

We are IntechOpen, the world's leading publisher of Open Access books Built by scientists, for scientists

6,900

Open access books available

185,000

International authors and editors

200M

Downloads

Our authors are among the

154

Countries delivered to

TOP 1%

most cited scientists

12.2%

Contributors from top 500 universities



WEB OF SCIENCE™

Selection of our books indexed in the Book Citation Index
in Web of Science™ Core Collection (BKCI)

Interested in publishing with us?
Contact book.department@intechopen.com

Numbers displayed above are based on latest data collected.
For more information visit www.intechopen.com



Current Development and Prospect of Turbine in OTEC

Weimin Liu, Yunzheng Ge, Lei Liu and Yun Chen

Abstract

This chapter mainly introduces the development and prospect of turbines utilized in ocean thermal energy conversion (OTEC), including brief introduction, aerodynamic design, mechanical and electric control system, problems, and prospect of the turbine in OTEC. At the beginning, the first section mainly introduces compositions and types of turbine in OTEC systems, different working fluids in the turbine, status of OTEC turbine currently in the world, and so on. After that, the aerodynamic design of turbine has the greatest impact on the efficiency of the turbine and is the most important process during the turbine development. Therefore, the second section focuses on turbine aerodynamic design. Furthermore, the third section mainly introduces the mechanical system and the electric control system of the turbine. Finally, the problems in the turbine currently are listed, and the prospects for the development of the turbine in OTEC are discussed in the fourth section.

Keywords: OTEC, radial-inflow turbine, aerodynamic design, mechanical system, electric control system

1. Introduction

This section mainly introduces compositions and types of turbine in ocean thermal energy conversion (OTEC) systems, different working fluids in turbine, current status of OTEC turbine in the world, and so on. In OTEC system, a turbine rotates at a high speed driven by a heated gas and then drives the generator to generate electricity. So turbine is the key equipment that converts ocean thermal energy into mechanical energy and finally output electricity. Moreover, the output power and thermal efficiency of OTEC system change with that of turbine, so the performance of turbine may directly affect the whole performance of OTEC system.

Since the temperature of working fluid flowing through the turbine such as ammonia, water, and so on is low and the enthalpy drop is small, OTEC power generation system is different from other high-temperature power cycles like steam Rankine cycle or gas turbine cycle. The inefficiency of overall power generation system about 3–5% requires that the turbine must have high efficiency. Since the efficiency of OTEC system which adopts an open cycle is low, a closed cycle is usually used to obtain more electricity. And in closed cycle, internal working environment is relatively good for turbine, and design difficulty is relatively low. In addition, since organic working fluid or ammonia is used as working fluid, turbine requires high performance of sealing, and sealing material should not be affected by working fluid. Furthermore, special lubricating oils suitable for organic working

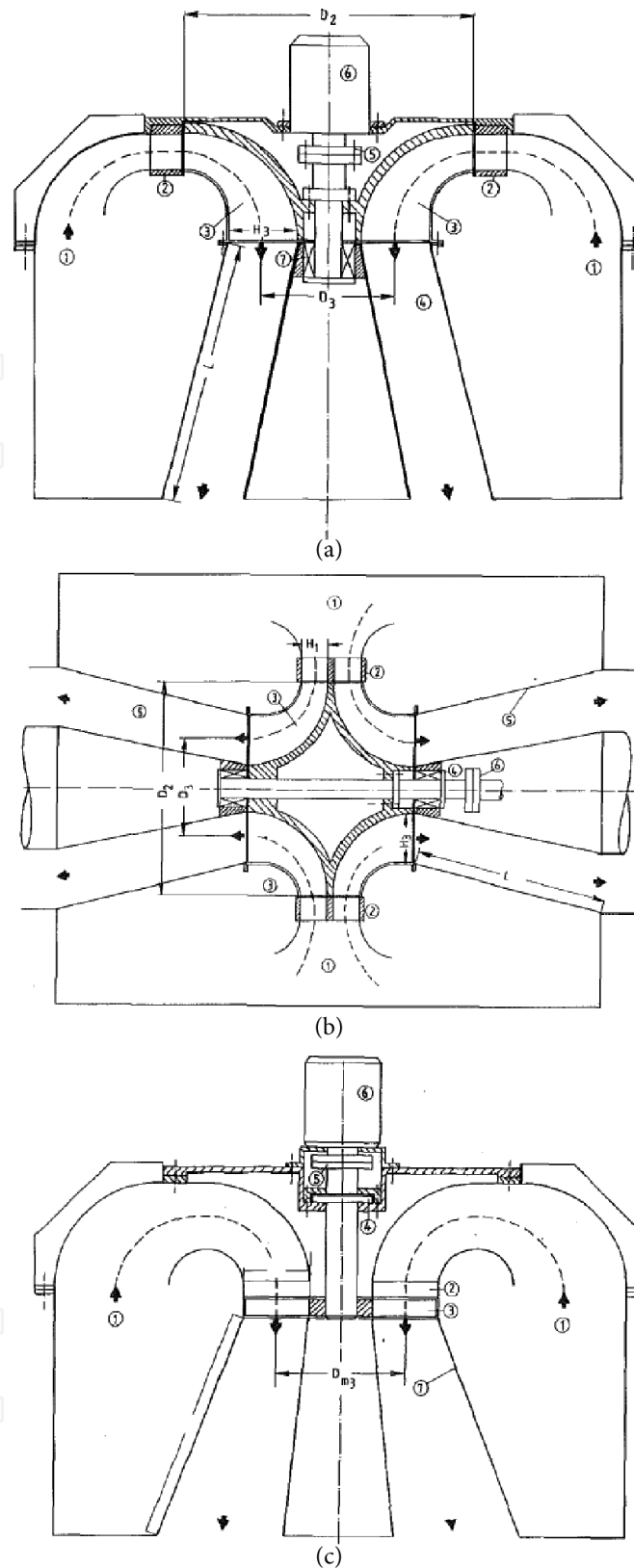


Figure 1.

Different types of turbines (a) single-inflow radial steam turbine: 1 = inlet from evaporator, 2 = stator, 3 = rotor, 4 = exit diffuser, 5 = coupling, 6 = generator, 7 = bearing, (b) double-inflow radial steam turbine: 1 = inlet from evaporator, 2 = stator, 3 = rotor, 4 = bearing, 5 = exit diffuser, 6 = coupling, (c) single-stage axial steam turbine: 1 = inlet from evaporator, 2 = stator, 3 = rotor, 4 = bearing, 5 = coupling, 6 = generator, 7 = exit diffuser.

fluids or ammonia should be selected. For large-scale systems, special lubricating oil separation devices or lubricating oil systems should also be used.

The efficiency of axial turbine, which was once used in open cycle of OTEC system, is lower than that of radial type. For example, the efficiency of single-inlet and double-inlet radial turbine in the Hawaii Natural Energy Institute (HNEI) is

increased by about 10% compared with that of axial turbine in Ref. [1], which is shown in **Figure 1**. Therefore, it is recommended to adopt radial turbine in open cycle. Similarly, radial turbine is often used in closed cycle of OTEC power generation system, considering the requirements of improving turbine efficiency and impeller strength.

In closed cycle of OTEC system, different working fluids are used depending on different systems. For Rankine cycle, organic working fluids are usually used as working fluid, such as R600a, R245fa, R152a, R22, and R134a, in Ref. [2, 3]; in Karina cycle and Uehara cycle, ammonia-water mixture is adopted as working fluid in Ref. [4, 5]. Therefore, in OTEC systems adopting different thermodynamic cycles, most of the working fluids in turbine are ammonia or other organic working fluids. The main difference is that for system using ammonia-water mixture, working fluids in turbine are almost pure ammonia, so turbine design can be carried out according to that using pure ammonia as working fluid.

The turbine used in 30 kW OTEC system of Saga University in Japan is a two-stage radial turbine with intermediate extraction. For 100 kW OTEC power generation unit built in Maja area of Kume Island, radial turbine is adopted as well as in 100 kW OTEC power plant built by Makai Ocean Engineering in Hawaii. In addition, radial turbines are also used as key equipment to drive generators in OTEC test systems of Korea [6] and China [7]. In view of this, this chapter mainly studies the status, design, and prospect of radial turbine in a closed OTEC system.

2. Aerodynamic design of turbine in OTEC

This section focuses on turbine aerodynamic design in OTEC. In offshore OTEC power generation system which adopts closed cycle, the temperature difference between surface warm water (25°C) and deep cold water (5°C) is only about 20°C, which results in a small temperature difference of working fluid at the inlet and outlet of turbine. Even if ammonia with high expansion ratio is used, the pressure ratio between inlet and outlet of turbine is still very low. At the same time, due to low efficiency of OTEC thermodynamic cycle, the efficiency of turbine restricts efficiency improvement in OTEC power plant. So in the turbine design process, the influence of aerodynamic design on turbine efficiency improvement is the key content of turbine research.

2.1 Principle and characteristics of turbine in OTEC

The radial turbine is a high-speed and rotating machine, which can convert thermal energy of high-temperature and high-pressure working fluid into mechanical energy. The working fluid flows from turbine inlet to volute at a certain initial speed, and then the airflow is gradually distributed evenly during the flow process, and furthermore the gas expands and accelerates to obtain a higher speed in the guide nozzle. The guide is mainly a nozzle ring composed of many fixed stator blades, namely, radial guide nozzle. The velocity of gas at nozzle outlet increases, and the pressure and temperature decrease. And then the gas flows into the high-speed rotating impeller at a relative speed. The gas continues to expand in impeller, and the pressure of working fluid at impeller outlet is reduced. Finally, working fluid is discharged to the subsequent process through outlet diffuser.

The radial turbine in OTEC system has the following characteristics:

1. Both residual velocity loss and flow loss inside the turbine are relatively small, so it has higher internal efficiency.

2. Working fluid flow can be adjusted by adjustable nozzle, so that the turbine has a wider operating range and enhances the load adaptability and operation stability of the whole system.
3. The efficiency will not be greatly affected by the inaccuracy of blade geometry and surface smoothness. In addition, the turbine has a lighter weight, a simple structure, and fewer parts, which can greatly reduce manufacture cost.
4. The power of radial turbine is limited to some extent by structural factors due to its large radial dimension and high speed.

Based on the above features, radial turbine is especially suitable for small flow rate and low pressure ratio, which is exactly the requirement for turbine of OTEC.

2.2 One-dimensional thermodynamic design and optimization of radial turbine in OTEC

The aerodynamic design of radial turbine mainly includes one-dimensional (1-D) thermodynamic design and three-dimensional (3-D) aerodynamic design of pneumatic parts including volute, nozzle, impeller, and diffuser as shown in **Figure 2**. Many works like [8–11] have introduced the design theory of radial turbine in detail; the readers can learn from them according to research situation. And this chapter only lists a design method for turbine, for example. 1-D thermodynamic design methods for OTEC turbine have been discussed in many literatures, including mean-line model, as shown in [12–15].

The flow of working fluid at the inlet and outlet of impeller can be expressed by velocity triangle shown in **Figure 3**. Absolute velocity c_j is the vector sum of circumferential velocity u_j and relative velocity w_j at inlet and outlet of impeller. The working fluid flows into impeller in direction of absolute flow angle α_1 at the inlet and flows out of impeller in direction of relative flow angle β_2 at outlet. At impeller inlet, relative flow angle β_1 is the angle between relative flow direction and u_1 , while the angle between absolute flow direction and u_2 at the outlet of impeller is the absolute outlet flow angle α_2 .

Firstly, the main 1-D thermodynamic and geometric parameters of radial turbine are calculated based on the given initial data and some other important empirical parameters. Generally speaking, the most important design parameters of turbine include the wheel diameter ratio \bar{D}_2 , degree of reaction Ω , and velocity ratio \bar{w}_1 .

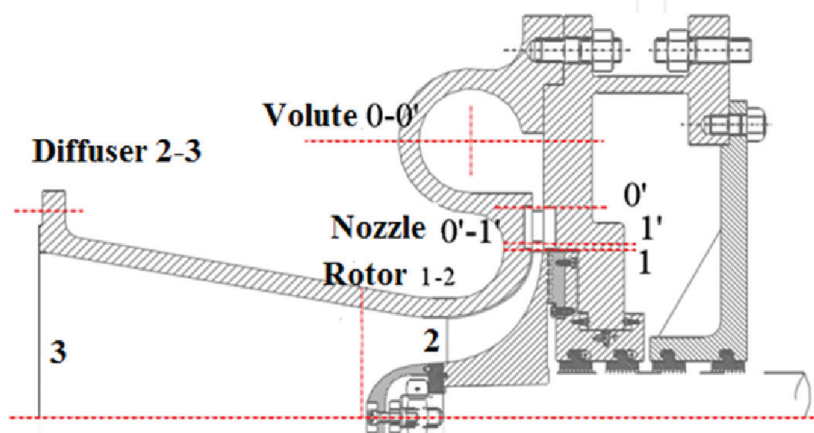


Figure 2.
Schematic diagram of turbine pneumatic structure.

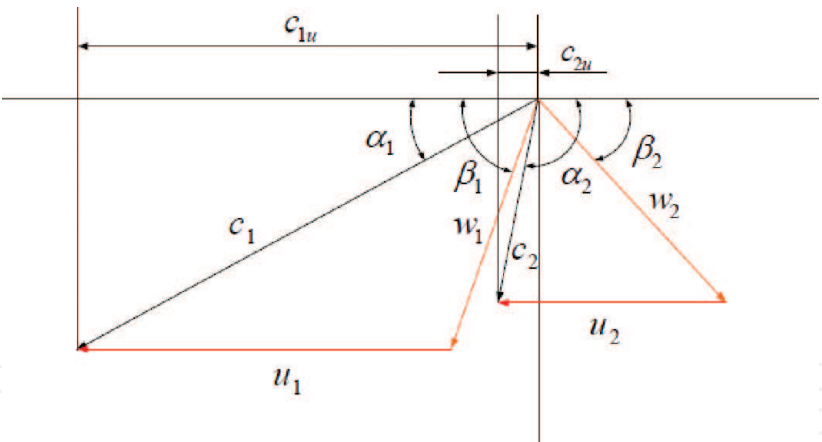


Figure 3.
Velocity triangle of radial turbine impeller.

The wheel diameter ratio refers to the ratio of impeller outlet diameter to inlet diameter, and the wheel diameter ratio determines the turbine working ability, which is generally selected between 0.35 and 0.55. Degree of reaction refers to the ratio of the isentropic enthalpy drop in impeller to the total isentropic enthalpy drop in turbine, which represents the distribution of energy when gas expands in nozzle and impeller. A larger degree of reaction means a faster velocity of the airflow discharged from impeller and more residual velocity loss. When the degree of reaction is too small, deceleration motion will occur. For turbine, it is generally selected between 0.3 and 0.5. The velocity ratio is the ratio of circular velocity at impeller inlet to the ideal velocity under isentropic condition, which reflects the influence of rotational speed on turbine, generally selected between 0.5 and 0.8.

There are many ways to determine the thermodynamic parameters of turbine, such as trial method, optimal velocity ratio method, and screening method. The trial method refers to the calculating wheel efficiency using selected parameters such as degree of reaction, wheel diameter ratio, velocity ratio, inlet nozzle flow angle, and impeller outlet flow angle in order to determine the optimal design. This method has great blindness and requires a lot of work. The optimal velocity ratio method determines the main design parameters such as the velocity ratio and the degree of reaction by interpretation method, such as “zero tail vortex” analysis method and specific velocity analysis method. The screening method is used to analyze the effect of the main parameters such as pressure ratio, Mach number, angle of attack, and allowable stress of wheel in determining turbine velocity ratio and degree in detail. This method has high efficiency and can prevent large changes.

In this chapter, the design of a turbine using ammonia as working fluid for 7.5 kW OTEC is shown as an example. According to the calculation result of cycle, the given thermodynamic design parameters of turbine are shown in **Table 1**.

In order to obtain 1-D design parameters, the important parameters to be determined include wheel diameter ratio \bar{D}_2 , degree of reaction Ω , velocity ratio \bar{u}_1 , velocity coefficient, and blade angle. Velocity coefficient includes those of impeller φ and nozzle ψ . Blade angle includes impeller inlet mounting angle α_1 and nozzle outlet mounting angle β_2 in [16]. The values of these seven basic parameters are selected by the above various methods in proper range. Finally, the range and values of the seven basic parameters selected are listed in **Table 2**. Based on design parameters, thermodynamic parameters in turbine are calculated and 1-D thermodynamic design is completed. The result is shown in **Table 3**.

In the process of 1-D design above, the parameters are determined by using traditional methods. There are, however, limitations in the methods, such as time-

Parameters	Units	Values
Gas constant	N.m/(kg.K)	488.2
Isentropic exponent		1.447
Relative molecular mass		17
Mass flow rate	kg/s	0.388
Inlet pressure	MPa	0.9
Inlet temperature	K	298
Outlet pressure	MPa	0.75

Table 1.
The design parameters of turbine.

Parameters	Units	Ranges	Values
Ω		0.35–0.55	0.48
\overline{D}_2		0.3–0.5	0.38
\overline{u}_1		0.65–0.7	0.656
φ		0.92–0.98	0.96
ψ		0.75–0.9	0.84
α_1	°	12–30	15
β_2	°	20–45	35

Table 2.
The range and values of the basic parameters.

consuming, blindness, and even unreasonable. Therefore based on design parameters obtained through the above methods, an optimization of the relevant parameters is carried out in order to obtain better design parameters. The optimization method mainly determines value ranges of the seven design parameters and establishes the following optimization model as Eq. (1).

$$\eta = 2\overline{u}_1 \left(\varphi \cos \alpha_1 \sqrt{1-\Omega} \overline{D}_2^2 \overline{u}_1 + \overline{D}_2 \psi \cos \beta_2 \sqrt{\Omega + \varphi^2 (1-\Omega) + \overline{D}_2^2 \overline{u}_1^2 - 2\overline{u}_1 \varphi \cos \alpha_1 \sqrt{1-\Omega}} \right) \quad (1)$$

For the above model, an optimization algorithm is needed to obtain the proper solution. **Table 4** lists the results of 1-D thermodynamic optimization using a genetic algorithm program. Compared with the results in **Table 3**, the size of nozzle and impeller has changed slightly, and the airflow angle has changed correspondingly, which results in the reduction of output and a little increase in isentropic efficiency. Although these parameters change a little, improvement of isentropic efficiency shows that the optimal design has obtained a better result.

2.3 Three-dimensional design, simulation, and optimization of radial turbine in OTEC

Aerodynamic design has the greatest impact on the efficiency of turbine and is the most important part in turbine design. Therefore, this part focuses on turbine aerodynamic design and optimization.

Parameters	Units	Results
Impeller speed	r/min	21,000
Nozzle outlet velocity	m/s	155
Nozzle outlet pressure	MPa	0.802
Nozzle outlet temperature	K	291.08
Impeller inlet height	mm	5.8
Impeller inlet absolute flow angle	°	14
Impeller inlet relative flow angle	°	86
Impeller inlet relative velocity	m/s	37
Impeller inlet diameter	mm	120
Wheel peripheral speed	m/s	156
Impeller outlet absolute flow angle	°	101.22
Impeller outlet absolute speed	m/s	49.16
Impeller outlet relative flow angle	°	35
Impeller outlet external diameter	mm	56.2
Impeller outlet inner diameter	mm	31.7
Power generation	kW	7.8
Efficiency of the wheel periphery	—	0.864

Table 3.
The result of 1-D thermodynamic design.

Parameters	Units	Results
Impeller speed	r/min	21,000
Nozzle outlet velocity	m/s	160.5
Nozzle outlet pressure	MPa	0.785
Nozzle outlet temperature	K	290
Impeller inlet height	mm	4.398
Impeller inlet absolute flow angle	°	16
Impeller inlet relative flow angle	°	90
Impeller inlet relative velocity	m/s	40.9
Impeller inlet diameter	mm	126.8
Wheel peripheral speed	m/s	159.4
Impeller outlet absolute flow angle	°	99.364
Impeller outlet absolute speed	m/s	49.16
Impeller outlet relative flow angle	°	35.745
Impeller outlet external diameter	mm	57
Impeller outlet inner diameter	mm	28.8
Power generation	kW	7.65
Isentropic efficiency	—	0.875

Table 4.
The results of 1-D thermodynamic optimization.

2.3.1 Three-dimensional aerodynamic design of radial turbine in OTEC

Based on 1-D parameters obtained in the abovementioned optimization process, 3-D structural design needs to be carried out for aerodynamic components of turbine. The role of volute is to distribute airflow evenly, so the cross-sectional area of volute decreases continuously, which could be realized by using spiral type. In this case, asymmetrical pear-shaped cross section is adopted for volute design in consideration of good guidance of airflow. Compared with other cross-sectional volutes, this type has the characteristic of good flow performance although the entire volute has a larger size. **Figure 4** shows the designed 3-D model of the asymmetrical pear-shaped volute.

Nozzles of turbine are mainly used to make impeller capture the maximum momentum as effectively as possible, so nozzles always need a certain inclination angle. Nozzles have many forms, such as airfoil, straight plate, wedge, island, and so on, but some of these blades have large flow loss or poor aerodynamic effect, which cannot meet the requirements of high-efficiency turbine. In order to achieve a high-efficiency aerodynamic performance of turbine, the NASA-designed Rotor series and the Soviet Union's TC-P series are selected as the original design profiles to obtain the required nozzles optimized. In this case, the TC-2P blade of the Soviet Union is selected to shape the nozzle, and the result is shown in **Figure 5**.

Meridian parameters are the basis of 3-D impeller design. On the basis of 1-D parameters, meridian parameters of the impeller, as shown in **Figure 6**, are calculated firstly as listed in **Table 5**. There are many references for 3-D design of impeller, and kinds of corresponding software are designed using internal algorithms such as ANSYS BladeGen and NREC. In this case, the 3-D design of impeller is carried out in two parts: the working wheel and the induce wheel. Generally, for strength, radial straight blades are used as working wheels. However, nonradial blades are preferred for flow, as shown in the literature [13, 15]. There are requirements for induce wheel such as good aerodynamic performance, geometric shape to meet the flow, better strength characteristic, and feasibility of production. For a better aerodynamic performance, parabolic surface is mainly used for blade modeling of induce wheel. Because the non-developable ruled parabolic of cylindrical base surface has good properties in aerodynamics, strength, process ability, and versatility, it can meet the basic requirements of the induce blade profile. In this case, the induce blade is modeled by a non-developable linear parabolic surface with a cylindrical base. The final designed impeller is shown in **Figure 7**.

For the design of diffuser, it is mainly considered that the gas discharged from turbine impeller still has a high speed, which will cause a large friction loss when

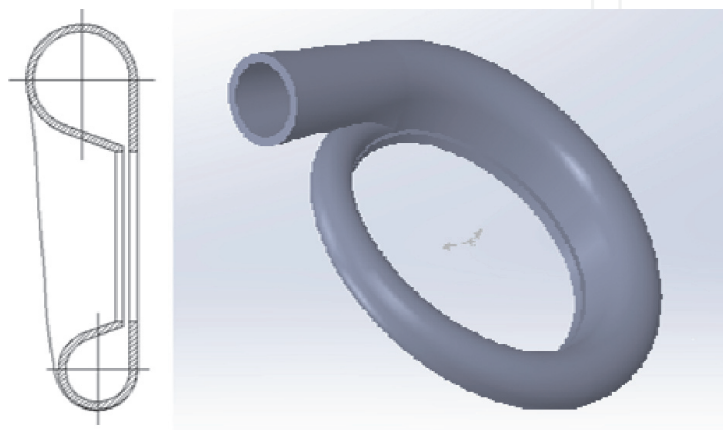


Figure 4.
The 3-D design of asymmetrical pear-shaped volute.

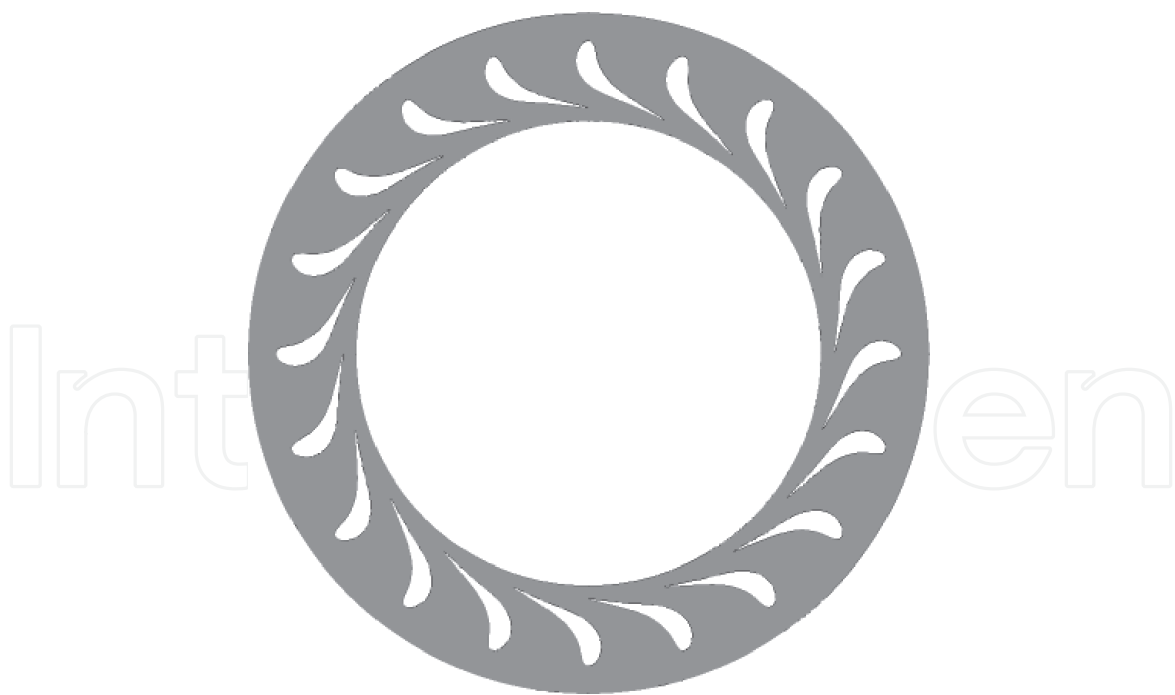


Figure 5.
The 3-D design of nozzle in turbine.

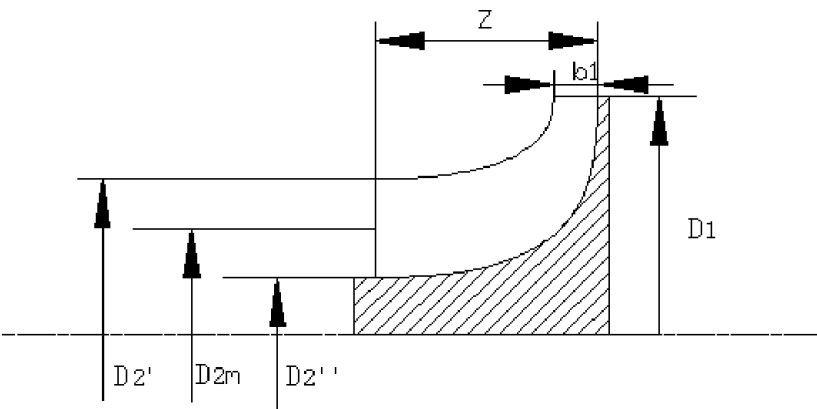


Figure 6.
The meridian plane of impeller.

Parameters	Units	Values
D_1	mm	117
D_2'	mm	57
D_2''	mm	28.8
D_{2m}	°	42.9
Z	mm	27
b_1	mm	4.43
Number of blades	—	12

Table 5.
The parameters of impeller meridian.

discharged into pipeline. Therefore, the function of diffuser is mainly to reduce the airflow velocity further in order to reduce airflow loss. Diffuser usually adopts a relatively simple conical cylinder with a certain expansion angle, and the

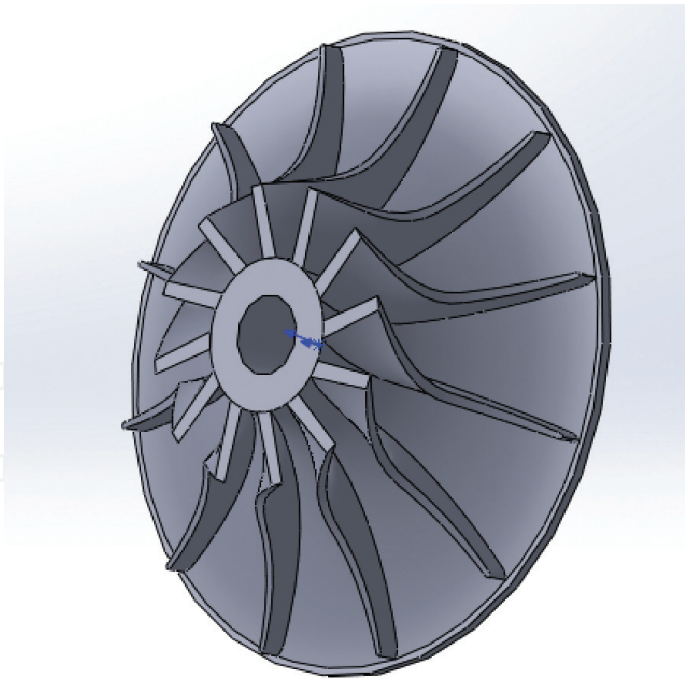


Figure 7.
The 3-D design of impeller.

cross-sectional area is calculated similarly to that of nozzle. **Figure 8** shows the designed diffuser, and the final 3-D design of turbine is shown in **Figure 9**.

2.3.2 Numerical simulation and performance analysis of radial turbine in OTEC

When turbine aerodynamic design in OTEC is completed, numerical simulation is used to study the aerodynamic performance of turbine. The aerodynamic part of turbine needs to be meshed before numerical simulation of turbine aerodynamic performance, and meshing can be carried out in many kinds of commercial software, including ICEM, TurboGrid, etc. The meshing of turbine nozzle and impeller can be carried out by Turbogrid, which can divide the structured grid with high grid quality and a short time. In this case, Turbogrid is used for meshing of nozzle and impeller, by dividing the structured grid with high grid quality and a short time. On the other hand, meshing of volute and diffuser is carried out in ICEM, as it can be used for regular and irregular geometric structures. **Figure 10** shows the grid of turbine.

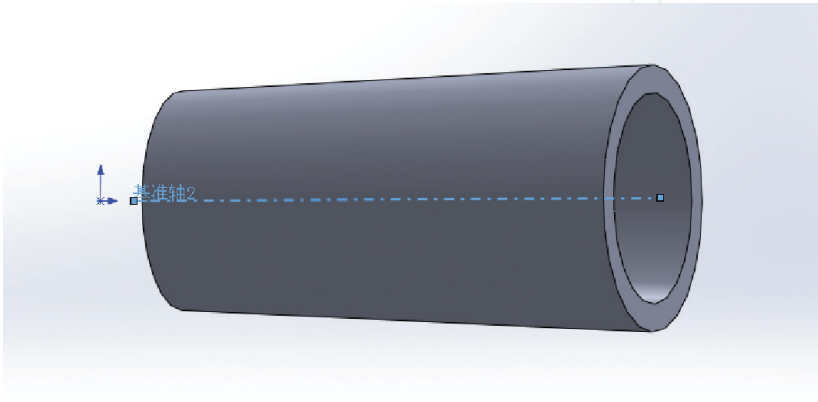


Figure 8.
The 3-D design of diffuser.

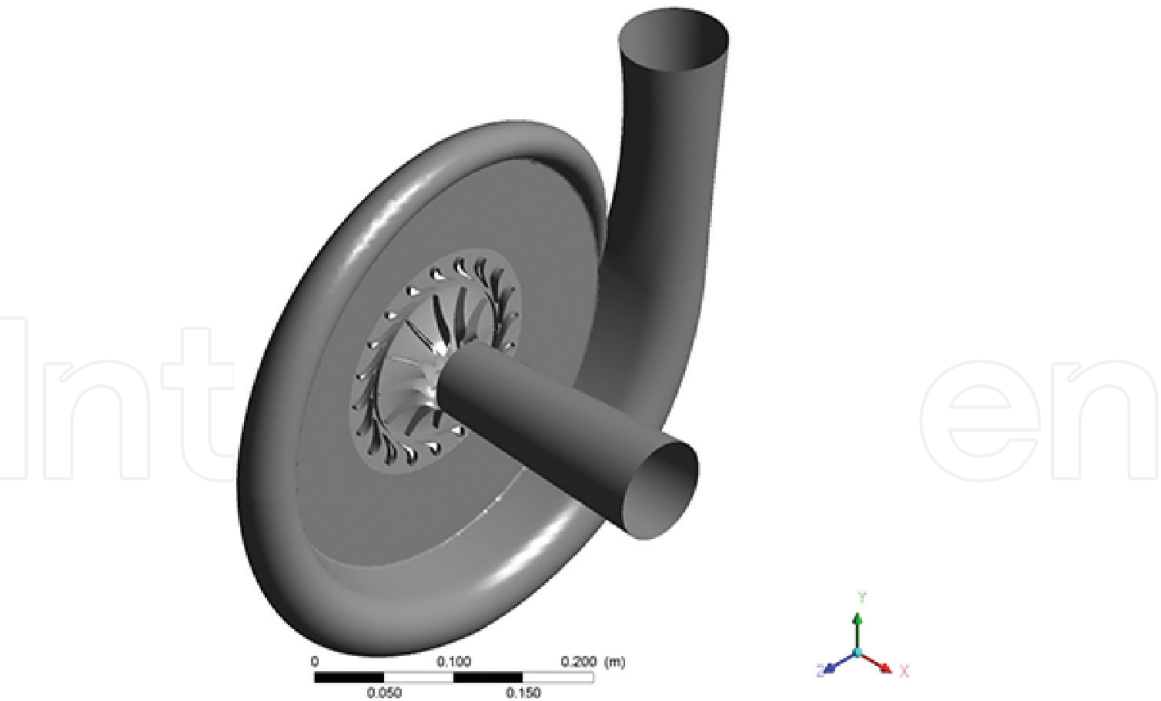


Figure 9.
The 3-D model of turbine.

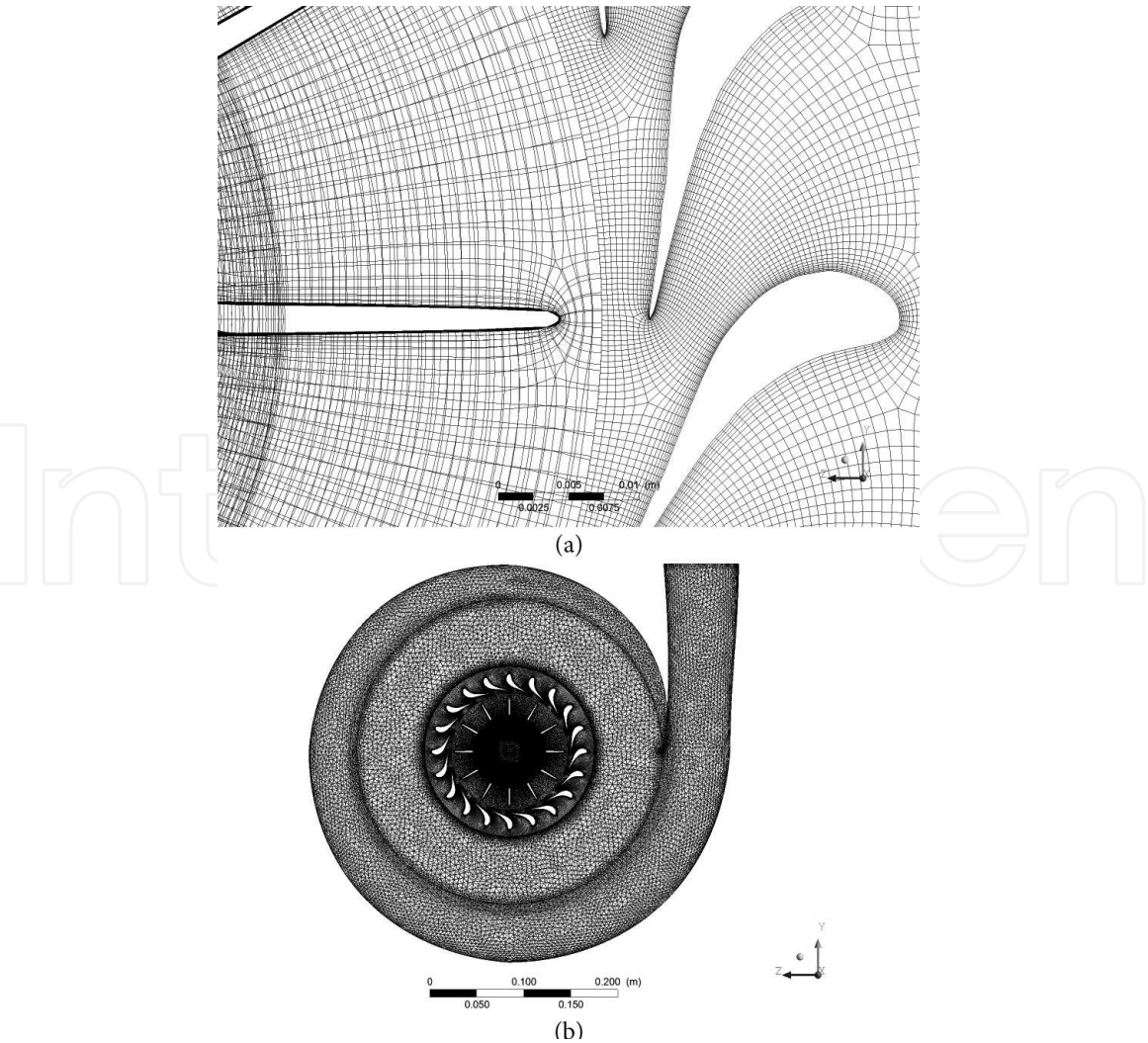


Figure 10.
Grid of radial turbine (a) hexahedral mesh of nozzle and impeller, (b) whole grid of turbine.

After overall meshing of turbine is completed, numerical simulation of turbine needs to be carried by commercial software, such as ANSYS CFX, ANSYS Fluent, and NUMECA. And these kinds of software are basically consistent in the simulation process, and internal algorithms include flow control equation, turbulence model, and solver. In this case, ANSYS CFX is used to simulate the turbine flow, including setting turbulence model, boundary conditions, simulation accuracy, and boundary condition initialization. The simulation method section is not repeated here, since it has been described in various kinds of simulation software.

In order to determine the accuracy of 1-D design, the performance of turbine needs to be simulated and analyzed, and then compared the results with that of 1-D design, as shown in **Table 6**. From the results, it can be seen that the simulation values of the main design parameters are basically the same as those of the design parameters except that the error of nozzle exit velocity is about 5%. Although isentropic efficiency and power are smaller than design values, the error is about 1%, which meets the design requirements.

Thereafter, the flow state of turbine is analyzed through streamline diagram, pressure contour, velocity contour, and Ma number contour. **Figure 11** shows a streamline diagram of turbine, and **Figure 12** visualizes pressure and velocity contour of nozzle and impeller passage section at 50% blade spanwise. As seen from the figures, the overall pressure gradient direction is roughly along the flow direction, the streamline flows uniformly along the nozzle and impeller channel direction, and no obvious back pressure gradient appears in the impeller. On the whole, although flow loss exists at the suction surface of blade, the efficiency of the turbine is higher,

Parameters	Design results	CFD results	Errors (%)
Nozzle outlet velocity ($\text{m}\cdot\text{s}^{-1}$)	160.5	152.5	4.98
Wheel peripheral velocity ($\text{m}\cdot\text{s}^{-1}$)	159.4	156.3	1.94
Mass flow rate ($\text{kg}\cdot\text{s}^{-1}$)	0.388	0.39	0.51
Isentropic efficiency ()	0.875	0.865	1.14
Power (kW)	7.65	7.54	1.44

Table 6.
Comparison of simulation results with design results.

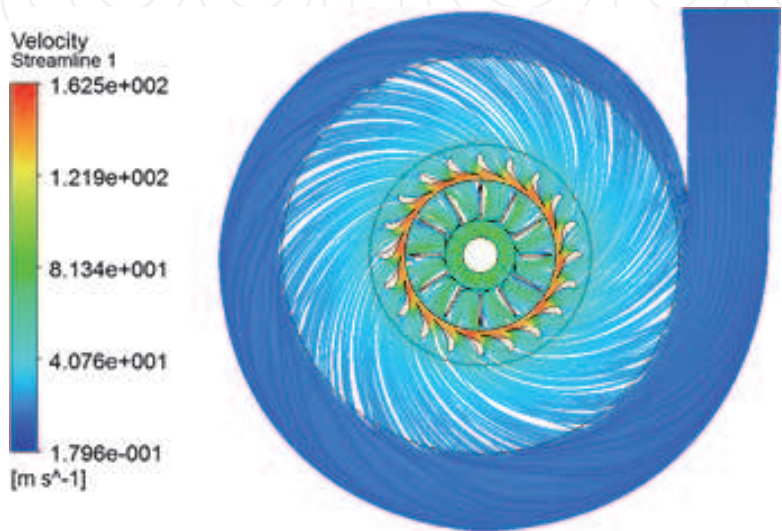


Figure 11.
Turbine overall streamline without including diffuser (a) pressure contour, (b) velocity streamline diagram.

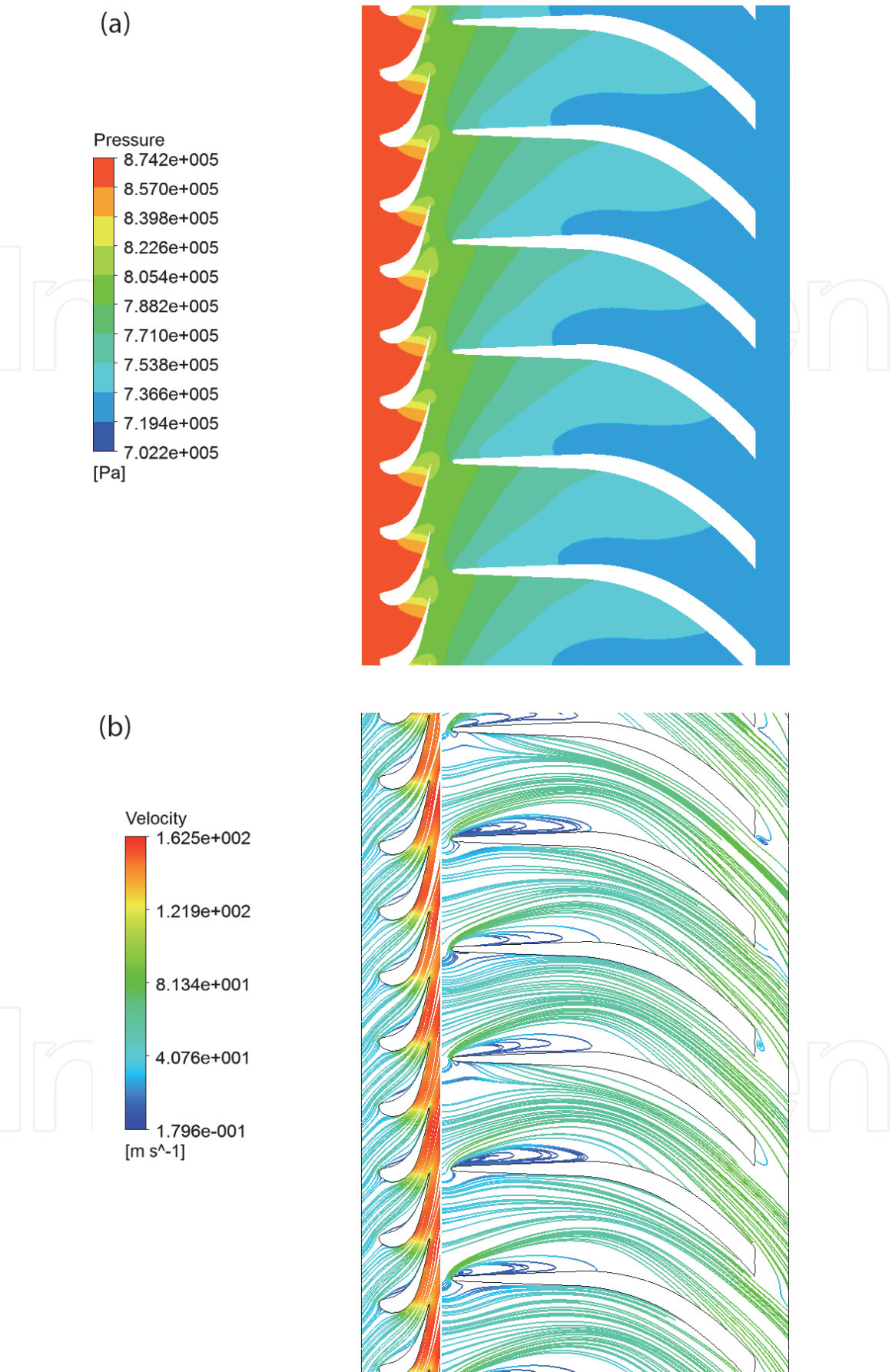


Figure 12.
*Pressure contour and velocity streamline diagram of nozzle and impeller passage section at 50% spanwise.
(a) Pressure contour, and (b) Velocity streamline diagram.*

meeting the design requirement. The pressure contour shows a small vortex area of low pressure at the suction side of blade trailing edge, which is also seen in the velocity streamline diagram.

Therefore, through numerical simulation, the actual performance of turbine can be estimated, and the part that needs to be optimized can be figured out to make the aerodynamic efficiency of turbine better. In addition, numerical simulation can also be used to simulate and validate the performance of OTEC turbine in other aspects. For example, in [17], the performance of OTEC turbine under design and off-design conditions is predicted in detail. And the authors in [18] discussed the effects of turbine blade number, mounting angle, and blade fillet on OTEC turbine performance.

2.3.3 Aerodynamic optimization research of turbine in OTEC

The aerodynamic performance of turbine in OTEC is significantly affected by the aerodynamic performance of nozzle and impeller. Therefore, aerodynamic optimization of flow geometry such as blade profile is crucial to improve flow characteristics and aerodynamic performance of turbine.

The aerodynamic optimization process of turbine is a reciprocating process of design modification, evaluation, and improvement. It mainly includes selecting design variables, determining and solving optimization objectives, and establishing the relationship between optimization objectives and design variables, as shown in **Figure 13**.

Before aerodynamic optimization of blade cascade, it is necessary to use some curve expression technology to parameterize the geometry of optimized object and determine the design variables, such as NURBS curve, Bézier curve, and nonrational B-spline methods. Then, the numerical software is used for geometric modeling, meshing, numerical solution of flow field, and flow field treatment and analysis after solving, so as to analyze and evaluate the aerodynamic performance of the optimized turbine and obtain the objective function value. Finally, the optimal design algorithm is used to search for the optimal design parameters, and then optimization parameters are used to improve the design variable parameters, so that the optimal solution can be quickly searched and the optimization process automation can be realized.

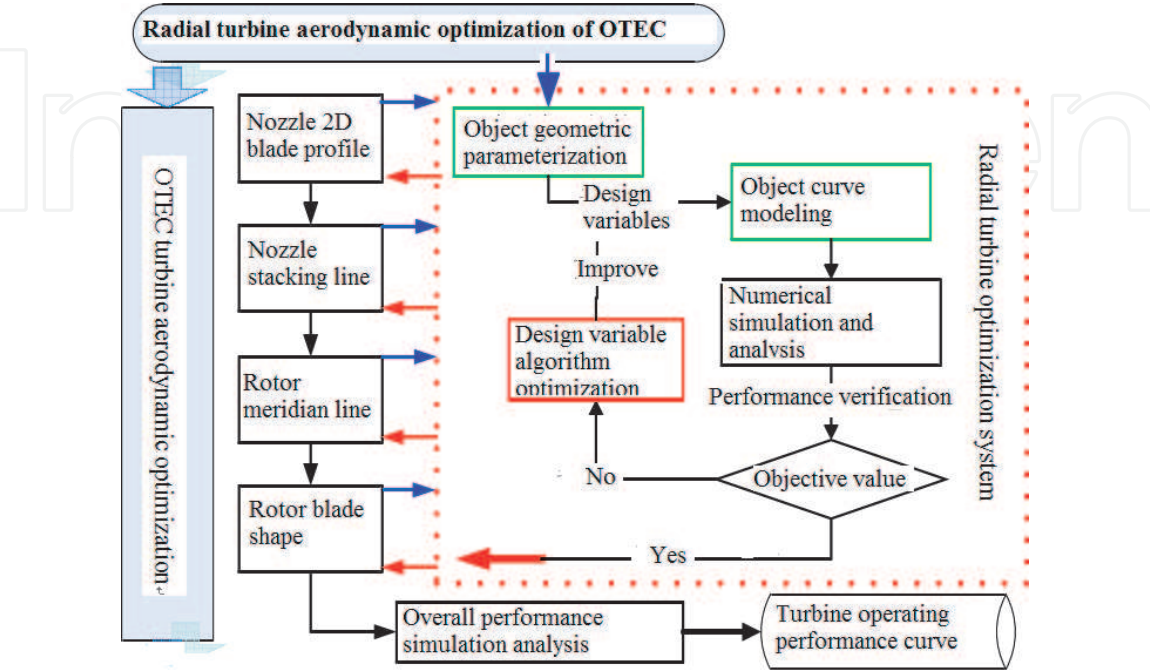


Figure 13.
Optimization process of radial turbine.

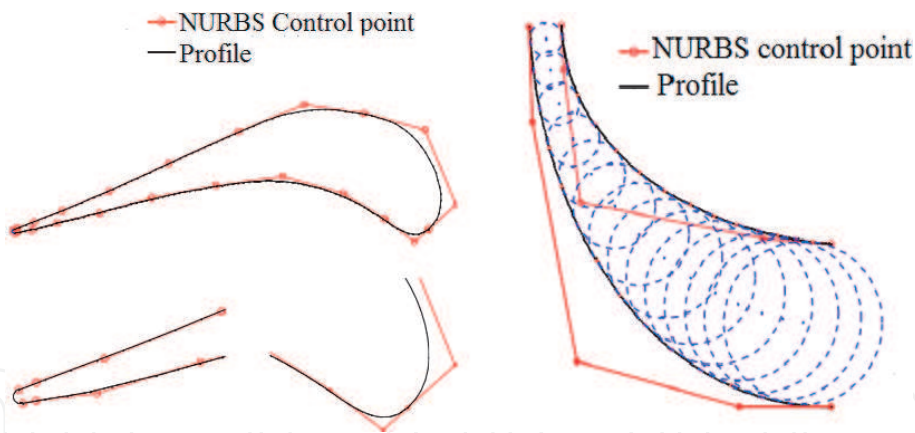


Figure 14.
Profile of nozzle blade and impeller meridian blade.

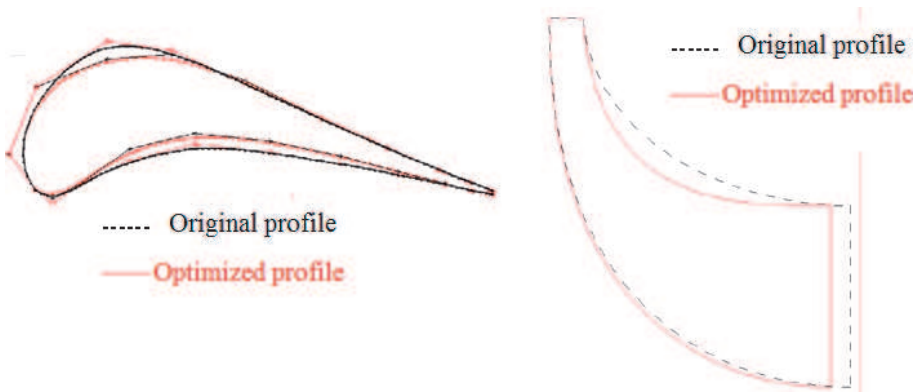


Figure 15.
Original and optimization profile of nozzle and impeller blade.

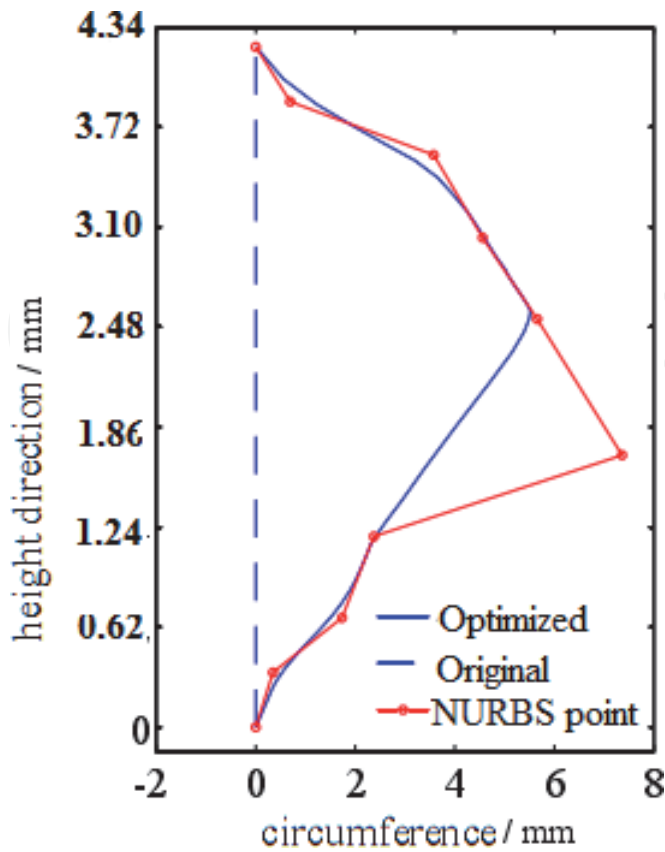


Figure 16.
Optimization of nozzle stacking line.

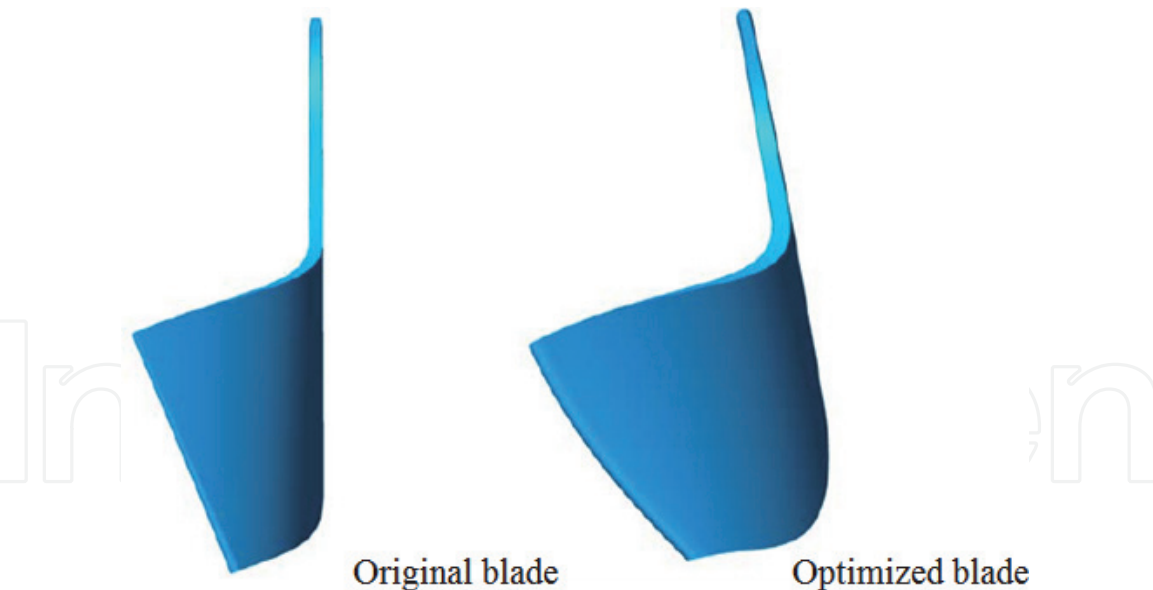


Figure 17.
Original and optimized blade of impeller.

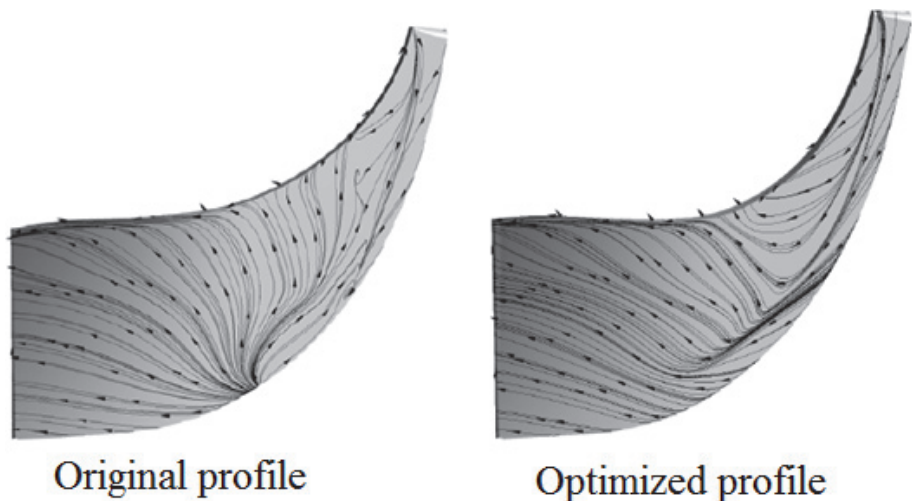


Figure 18.
Flow line at the suction surface of impeller.

In this case, NURBS curve is used to control nozzle blade profile and meridian profile of impeller, as shown in **Figure 14**. By integrating the above optimization method with ANSYS CFX program, geometric modeling, meshing, numerical solution, and performance analysis of flow field are carried out. And then the optimization algorithm is used to improve the designed variable parameters to search for the optimal solution quickly. Finally nozzle blade profile and impeller blade profile are obtained, as shown in **Figure 15**.

In addition to using control points to optimize nozzle and impeller, in order to ensure the design conditions and meet the performance requirements, the aerodynamic performance of cascade stacking line is optimized by “sweeping” method considering that nozzle is a radial cascade. The optimized stacking line is shown in **Figure 16**. For impeller blades, the backward-curved blades shown in **Figure 17** are used to improve the flow characteristics of turbine impellers further.

Finally, a comparative analysis of optimized nozzle and impeller performance is performed. **Figure 18** shows the streamline at the suction surface of impeller. It can be seen that optimized meridian channel curve changes uniformly, and the flow direction is controlled to change more gently, thus eliminating partial separation of

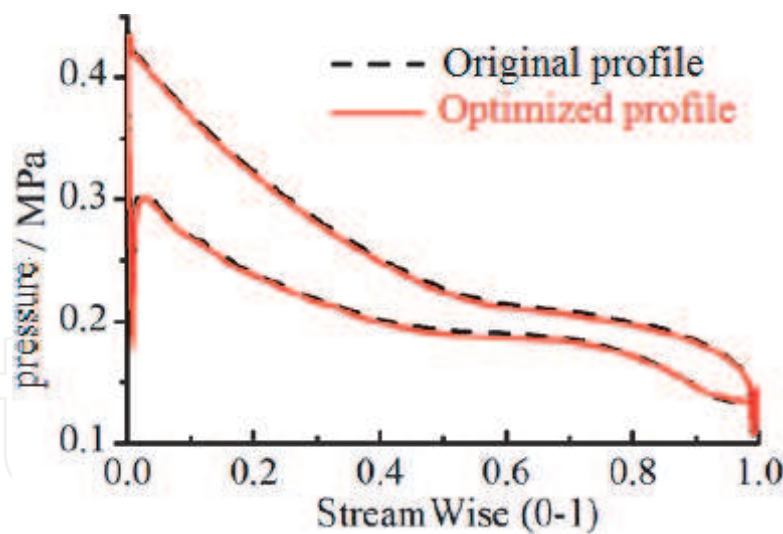


Figure 19.
Static pressure curve at 50% spanwise of impeller surface.

suction surface in impeller. The static pressure curve at 50% spanwise of impeller surface shown in **Figure 19** indicates that the static pressure is in the changes of favorable pressure gradients. The improvement of the static pressure change of blade suction surface can eliminate the local flow separation at suction surface of the original impeller. **Figure 20** shows the optimized streamline of nozzle and impeller at 50% spanwise. Comparing with **Figure 12**, it can be seen that the optimized flow field is more uniform without local vortex. Finally, the operation performance of turbine is numerically calculated using the optimized nozzle profile and impeller profile. The results show that operation efficiency of turbine is 1.05% higher than the original under design conditions.

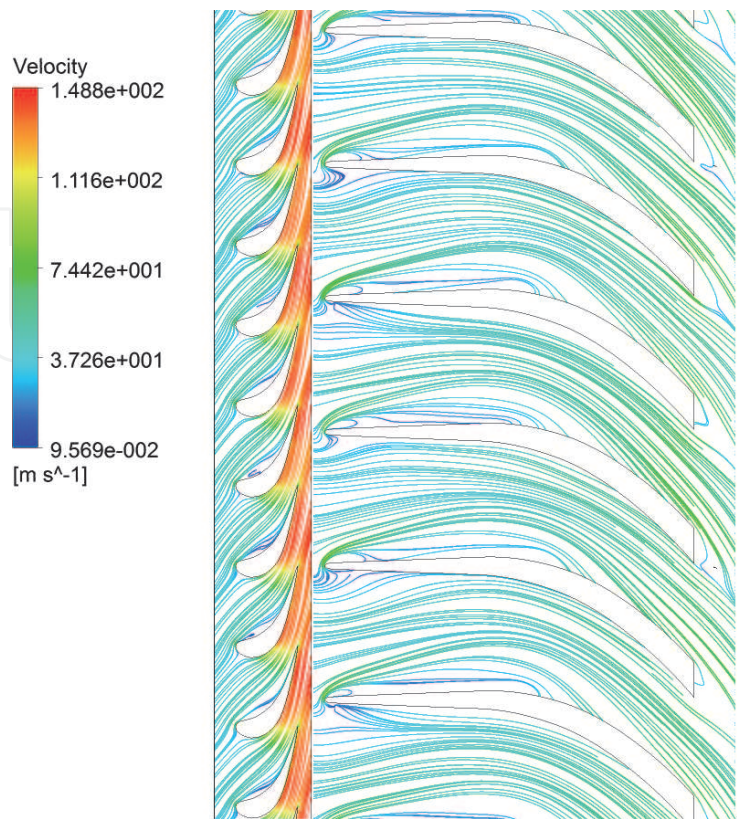


Figure 20.
Optimized streamline diagram of nozzle and impeller at 50% spanwise.

3. Mechanical and electric control design of turbine

This part contains the brief design of mechanical system and electric control system of turbine. In terms of mechanical design, in addition to the vibration and strength of impeller, the sealing design to prevent leakage of working fluid is also an important part of the research. Finally, in order to ensure the stability and safe operation, control part of turbine will also be introduced as the design content.

3.1 Mechanical design of turbine

3.1.1 Mechanical design of impeller

After the aerodynamic design of turbine is finished, mechanical research on material, strength, and vibration needs to be carried out in order to achieve the efficient and stable operation of turbine. Since working fluid is corrosive and turbine has high rotating speed, it is necessary to control the manufacturing process of blades. Therefore, requirements for impeller blade material of turbine include mechanical fatigue resistance, sufficient plasticity and toughness, better chemical stability with higher corrosion resistance, better vibration damping, superstructural manufacturability, and so on. Aluminum alloys, stainless steels, and titanium alloys are usually used as impeller materials according to the requirements of environment and economy.

In addition, the assembly and balance of rotating components are closely related to the safe operation of turbine. Therefore, in order to make turbine safe and stable, impeller must be assembled to have a specified axial clearance and radial clearance with respect to components such as nozzle ring and turbine cylinder. Before impeller is assembled, strict static balance must be carried out to check whether mass distribution is uniform, and unbalanced mass should be eliminated or reduced to permissible range. After assembly, dynamic balance of the whole rotating parts including impeller and shaft should be carried out to ensure the balance accuracy of dynamic parts.

Finally, the strength check of impeller is an indispensable part of turbine design. If the designed impeller does not meet the strength requirements, the aerodynamic scheme must be redistributed until the strength is satisfied. In this part, ANSYS is used to analyze the strength of impeller and rotor in terms of stress, strain, and vibration mode and check whether safety factors meet the requirements. The simulation of impeller strain is shown in **Figure 21**.

3.1.2 Mechanical structure design of turbine

In addition to pneumatic system, the turbine includes oil system, shafting system, sealing system, and power generation control system.

Oil system generally includes oil circuit for dynamic seal lubrication, oil circuit for bearing oil film, and oil circuit for generator heat dissipation. Generally, the first two parts have certain pressure, which not only provide lubrication and heat dissipation for sealing and bearing but also ensure that pressure is higher than pressure on working fluid side to have a certain sealing effect. The latter oil circuit is an ordinary oil system. The shafting system mainly includes the turbine main shaft, bearing, coupling, reducer, etc., which plays the role of transferring the work of impeller rotation to motor for power generation. The sealing system includes dynamic and static sealing parts. The dynamic sealing component is mainly used to

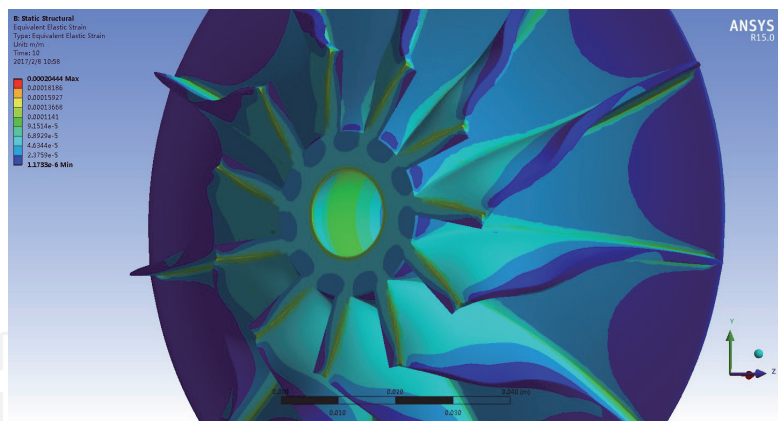


Figure 21.
Strain contour of turbine impeller.

seal flow channel system from external system at the shaft during the high-speed rotation of shafting. They have various structural forms, which will be introduced in detail in the next part. The static sealing component is mainly an O-ring that contacts the stationary components to isolate the working fluid, lubricating oil, and outside air. If high-speed generators are used for power generation, it can be driven by turbo-side shafting to generate electricity at a high speed and then converted into 50/60 Hz conventional power by rectification inversion. While the control part mainly controls turbine steam volume, speed, and generator operation, it also includes digital control system with emergency interruption, which will be described in the next section.

Figure 22 shows the schematic diagram of turbine. Mechanical seal is used for turbine in this case, and a high-speed generator is used for power generation. Therefore, the shafting system mainly includes the main shaft, high-speed bearing, and high-speed coupling. The main shaft is an important component connecting impeller and high-speed motor, which transfers the work of impeller to a high-speed coupling. There are two high-speed bearings, which play a supporting role in the main shaft rotating process and ensure that the shaft does not move in the circumferential and axial direction when rotating at a high speed. The high-speed coupling connects the main shaft and high-speed generator to ensure that the work of the main shaft can be smoothly transmitted to a high-speed generator. Oil circuit

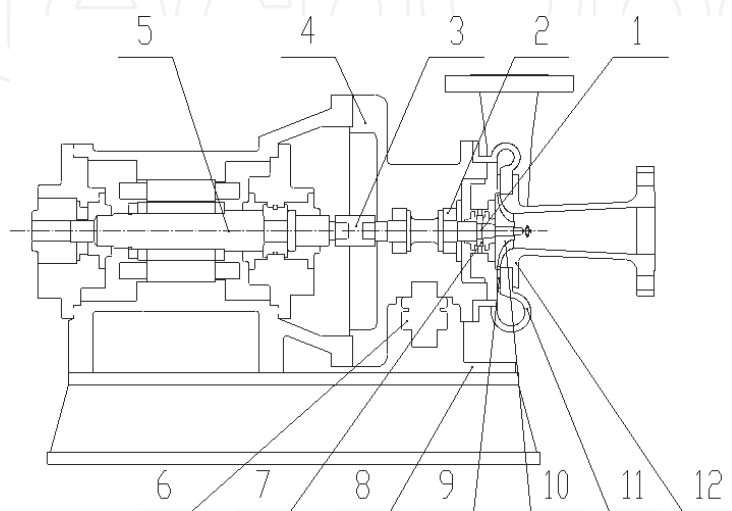


Figure 22.
Schematic diagram of turbine (1) spindle, (2) bearing, (3) coupling, (4) body, (5) high-speed generator, (6) oil pipe, (7) sealing, (8) plate, (9) nozzle, (10) impeller, (11) volute, (12) diffuser.

system includes three parts, oil circuit of sealing device, oil circuit of high-speed bearing, and oil circuit of high-speed generator, represented by oil supply pipeline. Lubricating oil passes through the above three parts and enters the tank for cooling and recycling.

The flow passage system is the passage for ammonia designed in the previous chapter, including volute, nozzle, impeller, and diffuser. The saturated ammonia vapor heated by warm seawater first flows into volute, so that the airflow is evenly distributed in circumferential direction. The airflow expands and accelerates after entering nozzle with a large kinetic energy and then enters impeller at a high speed to drive the main shaft. Finally, ammonia is discharged through diffuser. In addition, the body and plate mainly have the function of wrapping and supporting the turbine. Mechanical seal belongs to dynamic seal, and static seal is distributed in the connection of corresponding static components.

3.1.3 The sealing of turbine

Since ammonia or organic working fluid is used as working fluid in OTEC, leakage should be avoided as much as possible. Turbine is the device that drives high-speed motor or speed reducer to rotate at a high speed. Therefore, besides ensuring static sealing inside turbine, it is also necessary to ensure that shaft-end dynamic seal can prevent the leakage of working fluid as well. Sealing has become a key issue in turbine design, as well as the design of shaft sealing. The shaft-sealing device of turbine is a kind of dynamic sealing device installed on the main shaft to minimize the leakage of working substance inside the turbine to outside. There are generally two shaft-sealing devices inside the turbine. One is the shaft seal installed at the high-pressure end of turbine, which is mainly to prevent working fluid from entering the main shaft to corrode lubricating oil or bearing to cause bearing wear. The other is installed at the connection between the main shaft and motor, which is mainly to reduce the leakage of the working fluid to outside world and then to reduce the pollution to environment and harm to human.

The turbine shaft sealing mainly includes the following types: labyrinth seal, floating ring seal, carbon ring seal, mechanical seal, and dry gas seal. The principle of labyrinth seal is that when gas passes through the cogging gap, it obtains a certain speed. In the larger space between teeth, its kinetic energy is transformed into heat energy, while pressure decreases, and process of pressure reduction is repeated between a cogging gap. Although the labyrinth seal technology is very mature, it has the disadvantage of large leakage compared with other seals. Therefore, the labyrinth seal can be used as auxiliary seal, instead of directly sealing working fluid only with it.

Floating ring seal is mainly composed of an inner floating ring and an outer floating ring. The pressure between inner and outer floating rings is slightly higher than that of sealed working fluid. The inner and outer floating rings restrict internal and external leakage through small clearance throttling. The floating ring seal has the following advantages, long service life, high reliability, suitable for high-speed, wide operating range, etc., but it also has the disadvantages of large leakage and complex control system.

Carbon ring seal has structure which combines labyrinth seal and floating ring seal. According to the internal structure of carbon ring, it can be divided into flow resistance contact and non-contact sealing structures. The former has no gap between carbon ring and shaft and compensates itself with the wear of the carbon ring, while the latter has a working gap between carbon ring and the shaft. The selection of carbon ring seal structure is mainly based on the working conditions. It is widely used in steam turbines and compressors because of its convenient

installation, good sealing, easy maintenance and repair, simple system, and no need of lubrication and cooling.

Mechanical seal is a leakage-preventing device which is formed by a pair of end faces perpendicular to the rotation axis, which keep sticking and merging relative sliding under the action of elastic force of compensation mechanism and the cooperation of auxiliary seal. The main component of a mechanical seal is a seal ring consisting of a moving ring and a static ring. The mechanical seal mainly has the advantages of reliable sealing, stable sealing state in long-term operation, low power consumption, good vibration resistance, and wide application range. However, since the mechanical seal is internally matched to improve the sealing performance, the main disadvantages are that the structure is complicated, it includes high processing requirements, installation and replacement are troublesome, and the cost is high.

The structure of dry gas seal is similar to that of mechanical seal. It is also composed of static ring, moving ring, spring, auxiliary seal ring, and transmission mechanism of moving ring, and there is a hydrodynamic groove on the outer side of moving ring. When the shaft of the main engine rotates, the ring rotates, and then the gas in the sealing chamber enters the sealing end face along the hydrodynamic groove. Dry gas seals can be divided into single-end seals, tandem seals, tandem seals with intermediate labyrinth seals, and double-face seals. The dry gas seal has the characteristics of small leakage, low power consumption, and long service life and is suitable for sealing of toxic and harmful working fluids.

Considering the main features of various seals and high sealing requirement due to high speed of turbine, dry gas seal and mechanical seal are suitable for the turbine if improved well. For example, the turbine in this case adopts the improved double-end mechanical seal structure, which ensures zero leakage and stable operation of turbine. In addition to dynamic seal at the shaft end, for static seal, such as the seal at inlet and outlet, the corrosion-resistant O-ring is chosen to seal on the joint surface.

3.2 Control system design of turbine

The control system of turbine in OTEC power generation system mainly includes a speed regulating system and automatic shutoff system, as shown in **Figure 23**. Speed regulating system changes the inlet flow rate by controlling the opening of the control valve, so as to make turbine work as required. Auto shutoff system will shut down the intake valve quickly in emergency to protect the safety of unit. Digital electro-hydraulic (DEH) system is referred to as speed regulation system, and DEH system realizes the functions of program-controlled start-up, automatic adjustment, parameter limitation, protection, and monitoring of turbine by control loops working separately or jointly.

The automatic adjustment control function is mainly to set the turbine target speed; the unit can automatically control the regulating valve along empirical curve corresponding to current state and complete speed-up control until target speed. In the process of speedup, the operator can also control the speed-up process by modifying the target speed, rate of increase, and speed-keeping time.

When the unit dumps load, DEH accepts oil switch tripping and overspeed signal and quickly closes the regulating valve to reduce the conversion overshoot. The valve will open automatically after a delay or recovered normal speed and maintain unit speed as target speed. This process can be achieved by limiting control function.

Emergency manual means that when DEH communication fails, the automatic control will automatically or manually turn to emergency manual state. The control valve can be adjusted and reduced according to the hard disk, and the valve opening can be controlled directly through the servo plate.

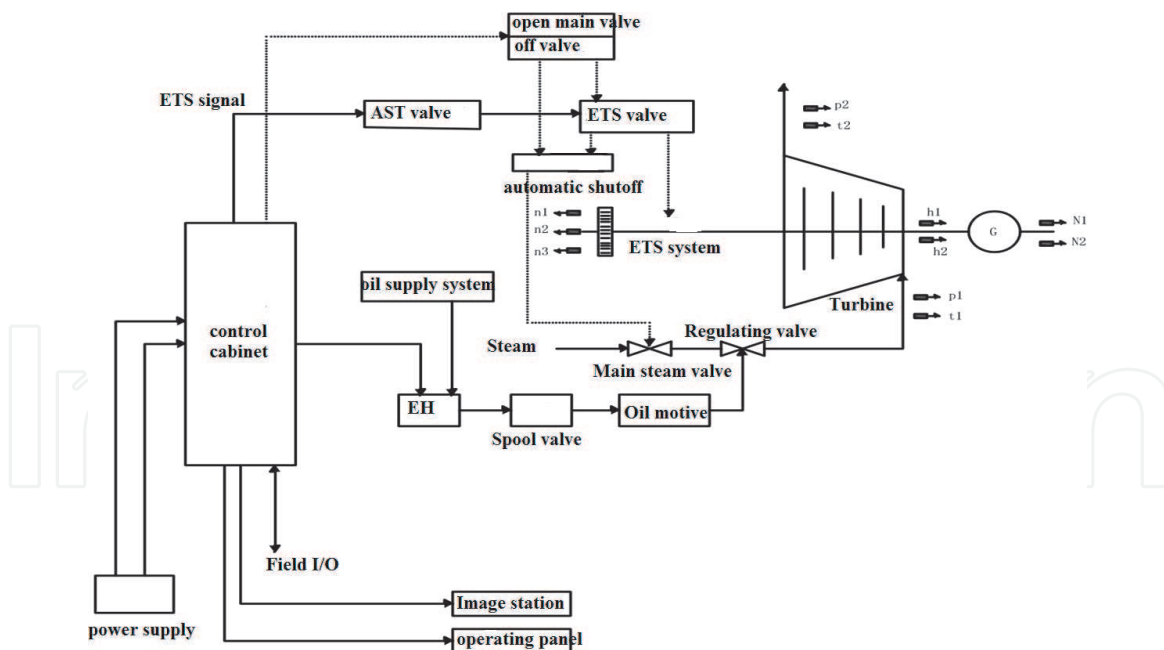


Figure 23.
Schematic diagram of turbine control system.

Overspeed protection means that when the speed exceeds the tripping speed, DEH sends out a signal to block the system action and quickly closes the main steam valve and regulates the steam valve.

In order to ensure the reliability of DEH system, system design should conform to safety design principles stipulated by the International Electrotechnical Commission (IEC). There should be preventive measures for possible operations, and turbine can be stopped safely when power source is lost. The whole control system should be dominated by electric control and the hard hand control is used as a backup. Redundancy design should be adopted in the whole system, and strict tracking measures should be taken to make switching between different modes without disturbance.

Emergency Trip System (ETS) for high-speed turbine needs to realize the following functions: continuously monitor turbine operation safety, and jump out in time to ensure safety in case of emergencies; the anti-interference performance complies with relevant standards to avoid interference of external radio signals leading to wrong operation; alarm signal uses sound-light alarm; and independent control processing module and measurement components are used to ensure reliability. When the following happens, turbine overspeed to tripping speed, thrust pad temperature is high, lubricant pressure is low, axial displacement is large, bearing vibration is large or manually shut down, etc., ETS will issue a jump instruction. In order to ensure the normal operation of bearings, it is necessary to monitor the temperature of bearings and lubricants as well as oil pressure of lubricants. In control system, turbines must be equipped with auxiliary system monitoring and protection functions as well as low oil pressure protection devices.

In addition, in order to ensure that the turbine control system has good dynamic response characteristics and sufficient valve lift force, it is necessary to have electro-hydraulic (EH) oil system to provide control and power oil for the turbine servo system and protection system. Lubricating system needs to provide lubricating oil and sealing oil for turbine bearings and high-speed generators and pressure oil for mechanical overspeed emergency interruption system.

4. Problems and prospect of turbine in OTEC

This section mainly introduces the problems in the turbine currently and the prospects for the development of the turbine in OTEC.

4.1 Technical issues of turbine in OTEC

There are technical issues to be resolved in the turbine systems currently running for low efficiency, leakage, operation stability, etc. So far, the authors have found the following technical difficulties of turbine in OTEC through research for several years:

1. At present, the design of turbine is not initiated for OTEC processes, but basically follows conventional radial turbine design methods. The selection of empirical parameters, calculation of geometric parameter, and 3-D modeling are all referenced to the turbines used in waste heat power generation and low-temperature power generation. There are barely any specific aerodynamic designs and performance studies for the small temperature difference of turbine, specifically for OTEC. Therefore, the aerodynamic efficiency varies with research methods applied, and therefore it may not be promising to reach the highest efficiency in OTEC.
2. Ammonia or organic working fluids used in the system have potential impacts on the environment and humans, especially because ammonia is toxic and irritating and may explode under unexpected thermodynamic conditions. Unlike other parts of the OTEC system such as heat exchangers and pumps, the turbine rotates at a high speed, so that the turbine sealing should be perfectly guaranteed for long-term operations of OTEC. The seals introduced above often have low performances and sophisticated internal structure so that it is hard to replenish while the OTEC system is on and only remote control is available. Therefore, the zero leakage of working fluid in turbine is an everlasting key issue to be addressed.
3. The power generated by OTEC system is relatively low, as low as 100 kW. In other words, the high-power systems are built at small-scale systems during R&D phases, having the high cost and slow recovery of benefits. As high-power generation systems are rare, the high-power turbine for OTEC is also one of the future development tasks.

4.2 Prospect of turbine in OTEC

Looking for solutions in view of the current problems of power generation turbines, this section looks to the future to find development direction of the turbine in OTEC. For the problems existing in turbines for OTEC, the authors have opinions that the turbines can be improved by addressing the following aspects of turbine design and manufacturing in the future:

1. The magnetic bearing and drive technology currently used in other rotating machinery can be adapted in the turbine of OTEC. Although magnetic bearing has the characteristics of little friction loss and high-speed performance, its direct application to OTEC turbine requires maturity of its own technology at a low cost. Magnetic drive technology is currently mainly used in high-power,

low-speed rotating machinery, which can completely replace the ordinary coupling to ensure zero leakage of working fluid. But due to its low transmission efficiency, it is necessary to ensure that it's suitable in high-speed turbines without hysteresis and great loss in the future.

2. In order to improve the aerodynamic performance of turbines in OTEC, it is necessary to seek a special optimization strategy for the blade profile of the turbines to optimize the aerodynamic part of the turbine. So as to ensure the maximum efficiency of the OTEC system, it's necessary to improve the aerodynamic efficiency of the turbine to the highest efficiency in OTEC.
3. In order to achieve the operation of the high-power OTEC system as soon as possible, it is necessary to have research on the performance of high-power turbine in which the main directions are low speed, turbine type, and so on.

5. Conclusions

This chapter mainly introduces the turbine in OTEC, including the design method of turbine such as aerodynamic, mechanical, and control design. Finally, the current problems and prospects of the power generation turbine in OTEC are discussed. It should be noted that the design method of turbine is not limited to the method introduced in the chapter. The authors hope that the readers can have a comprehensive and brief understanding of the turbine in OTEC through the introduction of this chapter. On this basis, the readers can also be expected to conduct a more in-depth study and discussion of the turbine in OTEC.

Acknowledgements

This work was supported by the China-Korea Cooperation Project “China-Korea Technology Exchange and Cooperation of Ocean Energy Development and Utilization” (PI-2018-4).

Nomenclature

Symbols

\overline{D}_2	wheel diameter ratio
\overline{u}_1	velocity ratio
c	absolute velocity
u	circumferential velocity

Greek symbols

η	efficiency
Ω	degree of reaction
ω	relative velocity
α	absolute flow angle
β	relative flow angle
φ	velocity coefficient of impeller
ψ	velocity coefficient of nozzle

Subscripts

$j = 1, 2$	inlet and outlet of impeller
u	circumferential direction
c	cold seawater
i	inlet
o	outlet

Author details


Weimin Liu^{1*}, Yunzheng Ge¹, Lei Liu¹ and Yun Chen²

1 First Institute of Oceanography, Ministry of Natural Resources (MNR), Qingdao, China

2 Shandong University, Qingdao, China

*Address all correspondence to: lwmxjtu@163.com

IntechOpen

© 2020 The Author(s). Licensee IntechOpen. This chapter is distributed under the terms Commons Attribution - NonCommercial 4.0 License (<https://creativecommons.org/licenses/by-nc/4.0/>), which permits use, distribution and reproduction for non-commercial purposes, provided the original is properly cited. 

References

- [1] Schobeiri T. Thermo-fluid dynamic design study of single and double-inflow radial and single-stage axial steam turbines for open-cycle thermal energy conversion net power-producing experiment facility in Hawaii. *Journal of Energy Resources Technology*. 1990; **112**(1):41-50. DOI: 10.1115/1.2905711
- [2] Yang MH, Yeh RH. Analysis of optimization in an OTEC plant using organic Rankine cycle. *Renewable Energy*. 2014; **68**(3):25-34. DOI: 10.1016/j.renene.2014.01.029
- [3] Yoon JI, Son CH, Baek SM, et al. Efficiency comparison of subcritical OTEC power cycle using various working fluids. *Heat and Mass Transfer*. 2014; **50**(7):985-996. DOI: 10.1007/s00231-014-1310-8
- [4] Uehara H, Ikegami Y. Optimization of a closed-cycle OTEC system. *Trans of the ASME Journal of Solar Energy Engineering*. 1990; **112**(4):247-256. DOI: 10.1115/1.2929931
- [5] Wu C. Specific power optimization of closed-cycle OTEC plants. *Ocean Engineering*. 1990; **17**(3):307-314. DOI: 10.1016/0029-8018(90)90007-s
- [6] Han S, Seo JB. 20kW turbine development for OTEC system. *The KSFM Journal of Fluid Machinery*. 2014; **17**(6):38-43. DOI: 10.5293/kfma.2014.17.6.038
- [7] Chen F, Liu L, Peng J, et al. Theoretical and experimental research on the thermal performance of ocean thermal energy conversion system using the Rankine cycle mode. *Energy*. 2019; **183**:497-503. DOI: 10.1016/j.energy.2019.04.008
- [8] Rama SRG, Aijaz AK. *Turbomachinery: Design and Theory*. Boca Raton: CRC Press; 2003
- [9] Whitfield A, Baines NC. *Design of Radial Turbomachines*. Harlow: Longman Scientific Technical; 1990
- [10] Balje OE. *Turbomachines: A Guide to Design, Selection and Theory*. New Jersey: Wiley; 1981. DOI: 10.2307/27851419
- [11] Whitfield A. The preliminary design of radial inflow turbines. *Journal of Turbomachinery*. 1990; **112**(1):50. DOI: 10.1115/1.2927420
- [12] Costall AW, Gonzalez HA, Newton PJ, et al. Design methodology for radial turbo expanders in mobile organic Rankine cycle applications. *Applied Energy*. 2015;S0306261915002573. DOI: 10.1016/j.apenergy.2015.02.072
- [13] Rahbar K, Mahmoud S, Al-Dadah RK, et al. Modelling and optimization of organic Rankine cycle based on a small-scale radial inflow turbine. *Energy Conversion and Management*. 2015; **91**:186-198. DOI: 10.1016/j.enconman.2014.12.003
- [14] Rahbar K, Mahmoud S, Al-Dadah RK, et al. Parametric analysis and optimization of a small-scale radial turbine for organic Rankine cycle. *Energy*. 2015; **83**:696-711. DOI: 10.1016/j.energy.2015.02.079
- [15] Rahbar K, Mahmoud S, Al-Dadah RK. Mean-line modeling and CFD analysis of a miniature radial turbine for distributed power generation systems. *International Journal of Low-Carbon Technologies*. 2014;ctu028. DOI: 10.1093/ijlct/ctu028
- [16] Li Y, Ren XD. Investigation of the organic Rankine cycle (ORC) system and the radial-inflow turbine design. *Applied Thermal Engineering*. 2016; **96**:547-554. DOI: 10.1016/j.applthermaleng.2015.12.009

[17] Nithesh KG, Chatterjee D.
Numerical prediction of the
performance of radial inflow turbine
designed for ocean thermal energy
conversion system. *Applied Energy*.
2016;**167**:1-16. DOI: 10.1016/j.
apenergy.2016.01.033

[18] Nithesh KG, Chatterjee D, Oh C,
et al. Design and performance analysis
of radial-inflow turboexpander for
OTEC application. *Renewable Energy*.
2016;**85**:834-843. DOI: 10.1016/j.
renene.2015.07.018