force	Acronyms	
u	Relative velocity ($m \cdot s^{-1}$)	PAT	Pump as turbine
∇	Hamiltonian operator	LE	Leading edge
S_r	Strain rate tensor	TE	Trailing edge
U	Uncertainty	PS	Pressure surface
$\bar{X}$	Average of the measured values	SS	Suction surface
z	Numbers of the measurement	RAM	Random access memory
X_m	Measured value	SD	Standard deviation

experimental data, and found that their theoretical equation can calculate the results closer to the existing experimental data than the equation proposed by Stepanoff [13] does. Tao Wang [14,15] proposed a hydraulic design method for front-curved impeller in PAT under the condition that the volute structure and the diameter of the impeller inlet and outlet remain unchanged. In terms of the research on the influence of geometric parameters of PAT components on hydraulic performance, references [16,17] found that the number of blades and the inlet diameter of the blades have a significant impact on the performance of the PAT. Singh et al. [6,18] suggested that the efficiency of the PAT can be significantly improved by appropriately adjusting the radial clearance between the volute and the impeller. Derakshan et al. [19] found that the efficiency of the PAT can be increased 5.5% by rounding the inlet and outlet of the impeller blades. Yang et al. [20] found that the efficiency of the PAT can be increased 10.6% by appropriately reducing the size of the impeller.

In the study on the relationship of the hydraulic performance, energy loss and internal flow characteristics of the turbine, entropy generation theory, a method to analyze the energy loss, had been widely applied in the various rotating machines, such as wells turbine [21], cyclone separators [22], pump [23,24], pump - turbine [25]. Gong et al. [26] firstly used the entropy generation theory to calculate the loss in a Francis turbine and determine the location where the loss mainly occurs. Soltanmohamadi [27] presented a comparison of total entropy generation between a suggested design and a constant chord Wells turbine. Xiaojun Li et al. [28,29] investigated the variation of hydraulic losses based on the relationship between velocity distribution and entropy generation fields. Xiaoqi Jia et al. [30] predicted the amount and location of the internal flow loss of the centrifugal pump by the entropy generation rate method. J. Cao et al. [31,32] revealed that the entropy generation rate inside rotating machinery is caused by fluid viscosity, turbulent pulsation and wall friction. And the turbulent pulsation and wall friction are the main causes of energy loss in

rotating machinery. In addition, the investigations revealed that vortex at the wake region, backflow region, separation zones and inlet shock area of the blades or vanes are responsible for the dominant turbulent dissipation. Compared with the traditional method that uses differential pressure to evaluate hydraulic loss, entropy generation can determine the location where the loss mainly occurs [33], which is beneficial to objectively modifying the geometric parameters of PAT to improve performance. However, it has not report yet to use entropy generation theory to analyze the energy loss characteristics of front-curved PAT and determine the main location of the loss. In addition, there are few reports on systematically comparing the performance and energy loss characteristics between the front-curved PAT and the back-curved PAT.

In this study, a back-curved and a front-curved PAT are designed. A test bench for PAT is built, and the hydraulic performances of two PATs are tested. The accuracy of the numerical calculation method is verified based on the experimental results. The entropy generation rate values of the PAT components are studied under different conditions according to entropy generation theory, and the main locations and energy loss mechanism of the impeller in the PAT are studied. Therefore, the performance characteristics of the two kinds of turbines are systematically analyzed and compared. The main sources of losses in turbine are deeply analyzed by using entropy production theory, and the main mechanism of loss in the impeller is revealed. These research results can provide reference and guidance for the hydraulic design of a PAT and the optimization design of the energy recovery PAT main components.

2. Entropy generation theory

Entropy generation is an unavoidable consequence of the irreversible energy conversion processes. Neglecting the heat transfer effects, the kinetic energy transport equation of fluid is as follows [32]:

$$\rho \frac{DE}{Dt} = \rho f \cdot u + (\nabla \cdot u)p + \nabla \cdot (S_T \cdot u) - \varphi \quad (1)$$

where S_T is the stress tensor: $S_T = 2\mu S_r - \left(p + \frac{2}{3\mu}(\nabla \cdot u)\right)I$. S_r , I and μ are the strain rate tensor, unit tensor and dynamic viscosity, respectively. The u is the relative velocity, ρ is the density of fluid.

φ is the viscous dissipation function of fluid with $\varphi = -2\mu(\nabla \cdot u)^2/3 + 2\mu S_r$. Take no account of the cavitation phenomenon, the water acts as an incompressible liquid. Following tensor decomposition law, φ can be expressed as:

$$\varphi = \mu|\omega|^2 - 2\mu \nabla \cdot ((\nabla \cdot u)I - \nabla u^T \cdot u) \quad (2)$$

Substituting Eq. (2) into Eq. (1) and neglecting the effect of body force, the energy transport equation is as follows:

$$\rho \frac{DE}{Dt} = -\nabla \cdot (\rho u) - \nabla \cdot (u \varphi \times u) - \mu|\omega|^2 \quad (3)$$

where ω is vorticity. On the right side of the equal sign, the first term is the unit work from the surface pressure gradient in the streamline direction, the second term is nonlinear effects of vorticity and velocity field in viscous fluids, and the last term is the loss of energy by viscosity and turbulent fluctuation. $W = \mu|\omega|^2$ is defined as the entropy generation rate which is always a positive value and directly related to the loss of energy [34].

For an incompressible flow, the entropy generation rate function can be written as follows:

$$W = \mu \left[\left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z} \right)^2 + \left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \right)^2 \right] \quad (4)$$

As for turbulent flow, however, the entropy generation rate can be separated into two terms after the Reynolds time averaged process: one is the viscous entropy generation rate (W_{vis}), and the other is the turbulent entropy generation rate (W_{tur}).

$$W = W_{vis} + W_{tur} \quad (5)$$

$$W_{vis} = \mu \left[\left(\frac{\partial \bar{w}}{\partial y} - \frac{\partial \bar{v}}{\partial z} \right)^2 + \left(\frac{\partial \bar{u}}{\partial z} - \frac{\partial \bar{w}}{\partial x} \right)^2 + \left(\frac{\partial \bar{v}}{\partial x} - \frac{\partial \bar{u}}{\partial y} \right)^2 \right] \quad (6)$$

$$W_{tur} = \mu \left[\left(\frac{\partial w}{\partial y} - \frac{\partial v}{\partial z} \right)^2 + \left(\frac{\partial u}{\partial z} - \frac{\partial w}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \right)^2 \right] \quad (7)$$

After solving the RANS equation, W_{vis} can be obtained from velocity field. However, W_{tur} cannot be obtained from the simulation results directly. Kock et al. [35] proposed that turbulent loss is closely related to the turbulence model used in the calculation. Therefore, the W_{tur} is defined as the product of turbulent eddy dissipation ε and the density of fluid.

$$W_{tur} = \rho \varepsilon \quad (8)$$

A velocity gradient existing in the wall generates large energy loss [36]. Hou et al. [37] suggested that the wall entropy generation rate (W_{wall}) can be calculated by.

$$W_{wall} = \sigma \cdot u \quad (9)$$

where σ is wall shear stress, u is relative velocity.

Therefore, the viscous entropy generation power (P_{vis}), turbulent entropy generation power (P_{tur}), wall entropy generation

power (P_{wall}) and total entropy generation power (P_{egr}) can be calculated by the integral method, respectively.

$$P_{vis} = \int_0^V W_{vis} dV \quad (10)$$

$$P_{tur} = \int_0^V W_{tur} dV \quad (11)$$

$$P_{wall} = \int_0^S W_{wall} dS \quad (12)$$

$$P_{egr} = P_{vis} + P_{tur} + P_{wall} = P_{vol} + P_{wall} \quad (13)$$

The sum of the first two terms on the right side of Eq. (13) is defined as volume entropy generation power (P_{vol}).

3. Experimental and numerical setup

3.1. Physical model

In this paper, a centrifugal PAT was selected as the research object. The flow rate (Q_d), head (H_d), speed (n_d) and specific speed ($N_{sd} = 3.65n_d \cdot \sqrt{Q_d}/H_d^{3/4}$) under design conditions are 21.8 m³/h, 15 m, 1450 rpm and 55, respectively. Based on the structural parameters of the volute, a back-curved impeller and a front-curved impeller were designed in order to systematically study the effects of impeller structures on the performance and energy loss characteristics. The main geometric parameters of the PATs were shown in Table 1. Fig. 1 shows the schematic views of the impeller and the volute.

Fig. 2 shows the back-curved and front-curved impeller and the computational domain of PAT. The computation domain was composed of four parts: volute, chamber, impeller and draft tube. It is noted that in order to reduce the effect of boundary conditions perturbations, to create a stable flow at the inlet, and to provide a fully developed flow at the outlet, the lengths of inlet pipe and the draft tube were extended to be 5 times the spiral casing inlet diameter and 5 times the pump suction diameter, respectively. LE and TE in the figure refer to the leading edge and trailing edge, and PS and SS refer to the pressure surface and suction surface

Table 1
Geometric parameters of the PATs.

Components	Parameters	Notation	Value
Volute	Volute inlet diameter (mm)	D_1	50
	Volute basic circle diameter (mm)	D_2	175
	Volute outlet width (mm)	b_1	20
Back-curved impeller	Inlet diameter (mm)	D_3	165
	Outlet diameter (mm)	D_4	65
	Inlet width (mm)	b_2	7
	Blade number	Z	6
	Blade inlet setting angle (°)	β_1	39
	Blade outlet setting angle (°)	β_2	30
	Blade wrap angle (°)	φ	120
Front-curved Impeller	Inlet diameter (mm)	D_3	165
	Outlet diameter (mm)	D_4	65
	Inlet width (mm)	b_2	8
	Blade number	Z	11
	Blade inlet setting angle (°)	β_1	90
	Blade outlet setting angle (°)	β_2	35
	Blade wrap angle (°)	φ	35

Fig. 1. The schematic views of the impeller and the volute.

Fig. 2. Hydraulic model of PAT.

respectively.

3.2. Experimental setup and process

Fig. 3 shows the diagram of the PAT test system. The front booster pump was used to provide high-pressure fluid for the PAT. The frequency converter was used to adjust the head and flow rate of the front booster pump, and then adjust the pressure and flow rate of the PAT inlet. The pressure transmitters installed the inlet and outlet of the PAT were used to measure the pressure of the PAT inlet and outlet, and thus the head of the PAT can be obtained. The

electromagnetic flowmeter was used to measure the flow rate of the PAT. In addition, the eddy current dynamometer system was used to measure the rotating speed and torque of the PAT, and to control the rotating speed of the PAT with the purpose of controlling PAT at the same speed under different flow rates. Computer system was used to collect and process data. Table 2 shows the range and accuracy of measurement apparatus. Fig. 4 shows the PAT and dynamometer.

During the performance test, open the valve at the inlet and outlet of the water tank and adjust the frequency converter to make the front booster pump operate and output fluids with different pressures and flow rates. When the flow rate and pressure at the outlet of the booster pump reach a certain value, the PAT starts to operate. Then increase the frequency of the frequency converter until the operating speed of the PAT exceed 1450 rpm, and keeping the rotating speed constant by adjusting the dynamometer system, and adjust the frequency converter to make the PAT operate under different flow rate conditions. Meanwhile, the relevant parameters including the pressure, the flowrate and the shaft power was obtained by corresponding measurement apparatus. Therefore, the external characteristic curves of the PATs under different flow rate conditions could be obtained.

3.3. Mesh generation

The AYSYS ICEM was used to mesh the 4 different components in this paper. The volute was divided by hexahedral structured mesh, and the meshes near the wall of the PAT volute and near the tongue were refined, the thickness of the first layer near the wall

Fig. 3. Diagram of PAT performance test system.

Table 2
Range and accuracy of measurement apparatus.

Apparatus	Type	Measurement parameters	Range	Accuracy
Electromagnetic Flowmeter	MEX-LDE	Flow rate $Q(\text{m}^3/\text{h})$	0–120	$\pm 0.5\%$
Pressure Transmitters	MEX-3051TG	Inlet, outlet pressure $P(\text{Pa})$	0–1.6	$\pm 0.05\%$
Eddy current dynamometer	CWF11D	Torque $M(\text{N} \cdot \text{m})$	0–35	$\pm 0.4\%$
Rotating speed sensor		Rotating speed $n(\text{r/min})$	1–10000	$\pm 1\text{r/min}$

Fig. 4. The PAT and dynamometer.

area was set to 0.03 mm. Fig. 5 shows the mesh generation of the PAT impeller. For the back-curved impeller, hexahedral structured mesh was used, and the nodes of the wall mesh were increased by refining the boundary mesh. The thickness of the first layer of mesh is 0.02 mm, as shown in Fig. 5 (a). For front-curved impeller, due to the large number of blades, the minimum mesh angle was less than 18° when using structural meshing, so unstructured hybrid mesh was used. In addition, the method of refining the boundary mesh was used to increase the number of nodes near the wall as shown in Fig. 5 (b).

The performance parameters of a PAT include head, shaft power and efficiency. They were calculated under different flow conditions by the following equations:

$$H = \frac{p_{\text{in}} - p_{\text{out}}}{\rho g} \quad (14)$$

$$P = M \cdot \frac{2\pi n}{60} \quad (15)$$

Fig. 5. Mesh generation (a) Back-curved impeller (b) Front-curved impeller.

$$\eta = \frac{P}{\rho g Q H} \times 100\% \quad (16)$$

where p_{in} , p_{out} is the total pressure of the PAT inlet and outlet, respectively. H is the head, ρ is the density of fluid, g is the gravitational acceleration. P is the shaft power, M is the torque of the PAT, n is the rotating speed. η is the hydraulic efficiency, and Q is the flow rate.

In order to validate the reliability and accuracy of the numerical solution, the solution independence from the generated mesh has been studied. On the basis of not changing the meshing method, the back-curved and front-curved PATs were divided into 7 group meshes with different numbers by adjusting the number of nodes of the different parts of the mesh, respectively. Table 3 shows the number of meshes of the impeller and volute and the numerical calculation time for each case. During numerical calculation, only the meshes of impeller and volute was refined, and the grid schemes of PAT chamber remain unchanged. The same boundary conditions and turbulence models were used in the numerical calculations to ensure the reliability of the computational results. The efficiencies of the PATs were selected as the benchmark to evaluate the independence of the numerical solution from the generated mesh. The model of computer dual processor used in this study is Intel (R) Xeon (R) CPU E5-2660 v4 @ 2.00 GHz, with 28 cores and 56 logical processors. The random access memory (RAM) of the computer is 128 GB. Table 3 lists the details on the mesh tests, computational time for each mesh case, and Fig. 6 shows the efficiencies of back-curved and front-curved PATs under different mesh numbers. The results show that the maximum error of the hydraulic efficiency of the back-curved and front-curved PAT is 0.07% and 0.06%, respectively. In order to ensure the accuracy of numerical calculations and save calculation costs, the mesh 4 were selected to numerical calculations for back-curved and front-curved PAT. The total number meshes in the computation domain of back-curved and front-curved PATs are 11078528 and 9084023 respectively.

3.4. Governing equations and numerical settings

Considering the high Reynolds number of the flow in the PAT, the CFX 19.2 platform was utilized to solve the fully 3D steady Reynolds-averaged Navier–Stokes (RANS) equations. In Reynolds decomposition, every instantaneous quantity is decomposed into its time-averaged part and fluctuating part, and the governing equations is defined as follows [22]:

$$\frac{\partial}{\partial x_i} (\rho \bar{u}_i) = 0 \quad (17)$$

Fig. 6. Effect of the number of meshes on efficiency of different PAT.

$$\frac{\partial}{\partial x_j} (\rho \bar{u}_i \bar{u}_j) = -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial \bar{u}_k}{\partial x_k} \right) \right] + \frac{\partial}{\partial x_j} (-\rho \bar{u}'_i \bar{u}'_j) \quad (18)$$

In the above equations, $\bar{u}_i$ and $\bar{\rho u'_i u'_j}$ are the Reynolds average velocity vector and the Reynolds stress tensor representing the effects of turbulence, respectively. The u'_i in term $\bar{\rho u'_i u'_j}$ is the fluctuating component of the flow velocity. The μ is dynamic viscosity.

To enclose the system of equations and to model the Reynolds stress, the shear stress transport (SST) turbulent model was used to compute the turbulent viscosity. The SST model is a two-equation model, including turbulent kinetic energy, k (Eq. (16)), and turbulent frequency, ω (Eq. (17)), equations [38].

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho \bar{u}_j k)}{\partial x_j} = \tilde{P}_k - \beta^* \rho k \omega + \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{k3}} \right) \frac{\partial k}{\partial x_j} \right] \quad (19)$$

$$\begin{aligned} \frac{\partial (\rho \omega)}{\partial t} + \frac{\partial (\rho \bar{u}_j \omega)}{\partial x_j} = & \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_{\omega 3}} \right) \frac{\partial \omega}{\partial x_j} \right] + \alpha_3 \frac{\omega}{k} \tilde{P}_k - \beta_3 \rho \omega^2 \\ & + (1 - F_1) 2 \rho \frac{1}{\omega \sigma_{\omega 2}} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j} \end{aligned} \quad (20)$$

where $\beta^* = 0.09$, $\sigma_{k3} = 1/0.856$, and the model coefficients σ_{k3} , $\sigma_{\omega 3}$, α_3 and β_3 are linear combinations of the corresponding coefficients of the k - ω and the modified k - ϵ turbulence models, and F_1 is the blending function of the wall distance. μ_t is the turbulent viscosity defined as:

Table 3

Details on the mesh tests, computational time for each mesh case.

Mesh cases	Back-curved impeller (million)	Front-curved impeller (million)	Back-curved volute (million)	Front-curved Volute (million)	Computational time of Back-curved PAT (h)	Computational time of Front-curved PAT (h)
Mesh 1	1.693	1.568	2.854	2.854	8 h	7.8 h
Mesh 2	1.774	1.687	2.854	2.854	9.6 h	9 h
Mesh 3	2.035	2.125	3.624	3.624	10.3 h	9.5 h
Mesh 4	3.705	3.033	3.624	3.624	11 h	10.3 h
Mesh 5	3.884	4.105	3.624	3.624	13.6 h	10.5 h
Mesh 6	5.327	4.693	4.835	5.625	15 h	12 h
Mesh 7	6.936	5.327	4.835	5.625	18 h	13.6 h

$$\mu_t = \frac{\rho a_1 k}{\max(a_1 \omega, SF_2)} \quad (21)$$

where $a_1 = 0.31$, F_2 is a blending function which restricts the limiter to the wall boundary layer, S is an invariant measure of the strain rate.

The high-resolution scheme was used to discretize the advection terms. The criterion for solution convergence in continuity equation and linear momentum equations was chosen based on the root mean square (RMS) method and was set to 10^{-5} . The boundary conditions are setting as following: the inlet of the PAT adopts static pressure and the outlet adopts mass-flow. The impeller of the PAT was set as the rotating domain with the rotating speed of 1450 rpm, and the other components were set as the static domain. The inner wall surfaces of the front and the back chamber were set as rotating wall surface, and the wall roughness was set to 30 μm . At the interfaces of between the stationary components the general connection was selected, and at the rotating-stationary interfaces, the frozen rotor option was selected.

3.5. Numerical validation

It is very necessary to verify the reliability of the numerical calculation method through the experimental results. Firstly, the experimental results were evaluated with the uncertainty analysis. The average of the measured values is given by $(\bar{X})$:

$$\bar{X} = \frac{\sum X_m}{z} \quad (22)$$

where z is the numbers of the measurement, X_m is the measured value. Standard deviation (SD) is given as follows:

$$SD = \sqrt{\frac{\sum_{m=1}^z (X_m - \bar{X})^2}{z - 1}} \quad (23)$$

Then, the uncertainty (U) is given by Eq. (14) as following [39]:

$$U = \frac{SD}{\sqrt{z}} \times 100\% \quad (24)$$

Finally, the numerical and experimental results have been compared with each other, and the maximum relative error is used to evaluate the reliability of the numerical method, which is calculated as [40]:

$$\Delta\eta_{\text{abs}} = |\eta_{\text{EXP}} - \eta_{\text{CFD}}| \quad (25)$$

$$\Delta\eta_{\text{rel}} = \frac{\Delta\eta_{\text{abs}}}{\max\{\eta_{\text{EXP}}, \eta_{\text{CFD}}\}} \times 100\% \quad (26)$$

where the η_{EXP} is the experimental efficiency, the η_{CFD} is the numerical efficiency, the $\Delta\eta_{\text{abs}}$ is the absolute efficiency error. The $\Delta\eta_{\text{rel}}$ is the relative efficiency error.

In this experiment, the external characteristic curves were repeatedly measured five times, and according to the method describing above, under the design condition, the efficiency uncertainties of back-curved and front-curved PAT are 8.1% and 4.7%, respectively. Fig. 7 (a) and Fig. 7 (b) show the comparisons between the experimental and numerical calculation results of a back-curved PAT and a front-curved PAT, respectively. It can be observed that the change trends of the PATs numerical calculation performance are basically consistent with the experimental results. There are still some errors between the numerical and the experimental results. These can be attributed to two reasons: the

Fig. 7. Comparison between the experimental and numerical results (a) Back-curved PAT (b) Front-curved PAT.

geometries used for computation is not as totally same as that for and experiment, and there is a certain error on the friction loss and volume loss between the numerical and the experimental results. However, the overall error ratio of numerical calculation and experiment is relatively small. The maximum errors between numerical and experimental back-curved PAT head and efficiency are 3.4% and 2.1%, respectively. The maximum errors between numerical and experimental front-curved PAT head and efficiency under the design condition are 2.8% and 2.1%, respectively. Therefore, the use of the generated computational mesh and numerical calculation presented in this study seems reasonable considering the verification and validation presented in this section.

4. Results and discussion

4.1. Comparison of PAT performances

4.1.1. Comparison of external characteristics

Fig. 8 shows the performance comparison between a back-

curved PAT and a front-curved PAT, in which the change characteristic of the head, shaft power and efficiency can be seen from Fig. 8 (a), (b) and (c) respectively. Fig. 8 (a) shows that the head of the back-curved and front-curved PAT has the same trend with the flow rate, the head increases with the increase of flow rate. However, the head increase rate of the back-curved PAT is higher than that of the front-curved PAT. The head difference between back-curved and front-curved PAT basically increases with the increase of flow rate. Under part-load condition ($0.8Q_d$), design condition (Q_d) and over-load condition ($1.2Q_d$), the head of the back-curved PAT is 0.4%, 3.4% and 7.9% higher than that of the front-curved PAT, respectively. Fig. 8 (b) shows that the shaft power of the back-curved and front-curved PAT has the increasing trend with the flow rate and that the shaft power of the back-curved and front-curved PAT is basically same under the same flow rate condition. Fig. 8 (c) shows that the efficiencies of the back-curved and front-curved PAT both first increase rapidly and then decrease slowly with the increase of flow rate. However, generally speaking, the high efficiency range of front-curved PAT is wider than that of back-curved PAT, and the efficiency of front-curved PAT is higher than that of back-curved PAT under the same flow rate condition. More specifically, the best efficiency of front curved PAT locates at high flow rate condition and it is 6.9% higher than that of back-curved PAT. Under part-load condition ($0.8Q_d$), design condition (Q_d) and

over-load condition ($1.2Q_d$), the efficiency of the front-curved PAT is 0.6%, 5.9% and 7.9% higher than that of the back-curved PAT, respectively. Therefore, the performance of front-curved PAT, especially the efficiency, is generally better than that of back-curved PAT.

4.1.2. Comparison of energy loss

Fig. 9 shows a comparison of energy loss in two PATs based on the entropy generation. The output power of the back-curved and front-curved PAT are basically the same, and the P_{egr} of the back-curved and front-curved PAT has the same increasing trend with the increase of flow rate. However, the P_{egr} of back-curved PAT is higher than that of front-curved PAT under the same flow rate condition, and the difference of P_{egr} increases with the increase of flow rate. Under part-load condition ($0.8Q_d$), design condition (Q_d) and over-load condition ($1.2Q_d$), the P_{egr} of front-curved PAT is 9.9%, 15.3% and 24% lower than that of back-curved PAT, respectively. Therefore, the reason why the efficiency of front-curved PAT is higher than that of back-curved PAT is that the energy loss of front-curved PAT is lower than that of back-curved PAT.

4.2. Entropy generation power of PAT components

In order to study the characteristics of energy loss, the P_{egr}

Fig. 8. Comparison of performance of PAT (a) Head versus flow rate (b) Shaft power versus flow rate (c) Efficiency versus flow rate.

Fig. 9. Comparison of PAT energy loss.

distributions in each PAT system and their components are shown in Fig. 10. For the back-curved PAT shown in Fig. 10 (a), the P_{egr} values in the back-curved PAT system and each component all increase with the increase of flow rate. In addition, comparing the P_{egr} in different components, it can be found that the P_{egr} in chamber is the largest from part-load condition to $1.2Q_d$ condition, which shows that the energy loss in chamber has a great influence on the performance of the PAT, this is because there are not only leakage loss, wall friction loss, but also eddy current loss in the chamber. The P_{egr} in impeller is only lower than that in chamber. The growth rate of P_{egr} is small from part-load condition to $1.0Q_d$ condition, and increases after $1.0Q_d$ condition. The P_{egr} in volute is smaller than that in chamber and impeller, but the growth rate in volute is the largest under full flow rate condition. The P_{egr} in the volute is lower than that in the impeller from part-load condition to $1.0Q_d$ condition. The P_{egr} in the impeller and volute has almost the same growth rate after $1.0Q_d$ condition, and the P_{egr} is almost the same. The P_{egr} in the draft tube is the lowest under the full flow rate condition, which indicates that the draft tube has little effect on the energy loss in the PAT.

(a)

(b)

Fig. 10. The P_{egr} distributions in PAT system and different components (a) Back-curved PAT (b) Front-curved PAT.

For the front-curved PAT shown in Fig. 10 (b), the P_{egr} in front-curved PAT system, chamber, volute and draft tube all increases with the increase of flow rate, however, the P_{egr} in impeller first decreases and then increases with the increases of flow rate, and reaches the minimum value under $1.0Q_d$ condition. The P_{egr} in chamber is the largest under $0.8Q_d - 1.2Q_d$ condition, the growth rate is small. But overall, the change range of P_{egr} in front-curved impeller is very small. The P_{egr} in impeller is only lower than that in chamber before $0.8Q_d$ condition. The P_{egr} in impeller is lower than that in chamber and volute after $0.8Q_d$ condition. The increase rate of P_{egr} in volute is the highest under the full flow rate condition. The P_{egr} in the volute is only lower than that in the chamber under $0.8Q_d - 1.2Q_d$ condition. The P_{egr} in the draft tube is the lowest under the full flow rate condition.

By comparing the P_{egr} of the back-curved PAT and front-curved PAT, it is shown that the P_{egr} values have obvious differences between back-curved and front-curved impeller, and the P_{egr} of front-curved PAT is lower than that of the back-curved PAT. The P_{egr} in the front-curved impeller changes little with the flow rate, and maintains a low energy loss in a wide flow range. This is the fundamental reason why the performance of front-curved PAT maintains high efficiency under multiple working conditions.

4.3. Comparison of different entropy generation power

In order to analyze the main loss sources of different PAT components, Fig. 11 shows the values of P_{vol} and P_{wall} under $0.8Q_d$, $1.0Q_d$ and $1.2Q_d$ conditions. As shown in Fig. 11(a), the values of P_{vol} in the back-curved and front-curved impeller are much higher than the values of P_{wall} from $0.8Q_d$ to $1.2Q_d$ condition. And the P_{vol} and the P_{wall} values increase with the increase of flow rate. By comparing the different entropy generation power items between two PATs impellers, it is found that the P_{vol} and P_{wall} values of the front-curved PAT are significantly lower than that of the back-curved PAT. This reveals that the energy loss in the impeller of the front-curved PAT is much lower than that of the back-curved PAT.

Fig. 11 (b) shows the different entropy generation power terms in the two PATs chamber. The results show that the P_{wall} in the chamber is significantly higher than the P_{vol} , which indicates that the energy loss in the chamber mainly comes from wall friction, and the force of wall friction increases with the increase of flow, so more wall friction loss is caused. Another point is that the P_{vol} and P_{wall} in the front-curved PAT chamber are lower than those of the back-

Fig. 11. Different entropy generation power items of PAT components (a) Impeller (b) Chamber (c) Volute.

curved PAT chamber, which is also the part of the reason why the energy loss of the front-curved PAT is much lower than that of the back-curved PAT.

As for the comparison of P_{wall} and P_{vol} in the two PATs volutes, Fig. 11 (c) shows that the P_{wall} and the P_{vol} values in the volute are increase with the increase of flow rate. Besides, it is found that the P_{vol} and P_{wall} values in the volute of the front-curved PAT are slightly higher than those of the back-curved PAT, which indicates that the impeller structure has little influence on the internal performance of the volute.

Finally, comparing Fig. 11(a), (b) and (c), it can be found that these two PAT impellers design mainly leads to big difference of entropy generation power in the impeller, the P_{vol} and P_{wall} in the front-curved PAT impeller are much lower than that in the back-curved PAT impeller. The P_{vol} and P_{wall} in front-curved PAT volute is not significantly different from that of back-curved PAT. Therefore, in the following section, the internal flow characteristics in the

impeller that is related to the entropy generation rate will be analyzed in detail.

4.4. Comparison of volume entropy generation rate in impeller

Fig. 12 shows the distribution of different volume entropy generation rate items at middle section of impeller. The results show that the W_{tur} is significantly higher than W_{vis} in impeller, and the area where the W_{tur} is higher is consistent with the area where the W_{vis} is higher. This indicates that the energy loss in the impeller mainly comes from turbulence dissipation. The W_{tur} in the front-curved impeller is lower than that in the back-curved impeller at the same position. This indicates that the turbulence dissipation at the front-curved impeller blade leading edge is lower than that of the back-curved impeller blade leading edge, which is the main reason why the energy loss of the front-curved PAT is lower than that of the back-curved PAT.

Fig. 12. Different volume entropy generation rate items of the two PATs impellers.

4.5. Distribution of turbulent entropy generation rate in impeller under multi-conditions

In order to further study the mechanism of energy loss in the impeller, the turbulent entropy generation rate and velocity distribution in the middle section of PAT impeller were comparatively analyzed. Fig. 13 shows the W_{tur} and velocity distributions of back-curved impeller under $0.8Q_d$, $1.0Q_d$ and $1.2Q_d$, respectively. It can be seen that the losses in the impeller are mainly concentrated on the leading edge and trailing edge of the impeller. According to the velocity distribution in the impeller, the losses in the impeller can be divided into four categories: A, B, C and D. The type A loss is mainly caused by the shock between fluid and blade, which mainly occurs on the blade pressure surface. The type B loss is mainly caused by the separation of fluid and blade, which mainly occurs on the blade suction surface. The type C loss is mainly caused by the jet

wake, which is mainly concentrated on the blade trailing edge. The type D loss is mainly caused by the velocity gradient when the vortex flow converges with the main flow. This kind of loss is mainly concentrated around the vortex flow. Due to different operating conditions, the main loss area caused by different generation mechanisms in the impeller changes. With the increase of the flow rate, the loss of the pressure surface at the leading edge of the impeller decreases due to the flow impact, while the loss caused by the flow separation increases, and the loss area at the trailing edge of the impeller due to the wake jet decreases. For type D loss, the main concentration position in the impeller is different because of the different operating conditions. Under $0.8Q_d$ condition, obvious vortices are generated on the suction surface of the blade trailing edge. The velocity gradient is generated when the vortex flow converges with the main flow, which results in significant energy loss. Under $0.8Q_d$ condition, the vortices are generated on

Fig. 13. Distributions of the W_{tur} and velocity in back-curved impeller (a) W_{tur} (b) velocity.

the suction surface in the middle and upstream of the blade, so the energy loss occurs when the vortexes flow converge with the main flow in the middle of the impeller.

Fig. 14 shows the W_{tur} and velocity distributions of front-curved impeller under $0.8Q_d$, $1.0Q_d$ and $1.2Q_d$, respectively. It can be seen that the energy losses in the front-curved impeller are also mainly concentrated on the leading edge and trailing edge of the impeller. However, compared with the energy losses in the back-curved impeller, the loss areas in the front-curved impeller due to different generation mechanism are significantly reduced. The leading edge loss of front-curved impeller is mainly concentrated

on the blade pressure surface, which is caused by flow shock. The trailing edge loss of front-curved impeller is mainly caused by wake jet. In addition, the wake flow interacts with the main stream, thus forming an obvious turbulent zone with large loss. With the increase of the flow rate, the loss area of the suction surface at the leading edge of the impeller due to the flow impact gradually decreases, and the flow pattern in the impeller is improved. At the trailing edge of the blade, the interaction between the wake flow and the main flow is gradually weakened, and the loss is also reduced.

Fig. 14. Distributions of the W_{tur} and velocity in front-curved impeller (a) W_{tur} (b) velocity.

4.6. Velocity triangle variation of impeller inlet under multi-conditions

For further study the vortex generation mechanism at impeller leading edge under multi-conditions, the velocities of point P1 and point P2 at impeller inlet were solved, and corresponding velocity triangle were made, as shown in Fig. 15. Fig. 15 (a) shows the velocity triangle change of point P1 at back-curved impeller inlet under multi-conditions. With the increase of flow rate, the axial velocity gradually increases from c_{ma} to c_{mc} , and the circumferential velocity gradually increases from c_{ua} to c_{uc} . At the same time, the

relative flow angle at the inlet correspondingly increases from β_{1a} to β_{1c} . There is a certain deviation between the inlet relative flow angle and the inlet blade setting angle (β_1). When the operating flow rate deviates from the design flow, the greater the angle of attack, the greater the shock, vortex and other hydraulic losses.

Fig. 15 (b) shows the velocity triangle variation characteristics of front-curved impeller inlet under multi-conditions. The variation of axial and circumferential partial velocity of front-curved impeller inlet with flow rate is consistent with that of back-curved impeller, and the relative flow angle of impeller inlet increases with the increase of flow rate. Under $0.8Q_d$ condition, the relative flow angle at

Fig. 15. Velocity triangle at impeller inlet (a) Back-curved impeller (b) Front-curved impeller.

the impeller inlet is far less than the inlet setting angle ($\beta'_1 = 90^\circ$), and the fluid impacts the pressure surface at the blade inlet, resulting in a large area of vortex and blocking the flow passage. Under $1.2Q_d$ condition, the relative flow angle at the blade inlet is basically consistent with the blade setting angle, so the flow pattern in the impeller is good and the loss is low. Therefore, the matching of the flow relative angle and the setting angle at the impeller inlet is the fundamental cause of the loss in the impeller.

4.7. Comparison of wall entropy generation rate in the impeller

Fig. 16 shows the distribution of wall entropy generation rate on the two PATs blades, shroud and hub in order to further analyze the wall loss in the impeller. For back-curved blades, the higher W_{wall} is concentrated on the suction surface of the blade leading edge and trailing edge. For front-curved blades, the higher W_{wall} is only concentrated on the blade outlet. Combined with the analysis in Section 4.1, it can be seen that the value of W_{wall} is closely related to the flow pattern inside the impeller, namely, where the flow pattern is poor, the shear stress is correspondingly large. The friction loss on the suction surface of the front-curved blade is much reduced than that of the back-curved blade. For the shroud and hub of the back-curved impeller, the positions where the W_{wall} is higher are mainly concentrated on the suction surface at the blade leading edge and the pressure surface at the blade trailing edge, which indicates that the energy loss caused by the wall friction is larger.

But the W_{wall} of the hub is larger than that of the shroud at the same location, and the W_{wall} of the front-curved hub is significantly lower than that of the back-curved hub. Therefore, the total W_{wall} of the front-curved impeller is lower than that of the back-curved impeller, which is also an important reason why the energy loss in the front-curved impeller is lower than that of the back-curved impeller.

5. Conclusions

In this paper, the back-curved and front-curved PATs were numerically and experimentally studied. The experimental results verified the accuracy of the numerical calculation results. The entropy generation theory was used to analyze performance and energy loss characteristics. The following conclusions were drawn.

The performance of the front-curved PAT is better than that of the back-curved PAT. The best efficiency point of the front-curved PAT is towards large flow rate condition, and the high-efficiency area of the front-curved PAT is obviously wider than that of the back-curved PAT. Under the part-load condition ($0.8Q_d$), design condition (Q_d) and over-load condition ($1.2Q_d$), the efficiency of the front-curved PAT is 0.6%, 5.9%, and 7.9% higher than that of the back-curved PAT, respectively.

Under part-load condition ($0.8Q_d$), design condition (Q_d) and over-load condition ($1.2Q_d$), the entropy generation power of the front-curved PAT is 9.9%, 15.3%, and 24% lower than that of the back-

Fig. 16. Counter of wall entropy generation rate at different position (a) Impeller blades (b) Shroud (c) Hub.

curved PAT, respectively. The entropy generation power of the front-curved PAT is very little affected by the flow rate, and the energy loss keeps low in a wide flow rate range. This is the main reason why the energy loss of the front-curved PAT is lower than that of the back-curved PAT.

The energy loss in the impeller mainly comes from the turbulent entropy generation rate. The larger turbulent entropy generation

rate is mainly concentrated on the blade leading edge, trailing edge and the wake area. Flow separation and flow shock caused by the mismatch between the relative flow angle and the blade setting angle are the main mechanisms of energy loss in impeller. In addition, the loss caused by the wall friction in the front-curved impeller is less than that in the back-curved impeller.

Credit author statement

Bing Qi: Writing – original draft, preparation, Conceptualization, Software, Experiment. Desheng Zhang: Conceptualization, Supervision, Methodology. Linlin Geng: Writing – review & editing, Investigation. Ruijie Zhao: Conceptualization, Data curation, Guidance. Bart P.M. van Esch: Review & editing.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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References

- Binama M, Su W-T, Li X-B, Li F-C, Wei X-Z, An S. Investigation on pump as turbine (PAT) technical aspects for micro hydropower schemes: a state-of-the-art review. *Renew Sustain Energy Rev* 2017;79:148–79.
- Zhang Y, Zhang Y, Wu Y. A review of rotating stall in reversible pump turbine. *Proceedings of the Institution of Mechanical Engineers, Part C: J Mech Eng Sci* 2016;231(7):1181–204.
- Zhang Y, Chen T, Li J, Yu J. Experimental study of load variations on pressure fluctuations in a prototype reversible pump turbine in generating mode. *J Fluid Eng* 2017;139(7).
- Fecarotta O, Aricò C, Carravetta A, Martino R, Ramos HM. Hydropower potential in water distribution networks: pressure control by PATs. *Water Resour Manag* 2014;29(3):699–714.
- Hao Y, Tan L. Symmetrical and unsymmetrical tip clearances on cavitation performance and radial force of a mixed flow pump as turbine at pump mode. *Renew Energy* 2018;127:368–76.
- Singh P. Optimization of internal hydraulics and of system design for pumps as turbines with field implementation and evaluation. Ph.D. dissertation. Karlsruhe, Germany: University of Karlsruhe; 2005.
- Qian Z, Wang F, Guo Z, Lu J. Performance evaluation of an axial-flow pump with adjustable guide vanes in turbine mode. *Renew Energy* 2016;99:1146–52.
- Tan X, Engeda A. Performance of centrifugal pumps running in reverse as turbine: Part II- systematic specific speed and specific diameter based performance prediction. *Renew Energy* 2016;99:188–97.
- Derakhshan S, Nourbakhsh A. Experimental study of characteristic curves of centrifugal pumps working as turbines in different specific speeds. *Exp Therm Fluid Sci* 2008;32(3):800–7.
- Fernández J, Blanco E, Parrondo J, Stickland MT, Scanlon TJ. Performance of a centrifugal pump running in inverse mode. In: *Proceedings of the institution of mechanical engineers, Part A: journal of power and energy*, vol. 218; 2004. p. 265–71. 4.
- Frosina E, Buono D, Senatore A. A performance prediction method for pumps as turbines (PAT) using a computational fluid dynamics (CFD) modeling approach. *Energies* 2017;10(1):103.
- Yang S-S, Derakhshan S, Kong F-Y. Theoretical, numerical and experimental prediction of pump as turbine performance. *Renew Energy* 2012;48:507–13.
- Stepanoff AJ. Centrifugal and axial flow pumps, centrifugal and axial flow pumps. 1947.
- Tao W, Kong F, Yang S, Fu Y. Numerical study on hydraulic performances of pump as turbine with forward-curved blades. 2014.
- Wang T, Wang C, Kong F, Gou Q, Yang S. Theoretical, experimental, and numerical study of special impeller used in turbine mode of centrifugal pump as turbine. *Energy* 2017;130:473–85.
- Yang S-S, Kong F-Y, Qu X-Y, Jiang W-M. Influence of blade number on the performance and pressure pulsations in a pump used as a turbine. *J Fluid Eng* 2012;134(12).
- Li D, Sun Y, Zuo Z, Liu S, Wang H, Li Z. Analysis of pressure fluctuations in a prototype pump-turbine with different numbers of runner blades in turbine mode. *Energies* 2018;11(6).
- Yang S-S, Liu H-L, Kong F-Y, Xia B, Tan L-W. Effects of the radial gap between impeller tips and volute tongue influencing the performance and pressure pulsations of pump as turbine. *J Fluid Eng* 2014;136(5).
- Derakhshan S, Mohammadi B, Nourbakhsh A. Efficiency improvement of centrifugal reverse pumps. *J Fluid Eng* 2009;131(2).
- Yang S-S, Liu H-L, Kong F-Y, Dai C, Dong L. Experimental, numerical, and theoretical research on impeller diameter influencing centrifugal pump-as-turbine. *J Energy Eng* 2013;139(4):299–307.
- Shehata AS, Saqr KM, Xiao Q, Shehadeh MF, Day A. Performance analysis of wells turbine blades using the entropy generation minimization method. *Renew Energy* 2016;86:1123–33.
- Duan L, Wu X, Ji Z, Fang Q. Entropy generation analysis on cyclone separators with different exit pipe diameters and inlet dimensions. *Chem Eng Sci* 2015;138:622–33.
- Qian B, Chen JP, Wu P, Wu DZ, Yan P, Li SY. Investigation on inner flow quality assessment of centrifugal pump based on Euler head and entropy production analysis. *IOP Conf Ser Earth Environ Sci* 2019;240(9):8. 092001.
- Pei J, Meng F, Li Y, Yuan S, Chen J. Effects of distance between impeller and guide vane on losses in a low head pump by entropy production analysis. *Adv Mech Eng* 2016;8(11).
- Li D, Gong R, Wang H, Xiang G, Wei X, Qin D. Entropy production analysis for hump characteristics of a pump turbine model. *Chin J Mech Eng* 2016;29(4):803–12.
- Gong R, Wang H, Chen L, Li D, Zhang H, Wei X. Application of entropy production theory to hydro-turbine hydraulic analysis. *Sci China Technol Sci* 2013;56(7):1636–43.
- Soltanmohamadi R, Lakzian E. Improved design of Wells turbine for wave energy conversion using entropy generation. *Meccanica* 2016;51(8):1713–22.
- Li X, Zhu Z, Li Y, Chen X. Experimental and numerical investigations of head-flow curve instability of a single-stage centrifugal pump with volute casing. In: *Proceedings of the institution of mechanical engineers, Part A: journal of power and energy*, vol. 230; 2016. p. 633–47. 7.
- Li X, Jiang Z, Zhu Z, Si Q, Li Y. Entropy generation analysis for the cavitating head-drop characteristic of a centrifugal pump. In: *Proceedings of the institution of mechanical engineers, Part C: journal of mechanical engineering science*, vol. 232; 2018. p. 4637–46. 24.
- Jia X-Q, Zhu Z-C, Yu X-L, Zhang Y-L. Internal unsteady flow characteristics of centrifugal pump based on entropy generation rate and vibration energy. In: *Proceedings of the institution of mechanical engineers, Part E: journal of process mechanical engineering*, vol. 233; 2019. p. 456–73. 3.
- Cao J, Pei J, Gu Y, Wang W, Yuan S. Flow losses analysis in a mixed flow pump with annular volute by entropy production evaluation. *IOP Conf Ser Earth Environ Sci* 2019;240.
- Lin T, Li X, Zhu Z, Xie J, Li Y, Yang H. Application of enstrophy dissipation to analyze energy loss in a centrifugal pump as turbine. *Renew Energy* 2021;163:41–55.
- Li D, Wang H, Qin Y, Han L, Wei X, Qin D. Entropy production analysis of hysteresis characteristic of a pump-turbine model. *Energy Convers Manag* 2017;149:175–91.
- Kazbekov A, Kumashiro K, Steinberg AM. Enstrophy transport in swirl combustion. *J Fluid Mech* 2019;876:715–32.
- Kock F, Herwig H. Local entropy production in turbulent shear flows: a high-Reynolds number model with wall functions. *Int J Heat Mass Tran* 2004;47(10–11):2205–15.
- Zhang F, Appiah D, Hong F, Zhang J, Yuan S, Adu-Poku KA, Wei X. Energy loss evaluation in a side channel pump under different wrapping angles using entropy production method. *Int Commun Heat Mass Tran* 2020;113.
- Hou H, Zhang Y, Li Z, Jiang T, Zhang J, Xu C. Numerical analysis of entropy production on a LNG cryogenic submerged pump. *J Nat Gas Sci Eng* 2016;36:87–96.
- Menter FR. Two-equation eddy-viscosity turbulence models for engineering applications. *AIAA J* 1994;32(8):1598–605.
- Arat H, Arslan O, Ercetin U, Akbulut A. Experimental study on heat transfer characteristics of closed thermosyphon at different volumes and inclination angles for variable vacuum pressures. *Case Studies in Thermal Engineering* 2021;26.
- Arat H, Arslan O, Ercetin U, Akbulut A. A comprehensive numerical investigation of unsteady-state two-phase flow in gravity assisted heat pipe enclosure. *Thermal Science and Engineering Progress* 2021;25.