INTRODUCTION

Renewable energy has gained more and more attention because of the contradiction between less fossil energy available and increased demand for it becoming more prominent. Geothermal energy is considered a renewable energy source and has the advantage of not being affected by the diurnal temperature difference and weather change. The medium–low-temperature heat source belongs to low-grade heat energy, and the energy quality is low. It is challenging to promote Steam Rankine Cycle System power generation. Organic Rankine cycle system can choose different working fluid matching according to different grade of heat source, that is, using low boiling point working fluid, it can absorb heat.

Analysis of impeller blade parameters and tip clearance of turboexpander in organic Rankine cycle system

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Abstract

The optimization of a turboexpander can significantly improve the performance of the organic Rankine cycle (ORC) system, which can use in low-temperature geothermal energy for high-efficiency power generation. Therefore, this paper studies and optimizes the expander in the ORC of low-temperature geothermal energy around 90°C. Firstly, the influence of impeller parameters on the performance of the expander is analyzed. Then, the grid is divided by the turbine mesh, and numerical simulation is performed by CFX. The gas state equation selects the Peng–Robinson equation, and the turbulence equation selects the SST model. Finally, the effect of tip clearance on the performance of the expander was studied. Research on impeller blade parameters shows that with the increase in the outer diameter of the impeller blade outlet, the isentropic efficiency of the expander decreases, and the output power increases. As the increase in the angle between the direction of the impeller blade and the meridian plane, the outlet velocity of the blade increases, the temperature decreases, and the efficiency and control of the turboexpander will decrease. The meridional section width has few effects on the performance of the expander. The study of the tip clearance by numerical simulation shows that the existence of tip clearance will cause clearance flow between pressure surface and suction surface that interferes with the mainstream direction. When the tip clearance increases from 0 mm to 1.2 mm, the efficiency of the expander is reduced by 8.9%.

KEYWORDS

CFX, impeller blade parameters, organic Rankine Cycle system, tip clearance, turboexpander

1 INTRODUCTION

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and evaporate to promote expander work at lower temperature, so it is regarded as an active power generation method in low- and medium-temperature heat source situation.\(^2\) The turboexpander has the characteristics of high rotation speed, high single-stage expansion ratio, and wide operating range. It is often used as a thermal power conversion device for a low-power ORC system.

Due to the lower operating temperature, the efficiency of the ORC is about 10%-20%, and the performance of the ORC system is closely related to the expander. However, the influence of impeller and tip clearance on the efficiency of the expander is significant. In this paper, a series of studies on the impeller and tip clearance of the turboexpander is carried out. At present, scholars have studied this aspect, and some of them have studied the material, design, and numerical simulation of the impeller. Hernandez-Carrillo et al.\(^3\) proposed a material replacement for the impeller of a tilted radial microturbine. Compared with aluminum, he used composite materials (PEEK-GF30) and thermoplastics (ABS). The results show that in the worst case (over speed), PEEK-GF30 is 11% stronger than aluminum in structure and ABS is 40% weaker than aluminum in construction, but both materials are strong enough. To optimize the performance and structural strength of the expander, Harinck\(^4\) designed the back-bend impeller with multiple blade angles. Song et al.\(^5\) optimized the geometry of the impeller, including the objective function, the meridian contour of the impeller, and the blade shape and effectively suppressed the cavitation. Feng et al.\(^6\) applied the traditional cylindrical parabolic geometry design method to the impeller design. And the final designed impeller meets the strength requirements and exhibits excellent aerodynamic performance. Fu et al.\(^7\) used the wheel diameter (D) and rotational speed (N) values as design parameters to directly help the turbine design. Man Wang et al.\(^8\) show that under saturated steam conditions, the optimization of impeller and blade parameters can obtain better system performance. Wang et al.\(^9\) used fluent to simulate the impeller of a turboexpander. By analyzing the flow field, the complex flow phenomena were obtained, including separation flow, secondary flow, and wake flow. Fredmonsk et al.\(^10\) used the three-dimensional Euler numerical simulation method to calculate the flow field and the aerodynamic optimization design of the centrifugal turbine blade and the flow channel. Sam et al.\(^11,12\) have studied the flow field of turbine impeller. It is shown that the triangular shape of the turbine blade, such as blade inlet angle, the blade shape, the tip clearance height, and the trailing edge thickness. To study the limitations of ORC performance, Robertson et al.\(^13\) applied the genetic algorithm to the turboexpander of the ORC system, aiming to optimize the geometry specification (radii, areas, blade heights, and angles) to provide maximal time-averaged power output. Khairuddin et al.\(^15\) used genetic algorithms to adjust the main features of the blade geometry—he hub, shroud, camber line, leading, and trailing edge profiles—to significantly improve the efficiency of the turboexpander. According to some scholars’ research on impeller blades, this paper proposes to study and analyze the parameters of outlet diameter, blade angle, and meridian width of impeller blades, to improve the flow expansion acceleration in the impeller and the distribution of flow field parameters at impeller outlet.

The influence of the tip clearance on the turboexpander is also particularly significant, and the tip clearance loss is mainly concentrated on the suction surface of the induction wheel to the mid-string section.\(^16\) Tiainen et al.\(^17\) studied the development of Reynolds number loss in a centrifugal compressor without a vane diffuser and distinguished the losses caused by different causes. Soo-Yong Cho et al.\(^18\) experimentally studied the loss of incidence in different parts. Also, the effect of tip clearance was studied since it is another critical parameter affecting turboexpander performance. Nithesh et al.\(^19\) studied the effects of different geometric parameters on turbine performance, with the impact of tip clearance being significant. Persky et al.\(^20\) systematically analyzed more than 1.5 million loss model configurations established by enumerating analytical models for each loss mechanism in the literature. In this paper, the study of the tip clearance is the comparison of the prediction results of the empirical loss model with the numerical simulation results of this paper to understand the impact of the existing loss model on the ORC turbine.

In this paper, the impeller parameters are analyzed, and the influence of tip clearance on the performance of turboexpander and the comparison between numerical simulation and empirical loss model of tip clearance are studied. Because scholars have done a lot of research on the design, numerical simulation, and experiment of turboexpander, this paper introduces this aspect in detail and chooses the conventional design method, organic refrigerant, and numerical simulation.

In this paper, the low-temperature geothermal energy of 90°C was studied and select R245fa as the organic working fluid. The dimension and aerodynamic parameters of the volute, guide vane, and impeller blade are obtained by the one-dimensional thermal design of 200 kW turboexpander, and the three-dimensional modeling and meshing of the volute, guide vane, and impeller blade are carried out. The flow field of tip clearance in turboexpander under design conditions was analyzed by CFX three-dimensional numerical simulation method.
2 ANALYSIS OF TURBINE EXPANDER IN GEOTHERMAL ORC SYSTEM

2.1 Brief introduction of ORC system of geothermal energy

As shown in Figure 1, the ORC geothermal energy power generation system is generally composed of five parts: evaporator, turboexpander, condenser, working the medium pump, and generator. The geothermal water is pumped to the surface by the high-pressure pump and enters the evaporator to exchange heat with the lower boiling organic working fluid. The organic working liquid is evaporated, and the high-temperature and high-pressure steam come to the turboexpander to push the impeller to rotate, and the generator is used to generate electricity. The low-temperature and low-pressure vapor from the expander enters the condenser and is further cooled to become a liquid, and then pressurized by the pump and sent back to the evaporator to exchange heat with the geothermal water to enter the next cycle.

2.2 Determination of design parameters of turboexpander

The geothermal energy ORC system thermal process analysis is shown in Figure 2. The ideal ORC cycle is 1-2s-3-4s-1, including isobaric endothermic process (4-1), adiabatic expansion process (1-2s), isobaric exothermic process (2-3), and adiabatic compression process (3-4s). The actual cycle is 1-2-3-4-1. Processes 5-6 are the heat exchange process between geothermal water and organic working substance in the evaporator to release heat, and processes 7-8 are the heat exchange process between cooling water and organic working material in the condenser to absorb heat.

In this paper, the impeller parameters and the tip clearance are mainly studied. Therefore, the conventional one-dimensional thermodynamic calculation and organic working fluid are used in this paper. The inlet fluid of the evaporator in the whole system is 90°C water, the rated power is 200 W, the organic working medium is R245fa, the evaporation temperature of the organic refrigerant is 80°C, the condensation temperature is 35°C, the outlet temperature of the heat source side of the evaporator is 71°C, and the inlet and outlet temperature of the cooling water is 25°C and 35°C, respectively. The efficiency of the turbine is 0.8, the generator efficiency is 0.96, and the organic working pump efficiency is 0.85. The calculated system parameters are shown in Table 1. Firstly, the initial parameters of the inlet and outlet of the expander are given, and then, the variable parameters are selected to carry out the one-dimensional aerodynamic design. The aerodynamic parameters and size parameters of the expander are obtained, and the geometrical shape of the guide vane and impeller of the turbine expander is determined. Then, the three-dimensional modeling of the guide vane and impeller is carried out through the modeling software. According to the given design requirements and the parameters of inlet and outlet temperature and pressure of turboexpander obtained from the calculation of ORC system, such as Table 2, the appropriate variable parameters are selected. By calling REFPROP, the specific enthalpy and entropy of the inlet and outlet of turboexpander can be obtained. The enthalpy drop of the turboexpander is calculated, and then, the enthalpy drop distribution of the guide vane and the moving vane is obtained. The aerodynamic parameters of the triangle of the inlet and outlet velocity of the moving vane are calculated. Finally, the geometric dimensions and aerodynamic parameters of the turboexpander are obtained, as shown in Table 3.

2.3 Numerical simulation of the turboexpander

Through the one-dimensional thermal design of 200 kW turboexpander, the dimensions and aerodynamic parameters of a volute, guide vane, and the impeller blade are obtained.
and the three-dimensional model of the volute, guide vane, and the impeller blade is built, and the mesh of volute, guide vane, and the impeller blade is divided.

In this paper, the convergence criteria for the steady flow calculation of the 200 kW turboexpander guide vane and impeller flow field for geothermal energy ORC waste utilization are as follows:

1. The global residual converges to less than $10^{-5}$;
2. The whole flow rate of the turboexpander does not change, or the variation amplitude is less than 0.5%;
3. In the steady calculation results, the flow field parameters of the turboexpander no longer change, and the result value is nearly constant.

The grids of vanes and impeller blades have no correlation, as shown in Table 4. The gas state equation selects the Peng–Robinson equation, and the turbulence equation selects the SST model. Organic working fluid based on the data in the software internal REFPRP (NIST Standard Reference Database 23, V8.0) physical property database, the physical parameters of R245fa in the database are called. The discrete format of the convection term is high resolution, and the dynamic–static interface model selects the mixing-plane model, and the convergence precision is $10^{-5}$.

The three-dimensional numerical simulation is carried out by CFX. The boundary conditions of the guide vane and impeller of the turboexpander are shown in Table 5. The results show that the organic refrigerant has completed the

### TABLE 1 System parameters of geothermal ORC power system

| System parameters            | Inlet temperature/°C | Output temperature/°C | Import pressure/MPa | Outlet pressure/MPa | Actual enthalpy difference/kJ kg⁻¹ | Flow/kg s⁻¹ |
|------------------------------|-----------------------|------------------------|---------------------|---------------------|-----------------------------------|-------------|
| Evaporator hot water side    | 90                    | 71                     | 0.1                 | 0.1                 | 109.1                             | 25.7        |
| Evaporator organic working side | 34.89               | 80                     | 0.789               | 0.79                | 215.94                            | 12.99       |
| Turboexpander                | 80                    | 47.58                  | 0.789               | 0.21                | 15.4                              | 12.99       |
| Condenser working side       | 47.58                 | 35                     | 0.211               | 0.211               | 41.79                             | 61.11       |
| Condenser cooling water side | 25                    | 35                     | 0.1                 | 0.1                 | 4.661E−12                         | 12.99       |

| Working fluid pump           | 35                    | 34.89                  | 0.211               | 0.79                | 4.661E−12                         | 12.99       |

| System indicator             | System efficiency/%   | Highest pressure/MPa   | Minimum pressure/MPa | Expander pressure ratio |
|------------------------------|-----------------------|------------------------|----------------------|-------------------------|
|                              | 0.0891                | 0.789                  | 0.211                | 3.73                    |

### TABLE 2 The initial aerodynamic design parameter of the turboexpander

| Parameter                          | Numerical value |
|------------------------------------|-----------------|
| Inlet pressure (MPa)               | 0.79            |
| Inlet temperature (°C)             | 90              |
| Outlet pressure (MPa)              | 0.21            |
| Output power (kW)                  | 200             |
| Speed (rad/min)                    | 15 000          |
| Speed ratio                        | 0.69            |
| Reaction degree                    | 0.4             |
| Guide vane velocity coefficient    | 0.85            |
| Impeller blade velocity coefficient| 0.95            |
| Absolute airflow angle at guide vane outlet | 18 |
| Relative airflow angle at impeller blades outlet | 36 |
| Wheel diameter ratio               | 0.42            |
| Turbine shaft efficiency           | $\eta_t = (\eta_s - \xi) \eta_o \eta_m - 0.24(1 - \epsilon)$ |
| The working fluid                  | R245fa          |

### TABLE 3 The main results of the design parameters of the turboexpander

| Parameter name                    | Symbol | Unit  | Numerical value |
|-----------------------------------|--------|-------|-----------------|
| Guide vane inlet diameter         | $D_o$  | mm    | 146             |
| Guide vane outlet diameter        | $D_n$  | mm    | 104             |
| Guide vane outlet width           | $l_o$  | mm    | 8               |
| Absolute flow angle at guide vane inlet |        |       | 18              |
| Impeller entrance diameter        | $D_1$  | mm    | 96              |
| The outer diameter of the impeller outlet | $D'_2$ | mm     | 68              |
| Impeller outlet diameter          | $D''_2$| mm    | 24              |
| Relative airflow angle at the impeller outlet |        |       | 36              |
| Guide vane velocity coefficient   | $\varphi$|       | 0.95            |
| Impeller velocity coefficient     | $\psi$ |       | 0.85            |
| Number of guide vanes             | $Z_o$  |       | 20              |
| Number of impeller blades         | $Z_i$  |       | 12              |
| Mach number at the impeller inlet | $M_{1i}$|       | 1.02            |
| Round cycle efficiency            | $\eta_r$|       | 0.88            |
| Power                             | $N$    | kW    | 200             |
transformation of heat energy to work in the flow passage of the turboexpander. The distribution of pressure and temperature fields is uniform, and the velocity streamline distribution is neat and axisymmetric. Since the turboexpander is axisymmetric and to improve computational efficiency, a channel is selected for the numerical study. In addition, the mechanical rotation speed is high, mainly studying the influence of the turboexpander impeller parameters and the tip clearance on the turboexpander efficiency. Therefore, it is considered that the steady-state calculation can better study this work.

### Table 4: Grid independent verification of vane and impeller blade for turboexpander

| Grid  | Number of guide vane grids | Number of impeller grids | Working fluid flow/kg s⁻¹ | Expander efficiency/η |
|-------|---------------------------|--------------------------|---------------------------|-----------------------|
| Grid 1 | 128 000                   | 317 000                  | 2.974                     | 85.6%                 |
| Grid 2 | 247 000                   | 536 000                  | 3.012                     | 86.0%                 |
| Grid 3 | 541 000                   | 945 000                  | 3.004                     | 85.9%                 |

### Table 5: Boundary and initial conditions of nozzle and impeller blade for 200 kW turboexpander

| Initial condition          | Value | Unit  | Boundary condition            | Type                |
|---------------------------|-------|-------|-------------------------------|---------------------|
| Total inlet pressure      | 0.79  | MPa   | Guide vane inlet              | Inlet               |
| Total inlet temperature   | 354   | K     | Impeller outlet               | Outlet              |
| Outlet static pressure    | 0.21  | MPa   | Guide vane pressure surface   | Wall                |
| Reference pressure        | 0.1   | MPa   | Guide vane suction surface    | Wall                |
| Rotating speed            | 15 000| rpm   | Impeller pressure surface     | Wall                |
| Working fluid flow rate   | 12.9  | kg/s  | Impeller suction surface      | Wall                |
| Working fluid type        | R245fa|       | Two sides of the computational domain | Periodic |
| Intake direction          | Radial|       | Dynamic–static interface      | Mixing-plane        |
| Turbulence model          | SST   | model |                               |                     |

3 | Analysis of Turboexpander Parameters

The specific parameters studied in this paper are the external radius of the impeller blade outlet, the angle of the impeller blade, and the meridional section width of the impeller blade to improve the flow expansion acceleration in the impeller and the distribution of the flow field parameters at the impeller outlet, and to further improve the turbine efficiency. At the same time, the influence of different tip clearances on the performance of the turboexpander is discussed.
Effect of outer diameter of impeller blade on expander

The outer radius of the impeller blade outlet has an essential influence on the efficiency of centripetal turbine expander. Excessive external radius increases the velocity difference between the hub and rim of the working fluid, increasing secondary flow loss, and at the same time, it does not reflect the characteristics of runoff. The outer radius is too small, the contact area between the working fluid and the blade decreases, and the moment of working fluid to the blade decreases accordingly, which reduces the working force of the expander. Therefore, the outer radius of the impeller blade outlet is a vital dimension parameter. Choosing the appropriate outer radius is helpful to improve the flow field performance. Figure 3 shows a comparison diagram of the different outer radius of the outlet.

Figure 4 shows the variation in velocity and temperature at the outlet of the impeller blade with the outer radius of the impeller blade. From the picture, it can be seen that the impeller outlet velocity decreases from 227.24 m/s to 184.32 m/s and decreases by 18.89%, with the increase in the impeller blade outer radius. The outlet temperature of the impeller rises from 324.8 K to 326.14 K, up by 0.4%, and the temperature increases slightly. This is because the increase in the outer radius of the blade strengthens the outlet area of the blade but weakens the expansion performance of the working fluid at the impeller torsion, and some of the working fluid flows out of the impeller without expansion, which reduces the velocity of the working fluid at the outlet and increases the temperature.

As can be seen from Figure 5, the efficiency of expander decreases and the output power increases with the increase in the outer radius of the impeller blade outlet. Because with the rise of the outer radius, the velocity difference between the hub and rim of the working fluid increases, so does the pressure difference and the secondary flow loss, but the efficiency of the expander decreases. As the outer radius of the impeller blade increases, when the working fluid acts on the blade with the same force, the moment generated on the blade increases, and the expander’s working force increases.

The effect of the change in the external radius of the impeller blade on the mass flow is shown in Table 6. When the diameter of the moving blade is from 24 mm to 34 mm, the and the temperature increases slightly. This is because the increase in the outer radius of the blade strengthens the outlet area of the blade but weakens the expansion performance of the working fluid at the impeller torsion, and some of the working fluid flows out of the impeller without expansion, which reduces the velocity of the working fluid at the outlet and increases the temperature.

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| The external radius of the impeller blade (mm) | Mass flow (kg/s) |
|-----------------------------------------------|-----------------|
| 24                                            | 12.942          |
| 26                                            | 12.992          |
| 28                                            | 13.103          |
| 30                                            | 13.31           |
| 32                                            | 13.404          |
| 34                                            | 13.567          |
mass flow rate is from 12.942 kg/s to 13.567 kg/s, and the mass flow rate is increased by 0.625 kg/s.

3.2 | Effect of blade angle of impeller blade on expander

The impeller blade angle is the angle between the direction of the impeller blade and the meridian plane. It directly affects the shape and area of the flow passage of the impeller blade and has a robust and decisive role in the flow field. Flexible impeller blade angle design can prevent the formation of eddy current, reduce the irreversible loss caused by flow separation, and improve the efficiency of turboexpander, so it is necessary to study the influence of impeller blade angle on the effectiveness of the expander. In this paper, six different types of blade angle distribution schemes are arranged, and the advantages and disadvantages of these six schemes are compared, as shown in Figure 6.

Figures 7 and 8 show the calculation parameters of six kinds of impeller blade angles. It can be seen from the figure that with the increase in impeller blade angles, the outlet velocity of the blade increases continuously, from 217.3 m/s to 233.09 m/s, an increase of 7.4%. The outlet temperature of the impeller blade decreases gradually, from 325.63 K to 324.75 K, with a small decrease. Expander efficiency and output power also showed a decreasing trend. Expander efficiency decreased from 83.86% to 77.21%, reduced by 33.22%, and output power decreased from 239.08 kW to 163.96 kW, reduced by 31.4%. Because the increase in the blade angle causes the working fluid to expand more work near the exit of the blade instead of in the impeller flow path, which increases the residual speed loss at the trailing edge of the impeller blade, making the efficiency of the expander reduce.

The effect of the change in the angle of the moving blade on the mass flow rate is shown in Table 7. When the angle of the moving blade is from 20° to 45°, the mass flow rate is from 13.319 kg/s to 12.857 kg/s, and the mass flow rate is reduced by 0.462 kg/s.

3.3 | Effect of meridional section width of impeller blade on expander

Figure 9 shows the diagrams of different meridional section widths. The width of the meridian plane is the width of the

| TABLE 7 | Variation in mass flow with the impeller blade angle |
| --- | --- |
| The impeller blade angle (°) | Mass flow (kg/s) |
| 20 | 13.319 |
| 25 | 13.305 |
| 30 | 13.167 |
| 35 | 12.959 |
| 40 | 12.895 |
| 45 | 12.857 |

| TABLE 8 | Calculation parameters under different meridional section width |
| --- | --- | --- | --- | --- | --- |
| Meridional section width (mm) | Impeller blade outlet velocity (m/s) | Impeller blade outlet temperature (K) | Isentropic efficiency (%) | Turboexpander power (kW) |
| 30 | 219.88 | 325.03 | 82.26 | 244.62 |
| 32 | 219.37 | 324.98 | 82.46 | 245.68 |
| 34 | 219.02 | 324.86 | 82.65 | 246.18 |
| 36 | 218.61 | 324.73 | 82.69 | 246.62 |
| 38 | 218.35 | 324.56 | 82.75 | 247.16 |
| 40 | 218.1 | 324.47 | 82.85 | 247.39 |
transverse section of the impeller. The width of the meridional section directly affects the flow distance of the working substance in the impeller. Therefore, the influence of the meridional section widths on the performance of expander is analyzed.

From Table 8, it can be seen that the meridional section width has few effects on the parameters of the expander. When the width of meridional section changes from 30 mm to 40 mm, the velocity of impeller blade outlet decreases by 0.8%, the temperature of impeller blade outlet reduces by 0.17%, the efficiency of expander increases by 0.72%, and the output power of expander increases by 1.13%. Therefore, the meridional section width has few effects on the performance of the expander. It is suggested that in a reasonable range, the smaller meridional section width should be chosen as far as possible so that the size of the expander can be reduced and the material can be saved to reduce the cost.

4 | EFFECT OF BLADE TIP CLEARANCE ON TURBOEXPANDER

In the process of designing turboexpander, there will be a small gap between the wheel flange and the impeller blade to prevent friction between the blade and the wheel flange.
wall. Figure 10 shows the geometry of the impeller flow path. When the working fluid flows into the impeller, a small part of the working fluid enters the blade tip clearance under the action of centrifugal force and pressure difference between the two sides of the impeller, leaking from one side of the impeller to the other side, and the remaining working fluid expands in the impeller passage. Table 9 shows different blade tip clearance schemes.

A total of 20% leaf height, 40% leaf height, 60% leaf height, and 80% leaf height were numerically simulated in this study, as shown in Figure 11. The pressure contour in the impeller blade is sparse near the leading edge and dense near the trailing edge, indicating that the expansion of the organic working fluid in the impeller blade mainly occurs at the trailing edge, and the impeller does work further during the expansion process. On the whole, the isobar and the flow field in the impeller blade have a certain angle, which is caused by the presence of shock waves and uneven flow in the flow field. In the study of tip clearance, the contrast between 20% leaf height and 80% leaf height is more obvious. Therefore, this paper makes an analysis and comparison between 20% leaf height with or without tip clearance and 80% with or without tip clearance. As shown in Figure 12, at the 20% leaf height, the pressure contour changes without the tip clearance are 6.23 × 10^5 pa, 5.71 × 10^5 pa, 5.18 × 10^5 pa, 4.65 × 10^5 pa, 3.57 × 10^5 pa, 2.0 × 10^5 pa, respectively. The pressure contours with the tip clearance are 6.79 × 10^5 pa, 5.72 × 10^5 pa, 5.1 × 10^5 pa, 4.1 × 10^5 pa, 3.58 × 10^5 pa, and 2.1 × 10^5 pa, respectively. There are a tip clearance and no tip clearance at 20% leaf height. The pressure difference is between 0.1 × 10^5 pa–0.56 × 10^5 pa. From the overall level, the pressure contours of the two have little change. At 80% of the blade height, the pressure contour with tip clearance no longer decreases along the flow field direction. Basically, the low-pressure zone begins to appear at the inlet of the suction surface of the impeller, and the pressure is about 5.2 × 10^5 pa, which runs through the flow field passage of the impeller. The pressure field without tip clearance basically decreases along the flow field direction. According to the analysis, the influence of tip clearance on the flow field of the impeller is related to the direction of blade height. The tip clearance at 80% of the blade height has a greater impact on the flow field near the top of the blade, while the tip clearance has

![Pressure Contours](image_url)

**FIGURE 12** Distribution of pressure contours with or without blade tip clearance
a smaller impact on the flow field at the root of the blade as a whole.

Figure 13 is a pressure contour map at different cross sections along the flow field direction, as shown in Figure (A). It can be seen from Figure (B) that there is a pressure gradient along the circumference of the guide vane when the organic working gas enters the moving vane. As can be seen from the Figure (C), when organic working gas enters the impeller, a positive pressure gradient from the pressure surface to the suction surface occurs at the leading edge of the blade. It can be seen from the Figure (D) that in the cross section where the runner turns from radial to axial, the velocity difference between a hub and rim increases due to the different radius of hub and rim, resulting in the pressure on hub side being greater than that on rim side. Along the radius direction, the pressure decreases from hub to rim, while along the circumferential direction, the pressure still keeps decreasing trend from pressure surface to suction surface. As shown in Figure (E), because of the rapid expansion of working gas caused by the torsion of the top section at the outlet of the moving blade, a high-pressure zone is created on the pressure surface, which drives the impeller to rotate,

**FIGURE 13** Pressure distribution along different sections of the channel

(a) Schematic diagram of different flow directions

(b) 1-1 section  (c) 2-2 section

(d) 3-3 section  (e) 4-4 section

(f) 5-5 section
and a low-pressure area is created on the suction surface. As can be seen from Figure (F), when the gas expanding at the trailing edge of the blade flows out of the blade, the airflow expands further due to the round corner of the blade, and the pressure increases with the speed decreases, forming two low-pressure zones, which are consistent with the results of Figures 11 and 12.

When there is a gap at the top of the blade, the working fluid leaks from the pressure side of the blade to the suction side. Figure 14 shows the velocity streamlines distribution in the impeller passage of the turboexpander, and Figure 15 shows the velocity vector diagram of blade tip clearance. From Figure 14, it can be seen that when there is tip clearance, streamlines pass over the top of the blade and move to the trailing edge after mixing with the mainstream from the pressure side to the suction side. The fluid flowing to the suction surface will mix with the fluid flowing in the main direction to produce eddy current, which will affect the performance of the flow field and weaken the flow performance. Figure 15 shows that the velocity vector at tip clearance points from one side of the pressure surface to one side of the suction surface, indicating that there is fluid flowing through tip clearance. And the leakage area of the tip clearance runs through the entire impeller area, the closer to the trailing edge, the higher the velocity of the leakage fluid.

Due to the existence of the tip clearance, the working medium is ejected from the suction side of the blade through the clearance under static pressure difference and mixed with the mainstream fluid to generate a gap vortex flow along the flow direction. Also, the viscous forces in the gap cause a large shear force between the top of the blade and the fixed casing, which in turn causes air to flow from the pressure face to the suction side of the adjacent blade. Because of the velocity difference between the outflow fluid and the mainstream fluid, the tip clearance vortices will form. The tip clearance vortices will change the pressure field at the top of the blade and make the pressure load at the top of the blade unstable. Figure 16 shows that the eddy current at tip clearance of 1.2 mm is much larger than that at 0.2 mm, which indicates that the eddy current in tip clearance increases with the increase in tip clearance. The eddy current caused by the flow in the tip clearance runs through the whole impeller flow passage and increases continuously from the impeller inlet to the impeller outlet. These eddies interfere with the flow in the main flow area mixed with the flow in the main flow area. These eddies form a backward diffusion toward the suction surface, which reduces the efficiency of the turbine.

As can be seen from Figure 17, according to the results of numerical simulation, the existence of tip clearance dramatically reduces the output power of the turboexpander. With the increase in tip clearance, the efficiency of the expander decreases continuously, showing a linear downward trend. When the tip clearance increases from 0 mm to 1.2 mm, the effectiveness of the expander reduces by 8.9%. It can be seen that the tip clearance has a significant influence on the ability of the expander, because of the increase in tip clearance. On the one hand, when the tip leakage flow in the expander increases, the flow loss increases, and the efficiency decreases. On the other hand, due to the existence of the leakage flow, the gas flow used for the impeller is reduced. The force at the
tip of the blade decreases with the increase in the tip clearance, because the dynamic pressure effect at the tip of the blade is weakened with the rise of the tip clearance resulting in the smaller force at the tip of the blade the lesser void and the more significant dynamic pressure effect. Moreover, as the dynamic pressure effect at the tip of the blade decreases, the fluid on both sides of the blade can pass through the air film at the tip of the blade more easily, because the pressure difference between the two sides of the blade is as follows: one side is on the pressure surface, and the other side is on the suction surface. As a result, the fluid on both sides of the moving blade tends to flow. With the decrease in the dynamic pressure effect, the working medium on the pressure side is more likely to flow into the suction side through the gap, which makes the formation of the gap vortex easier.

4.1 Result comparison

Modern loss modeling includes six fundamental loss models, and these are incidence loss, passage loss, trailing edge loss, clearance loss, windage loss, and exit energy loss. Therefore, the turbine efficiency can be written as follows:

\[ \eta_{\text{turbine}}(\%) = (1 - L_{\text{total}}) \times 100 \]  

where, \( L_{\text{total}} = L_i + L_p + L_c + L_w + L_e \)

By comparing the loss models introduced in the literature, such as Baines,\(^{21}\) Ventura et al.,\(^{22}\) Ghosh et al.,\(^{23}\) Qi et al.,\(^{24}\) Wei,\(^{25}\) and Hu et al.,\(^{26}\) in this paper, the research of tip clearance is mainly focused on. Six loss models are analyzed. Five of them are the same. Two models of tip clearance loss are analyzed in this paper. They are as follows:

\[ \eta_{\text{turbine}}(\%) = (1 - L_{\text{total}}) \times 100 \]
where \( U_\text{in} \) — Rotor speed at inlet \([\text{m/s}]\), \( Z_r \) — number of blades on the rotor.

\[
U^3 \frac{Z_r}{8\pi} \left( K_x \epsilon_x C_x + K_r \epsilon_r C_r + K_x \sqrt{\epsilon_x \epsilon_r C_x C_r} \right)
\]

(2)

where \( U_\text{in} \) — Rotor speed at inlet \([\text{m/s}]\), \( Z_r \) — number of blades on the rotor.

\[
K_x = 0.4, K_r = 0.75, K_xr = -0.3
\]

\[
C_x = \frac{1 - \left( \frac{r_{\text{out},p}}{r_{\text{in}}} \right)}{C_{m,\text{in}} b_{\text{in}}}
\]

\[
C_r = \left( \frac{r_{\text{out},p}}{r_{\text{in}}} \right) \left( \frac{z_r - b_{\text{in}}}{C_{m,\text{out}} r_{\text{out},p}^2 b_{\text{out}}} \right)
\]

where \( r \) — radius \([\text{m}]\), \( C_m \) — meridional component of velocity in an absolute frame \([\text{m/s}]\), \( b \) — blade height \([\text{m}]\), \( z_r \) — axial length of rotor.

\[
\Delta \eta_t = 1.355 - 0.08371A - 1.772R - 0.2285AR + 0.9725R^2
\]

(3)

where \( A \) — area \([\text{m}^2]\), \( R \) — universal gas coefficient.

The isentropic efficiency of the turboexpander in this paper decreases from 84.4% to 75.5% with the increase in tip clearance. The total energy loss in this paper ranges from 15.6% to 24.5%. The efficiency of the expander decreases by 8.9% with the change in tip clearance because other conditions remain unchanged, and only the variable of tip clearance is changed. By calculating the tip clearance loss model of Equations (2) and (3), as the tip clearance increases, the effect of the tip clearance on the expander efficiency is shown in Table 10. The content of this paper is that the expander efficiency decreases by 8.9% with the change in the tip clearance. The prediction results of the empirical loss model are slightly different from the numerical simulation results in this paper. The tip clearance loss models of Equations (2) and (3) have reliability for the study of turboexpanders.

5 | CONCLUSIONS

Through the analysis of blade and tip clearance of impeller, it can be seen that the isentropic efficiency of expander decreases and the output power increases with the increase in outer radius of the impeller blade. The larger the outer radius of the impeller blade, the higher the velocity difference between the hub and rim of the working fluid, the more significant the secondary flow loss and the smaller the outer radius of the impeller blade, the lower the working capacity of the expander. With the increase in the blade angle, the efficiency and output power of the expander are reduced, so it is not appropriate to adopt too large blade angle. When the width of the meridian plane increases from 30 mm to 40 mm, the efficiency of expander increases by 0.72%, and the output power of expander increases by 1.13%. It is indicated that the width of the meridian plane has few effects on the performance of expander. The existence of tip clearance will cause tip clearance flow between pressure surface and suction surface, which will interfere with the mainstream direction and reduce the efficiency of the expander. The eddy current caused by tip clearance increases with the increase in tip clearance, and the eddy current runs through the entire impeller flow channel. According to the loss model of formula (2), the efficiency of expander decreases by 1.49%, 1.64%, 1.76%, 1.89%, 2%, and 2.1%, respectively, with the increase in tip clearance. The loss model of formula (3) predicts that the efficiency of expander decreases by 1.08%, 1.19%, 1.28%, 1.37%, 1.47%, and 1.49%, respectively, with the increase in tip clearance. Through the calculation and prediction of the two-loss models, the loss model of Equations (2) and (3) reduced the expander efficiency by 10.88% and 7.88% in total. The content of this paper is that the expander

| Empirical loss model | Expander efficiency loss (%) | Total efficiency loss (%) |
|----------------------|----------------------------|--------------------------|
| The loss model of formula (2) | 1.49 1.64 1.76 1.89 2 2.1 | 10.88 |
| The loss model of formula (3)      | 1.08 1.19 1.28 1.37 1.47 1.49 | 7.88 |

FIGURE 17 The influence of the blade tip clearance on the entropy efficiency and the tip force

TABLE 10 Prediction of changes in tip clearance by empirical loss model
efficiency decreases by 8.9% with the change in the tip clearance. The prediction results of the empirical loss model are slightly different from the numerical simulation results in this paper. The tip clearance loss models of Equations (2) and (3) have reliability for the study of turboexpanders.

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CONFLICT OF INTEREST

The author declares no conflict of interest.

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