Comprehensive Calculation And Performance Analysis Of Gas Turbine Reversible Power Turbine

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Abstract
Gas turbine technology trends to be maturing now, but the problem of which not being able to reverse directly remains resolve. In the field of marine, most ships reverse by adjustable pitch propellers, which limit turbine power and increases navigational resistance and maintenance costs. This paper designs a directly reversible turbine – based on a four-stage turbine, which is added a single-stage turbine to achieve the goal of direct reverse for the turbine. The reversible turbine is external to the four-stage turbine, and they share a turbine shaft to work. The rotor of the reversible turbine is connected with the last stage rotor of the four-stage turbine, and the double-layer blade is formed by the intermediate ring. The upper and lower parts of the double-layer blade have opposite profile shape, which allows the redesigned turbine to rotate positively and reversely. This paper uses CFD software to simulate the work conditions of the reversible turbine. The flow loss in the internal flow field of the redesigned turbine and the four-stage was analyzed in this paper, and find that the flow separation is the main factor to reduce efficiency of the reversible turbine.

Keywords: reversible turbine, integrated design, windage loss, variable working conditions

Introduction
Gas turbine technology is becoming more and more perfect nowadays. As one of the main power equipment, the gas turbine has a serious drawback, it cannot be directly reversed. Over the years, in order to solve the problem, people have spent a lot of effort, but so far there is no ideal result. If the gas turbine can be directly reversed, it will reduce the time of asternway for the ship, and boost maneuverability and fuel efficiency.

At present, the reversing problem of gas turbines is mainly realized by adjustable pitch propellers, but its transmitted power has the upper limit. The larger size of the underwater component compared with the fixed pitch propeller results in about 10% increase in hull resistance at full power, while a 6% increase in drag under cruising, and the construction of the adjustable pitch propeller is complex and difficult to maintain (Niu et al., 2016).

The study of reversible gas turbines began roughly in the early 1970s, and now there are two forms of directly reversible turbine: one is called blocking type and the other is called baffle type. Allen, R. P. et al.,1967 introduced the potential advantages of axial-flow Marine gas turbine and demonstrated the aerodynamic and mechanical feasibility. In the early 1970s, GE Corporation of the United States conducted the development and testing of a directly reversible gas turbine that is blocking type. The back-to-back arrangement of the power turbine and the reverse turbine enables the output shaft to rotate forward and backward (Niu et al.,2016). This form of reversible gas turbine has much energy loss in the process of gas transportation and its efficiency is low. Later, Rucigay, J. C. et al.,1976 and Bowen, T. L. et al.,1979 studied the feasibility of the concept of independent reversible turbine in ship propulsion; so far, the subsequent results tend to support the feasibility of the reverse turbine concept.

The designers of the “Machine Design” scientific production Consortium in Ukraine have already developed a gas turbine that can directly reverse. Its structural form belongs to the baffle type. This directly reversible gas turbine has two inner and outer flow passages: the outer flow passages are reverse flow passages, and the inner flow passages are forward flow passages; the reversible turbine is located above the last stage of the forward turbine, and the front and the moving buckets. There are two forms of changes for the flow path: one is achieved by a circular steering valve, the other is to set a baffles at the front of the forward and astern flow channels. When the baffle is lowered, the flow path of forward turbine is blocked, the reversing system starts to...
work, and when the baffle is lifted, the flow path of reversible turbine is blocked, and the forward turbine works. Because of the existence of the baffle, leakage losses are formed, which reduces efficiency.

Russia's research results were the most successful, Yu et al., 2016. Ships equipped with reversible turbine were introduced in 1969. These ships almost rotate around the "point" by the gas turbines equipped on both sides of the ship. But now gas turbines still cannot reverse directly.

Niu et al., 2016 designed a two-channel reversible turbine that can be directly reversed for the problem of turbine reverse, and carried out corresponding simulation and experimental research (Niu et al., 2017). However, they didn't study the flow loss mechanism in the reversible gas turbines and explore the factors affecting efficiency.

Through the analysis of the existing data of reversible turbine technology at home and abroad, a gas turbine with reasonable design, low manufacturing cost and direct reversing function is proposed. Literature mainly put forward the concept of reversible turbine and analyze its great potential. There are few studies on the design and flow of reversible turbines.

This paper studies the internal flow field of reversible turbine, analyzes their internal flow characteristics and finds out their important loss sources. The flow loss of the internal flow field of the reversible turbine and the loss mechanism was discussed. This study provides guidance for the optimization research of reversible turbines in the future.

Numerical method

Physical model

In this paper, the reversible turbine has been redesigned and numerically simulated by CFD software. This method is used to simulate the performance of the turbine in ahead rotation, astern rotation and rotation with double channel interaction.

The cross section of power turbine with reversible turbine stage is shown in Fig 1. The redesigned turbine is a double-layer structure: the outer is reversible turbine that has only one stage; the inner is the Ahead one, which has four stages. The redesigned turbine can rotate positively or reversely with the lift and fall of the baffle. The reversible turbine adopts a double-channel structure. When the baffle is raised and located in position 1, the gas coming out of the combustion chamber enters the inner passage that is connected with the four-stage turbine, while the structure of the inner passage is the same as the conventional turbine structure, and the external output is done by the four-stage turbine. When the baffle falls and is located in position 2, the gas enters the outer passage, and the output work is done by the reversible turbine. The specific data about the geometry is shown in Table 1.

Calculation model

The redesigned turbine structure is here used to design the calculation model. The turbine is divided into two parts: the four-stage turbine and the reversible turbine, for getting detailed calculation results. In the calculation process, the two parts are calculated individually, and finally the two are integrated for overall calculation. The calculation model is shown in the Figs. 2, 3 and 4. The Ahead turbine is a four-stage power turbine, it is shown in Fig. 2. With the increase of the stage, the blade height increases gradually. The reversible turbine is located in a higher position; and there is a long passage which is connected to the inlet, as shown in Fig 3. In the integral joint calculation of the turbine, the gas selection channel, simulating baffle opening and closing, is added in front of the two turbines; and a 1mm gap is left between the two channels of the gas selection channel in consideration of tightness. At the same time, an exhaust passage is added behind the turbine, which is shown in Fig 4.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
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<tbody>
<tr>
<td>Hub radius(mm)</td>
<td>610</td>
</tr>
<tr>
<td>Outlet casing radius(mm)</td>
<td>958</td>
</tr>
<tr>
<td>The rotor blade height(mm)</td>
<td>50</td>
</tr>
<tr>
<td>span-chord ratio</td>
<td>1.9</td>
</tr>
</tbody>
</table>
Meshing
NUMECA8.9.1/Autogrid5 was used to generate structured grids for the power turbine. Three-dimensional computational grids of the baseline power turbine and reversible power turbine are shown in Fig. 5. Meanwhile, the leading edge and trailing edge of the blade, blade tip clearance, blade wall surface, hub and shroud wall surface of the blade are encrypted during the grid division., the thickness of the first layer of grid on the solid wall surface is 0.001mm. The static pressure coefficient is defined as follows:

\[ C_p = \frac{p}{p_1} \]  

(1) 

\( p_1 \) is relative total pressure of inlet; \( p \) is static pressure of the blade surface. The grid sensitivity was verified by the static pressure coefficient of stator blade in the reversible turbine. Grid independence validation is shown in Fig 6. Based on the verification of irrelevance to grid, the power turbine is divided into 2.84 million grids, the reversible turbine is divided into 2.7 million grids and the integral reversible turbine is divided into 6.07 million grids. In order to assure the calculating precision, the maximum of dimensionless distance \( y^+ \) on the wall is about 5.

Boundary conditions
For boundary conditions, total temperature, total pressure, and flow direction are specified at the inlet boundary. Static pressure is specified at the outlet. The periodic boundary conditions are used along the circumferential direction. No-slip condition is applied and all blade walls and end-walls are assumed to be adiabatic.
The tip clearance of rotor blade in four-stage turbine is 1mm, while the reversible turbine stage temporarily ignores the rotor blade tip clearance. Considering the load capacity and power output of the reversible turbine, the fuel delivery and output of the compressor will be reduced when reversing, and the inlet temperature and flow of the reversible turbine stage will be reduced. At the same time, the power output of the combustion chamber and the compressor is reduced, which is conducive to the switching smoothly and safely between the two modes. So the design index of reversible turbine is lower than ahead turbine. The specific parameters, which refer to the actual operating parameters of the turbine, are listed in table 2. The turbine flow field and aerodynamics were calculated by CFD software ANSYS CFX 17.0. The CFX code employs a pressure-based formulation to solve the compressible Reynolds-Averaged N-S transport equations. This study is based on literature 4 and select the Same turbulence model (standard k-ω two-equation model) in this paper. High resolution is selected for turbulence numerics and auto timescale is chosen for timescale. The convergence criteria for the CFD computations are that the reduction of RMS residuals below 1×10⁻⁴ and the convergence curves of flow rate, pressure ratio and isentropic efficiency tend to stabilize.

<table>
<thead>
<tr>
<th>Table 2 The boundary condition parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
</tr>
<tr>
<td>Total-to-static pressure ratio</td>
</tr>
<tr>
<td>Angular velocity(RPM)</td>
</tr>
</tbody>
</table>

Results and discussion
In order to explore the actual working performance of the turbine, numerical simulation calculations of the turbine were carried out. This section will discuss the internal flow and loss mechanism of the turbine in detail.

Analysis of results under the single calculation and joint calculation for the four-stage turbine
As shown in Fig. 7, the Mach number distribution at the 5% blade height of the four-stage turbine in single and joint calculations, the two simulation results are similar. In the joint calculation, because of the increase of the length of inlet channel and the leakage at the baffle, the loss of turbine increases, and the flow velocity decreases, so that the maximum Mach number of the turbine is lower than the maximum Mach number in single calculation. The Mach number at the turbine outlet is also relatively low. In the four-stage turbine, the relative Mach number is always less than 1, which reduces the transonic loss of the airflow in the turbine, and improves efficiency and output of the turbine.

Fig. 7. The Mach number distribution of the four-stage turbine.

The exit flow angle of the cascade is one important indicator for measuring the aerodynamic performance of the turbine. Whether the outlet airflow angle of the turbine cascade can be accurately obtained has a great influence on measuring the turbine shaft power. As the exit airflow angle of the cascade decreases, the output of the turbine increases, but if the exit airflow angle too small, which would increase the energy loss in the turbine cascade. The exit airflow angle is defined as follows:

$$\alpha = \arctan \left( \frac{w}{v} \right)$$

where $w$ is relative axial velocity of airflow; $v$ is the circumferential component of the absolute velocity.

Figure 8 shows the exit airflow angle distribution of each stage stator blade in the four-stage turbine in the single calculation and joint calculation. In the simulation calculation, the stator blades are fixed and connected to the shroud end-wall and the hub end-wall in the air passage. When the airflow flows through the stator cascades, there is some frictional resistance loss because of the frictional resistance on the end-wall surface. In the end-wall boundary layer, the velocity of airflow becomes lower or even zero, and the pressure increases. A positive pressure gradient in the direction toward the centre of the flow channel is generated near the surface of the end-wall, and that causes a part of the airflow to flow from the end-wall to the flow channel. At the same time, the boundary layer near the surface of the shroud end-wall and hub end-wall, interacting with the circumferential boundary layer on the blade surface, becomes thicker, which causes the birth of a pair of channel vortices at the top and the root of the blade. As the stator blade bends, the intensity of the passage vortex increases in the downstream, and the outlet airflow angle increases. As shown, the exit airflow angle at 5% and 80% height along the blade is greater than the exit airflow angle in the middle of the blade, so two peaks are formed on the curve of the outlet airflow angle.
(a) The first stage stator blade

(b) The second stage stator blade

(c) The third stage stator blade

(d) The fourth stage stator blade

**Fig. 8.** Distribution of exit airflow angle in each stage stator blade of the four-stage turbine in the single calculation and joint calculation.

The four-stage turbine is additionally provided with a device, which selects flow passage and adjust the inlet of the turbine, and an outer flow passage for the reversible turbine. The added devices result in an increase of the total pressure loss, changing the flow state of the airflow in the four-stage turbine, so that the intensity of the passage vortex increases in the first stage stator blades and the peak on the curve of the outlet angle becomes larger. The installation of the reversible turbine affects the flow near the shroud end-wall surface in the four-stage turbine, and the outlet airflow angle increases at the top of the stator blades.

In general, when the reversible turbine is operated together with the four-stage turbine, the introduction of the reversible turbine has less influence on the distribution of exit airflow angle in each stage stator blades of the four-stage turbine. That is nearly maintained in a same state in which the four-stage turbine is operating alone.

Figure 9 shows the distribution of exit airflow angle in each stage rotor blade. In the rotor blades of the four-stage turbine, the airflow pushes the blades to work. In order to increase the output work, the rotor blade bends sharper than the stator blade, so that the exit airflow angle is relatively larger. The outlet airflow angle near the hub is affected by the passage vortex, so that the relative exit airflow angle is larger. As the blade height increases, the relative circumferential velocity of the airflow increases continuously; with the influence of the tip clearance vortex, the upper passage vortex at the tip of the blade is suppressed. So the outlet airflow angle becomes smaller as the blade height increases, so that the exit airflow angle curve only has a peak at the blade root.
Fig. 9. Distribution of the exit airflow angle in each stage rotor blade of the four-stage turbine in the single calculation and joint calculation.

The inlet pressure of the rotor blades is greater than the outlet pressure. At the tip of the rotor blades, because of the existence of the tip clearance, the airflow can flow directly from the gap, causing a loss of air leakage, so the angle of airflow at outlet of each stage rotor blades rapidly increases at the tip of the rotor blades. In the joint calculation, the fourth stage rotor blade of the four-stage turbine and the rotor blade of the reversible turbine are connected by the intermediate ring to form a double-layer blade. When working together, the intermediate ring and the two blades rotate together about the shaft, there is no tip clearance, and so, flow angle will not increase rapidly.

The total pressure loss coefficient is used to measure the share of the cascade pressure drop. The larger the total pressure loss coefficient, the greater the pressure loss, which further affects the turbine efficiency. The coefficient is defined as follows:

$$C_{p,t} = \frac{P_1^* - P_2^*}{P_1^* - P_1}$$  \hspace{1cm} (3)

where $P_1^*$ is relative total pressure of inlet; $P_2^*$ is relative total pressure of outlet; $P_1$ is relative static pressure of inlet.

Compared the total pressure loss coefficient of each stage cascade in the four-stage turbine under the single calculation and the joint calculation, the total pressure loss coefficient of each cascade is analogous between the two simulation results showed in Fig. 10.
In the joint calculation, the fourth stage rotor blade in the redesigned turbine is double-layered, and observing the cloud map which belongs to the fourth stage rotor blade in the Fig. 11; compared with the four-stage turbine in the single calculation, there is no tip clearance vortex, which reduces the total Pressure loss. In the single calculation, the loss of the fourth-stage turbine is mainly composed of the passage vortex loss, the tip clearance vortex loss and the wake mixing loss, while doing the joint calculation, there is no tip clearance vortex loss in the fourth stage rotor blades. Seeing the pressure loss coefficient curve along the blade height, the total pressure loss coefficient near the tip of the blade is significantly reduced due to the absence of the tip clearance vortex in the joint calculation.

In the turbine, the loss of energy is accompanied by the increase of entropy. The increase of entropy is an irreversible process, the larger the loss, the more the entropy increases. The entropy-increasing graph of the turbine cross section in the single calculation and joint calculation is shown in Fig. 12.

Comparing the entropy-increasing graphs under the two calculation conditions, the isentropic line of the turbine internal cross section is similar to the anti-“C” shape; and the entropy at the surface of the shroud end-wall and hub end-wall is higher than the entropy of
the central channel. The reason is: when the airflow flows through the passage, the airflow boundary layer, near the shroud end-wall and hub end-wall, is generated because of the frictional resistance; the pressure at the boundary layer is higher than the pressure of gas flowing in the channel so that the direction of partial airflow changes with the pressure gradient and the passage vortex generates. There is also a tip clearance vortex generated because of the tip clearance. Channel vortices and tip clearance vortex multiply the energy loss, resulting in that entropy increases evidently in the flow field. In the center of the channel, the profile loss and wake mixing loss at the outlet of the cascade cause the entropy of the airflow to increase.

It can be seen that the loss of the four-stage turbine is mainly the passage vortex loss, the tip clearance vortex loss and the wake mixing loss from the two pictures. In the joint calculation, the change trend of entropy is the same as that in the single calculation. Since there is no tip clearance vortex loss at the tip of the last stage rotor blades in the joint calculation, the maximum value of entropy is lower than the maximum value of entropy in single calculation. However, in the joint calculation, the profile loss and the wake mixing loss are so large that the increase of entropy in the overall turbine is larger than the increase of entropy in the single calculation. That is, the loss of the turbine in the joint calculation is greater than the loss in the single calculation.

Under the reversing condition, the four-stage turbine reverses and is in abnormal working condition. The four-stage turbine might compress the working medium. However, under actual working condition, less gas is leaking from the inlet and it is difficult to be compressed again. While the ahead turbine is inverted, the leaked gas keeps expanding with pressure dropping in the four-stage turbine. At the same time, inversion results in a local high pressure region that is located at the front edge of rotor pressure surface. At the outlet of the final rotor blade, the pressure increases due to the influence of the working medium flowing from the reversible turbine.

The overall redesigned turbine consists of a four-stage turbine, a reversible turbine, a device selecting inlet passage and an outlet. Compared to a separate four-stage turbine, the overall reversible turbine additionally increases the loss at the inlet passage and outlet. Moreover, there is a loss at the double-layer blades and the baffle because of gas leakage. The introduction of the reversible turbine results in a reduction of the efficiency and power of the turbine. The specific data is shown in Table 3.

**Table 3 Performance parameters of the four-stage turbine**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Single calculation</th>
<th>Joint calculation</th>
<th>Experiment</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total pressure ratio</td>
<td>3.39</td>
<td>3.41</td>
<td>3.483</td>
</tr>
<tr>
<td>Isentropic efficiency</td>
<td>94.22%</td>
<td>93.16%</td>
<td>92.49%</td>
</tr>
</tbody>
</table>

Compared the pressure ratio and isentropic efficiency of the four-stage turbine between single calculation and joint calculation, the total pressure ratio of the four-stage turbine under the joint calculation increases, while the isentropic efficiency decreases. The total pressure ratio increased by 0.02, the isentropic efficiency decreased by 1.06%, and the turbine power decreased by 0.08MW. In addition, the CFD results are better than the experiments. That is mainly because the experiment date is influenced by human operation and measuring equipment. Also, there is no perfect match between the equipment and the model. However, this proves the reliability and authenticity of the simulation results.

**Analysis of results under the single calculation and joint calculation for the reversible turbine**

When the reversible turbine and the four-stage turbine make up the integral reversible turbine, the length of inlet passage of reversible turbine increases. The speed of gas at the inlet of the static blades and the total pressure decreases; the pressure loss also becomes relatively larger. Airflow leakage occurs at the intermediate junction of the two-layer blades of the final stage in the four-stage turbine, which affects the flow field in the final cascade of reversible turbine. As shown in the Fig. 13, compared the Mach number distribution at the 5% blade height in the reversible turbine under single calculation with the joint calculation, the maximum of Mach number in the flow field in the reversible turbine is about 1.6 in joint calculation, which is reduced by about 0.3, compared with the single calculation. Under both operating conditions, the supersonic regions in the cascade and downstream of the cascade result in a large amount of energy loss, making the efficient of the reversible turbine less than the four-stage turbine.
Fig. 13. The Mach number distribution of the reversible turbine.

Similar to airflow outlet angle of stator blade in the four-stage turbine, there is a pair of channel vortices generated at the root of the blade on account of the effect of the boundary layer near the hub end-wall surfaces and the circumferential boundary layer on the blade surface. The channel vortices cause that the airflow outlet angle increases rapidly and reaches a peak showed in Fig. 14. However, in the reversible turbine, the location of passage vortex occur is close to the trailing edge of the stator blade, and the peak of the outlet angle is relatively smaller compared to the angle of stator blade in the four-stage turbine. At the tip of the vane, because of the curvature of the outer passage, the boundary layer at the shroud end-wall surface encounters the mixing of the upstream airflow, so that the turbulent zone at the end-wall layer increases, and the pressure gradient, from the shroud end-wall to the flow-path, increases. The airflow at the back of blade affected by the pressure gradient and the upstream flow flows downward obliquely, and the upper passage vortex is suppressed, so that the effect of the passage vortex at the tip on airflow angle is decreased.

Fig. 14. Distribution of exit airflow angle of the rotor blade and stator blade in the reversible turbine under the single calculation and joint calculation.

The reversible turbine is affected greatly by the four-stage turbine under the joint calculation. There is no passage vortex generated at the root of the vane, and so that the outlet angle is reduced. The reversible turbine has only one stage, and serious airflow separation occurs at the back of the blade, which results in that the airflow exit angle is larger than that in the four-stage turbine; and the workability of the reversible turbine is also lower than the four-stage turbine. The passage vortex and the tip clearance vortex are suppressed by the separation vortex. As shown in Fig. 15: there is no increase or decrease caused by the passage vortex and the tip clearance vortex. In the joint calculation, the airflow separation is more serious, the total pressure loss is larger, and the axial velocity of the airflow is decreased. The airflow outlet angle of the rotor blade is increased by about 4° compared with the single calculation.

As shown in Fig. 15, in the reversible turbine, severe airflow separation occurs at the back of rotor blade. In the joint calculation, the flow separation is more serious. The separation vortex caused by the separation of the airflow not only affects the interior of the cascade, but also spreads into the flow channel behind the exit of the cascade. With the influence of the separation vortex, the wake vortex is formed at the exit of rotor blades. These vortices cause a huge energy loss, which reduces the efficiency of the reversible turbine.
In the rotor blade of the reversible turbine, the large scale separation at the back of the blade is caused by the large hub radius and low cascade consistency. Since the separation occurs near the maximum thickness of the rotor blade, the turbine presents obvious preloading characteristics, and the load at the front of the rotor increases. This separation is the main reason for the decreasing efficiency of reversible turbine. How to reduce the separation at the blade back is the key to improve the efficiency of reversible turbine and the main direction of research.

In the future, studies can be carried out on inhibition separation, such as adjusting the distribution position of the maximum thickness of the rotor blade, using tandem cascade, etc. These need further study.

In the joint calculation, the reversible turbine has only one stage and is located at the last stage of the four-stage turbine. When the turbine is running, the reversible turbine is greatly affected by the four-stage turbine. The airflow, leaking into the passage of the four-stage turbine from the gap of the baffle and compressed by the four-stage turbine, flows into the reversible passage along the pressure gradient at the junction of the double-layer blades and causes the outlet pressure to increase. Seeing Fig. 16, Compared the total pressure loss coefficient of the static blades and the rotor blades, the result of the joint calculation is better than the single calculation because of the air leakage at the rotor blade; but in actual conditions, the aerodynamic performance of the joint calculation is worse than single calculation.

The static pressure coefficient of the rotor blade in the reversible turbine is shown in Fig. 17. In the two simulation calculations, the pressure at the back of the blade decreases rapidly owing to the accelerated flow of gas. The separation vortex is generated because of the flow separation, and there is a backflow against pressure gradient on the back surface of the blade. This separation of airflow causes the pressure to gradually increase again on the back surface of the blade. In the single calculation, the pressure starts to increase at 30% from the leading edge; while in the joint calculation, the air separation is more serious, and the location where the pressure is minimal is around 20% from the leading edge. Severe separation causes more energy loss and further decrease of turbine efficiency.

The isentropic efficiency curve at the exit of rotor blade in the reversible turbine is shown in Fig. 18: Compared the efficiency curves of the two calculations, the efficiency is significantly reduced in the joint calculation. Since the air separation at the back of the
blade is more serious and there is even the wake vortex generated, the pressure gradient between the wall and the flow path increases. The proportion that the gas flow to the center of the flow channel from the hub end-wall increases, so that the efficiency near the blade root is further reduced. At the top of the blade, as the gas flows in the cascade, the position of tip clearance vortex gradually moves down, which causes the isentropic efficiency to drop at 80% blade height.

**Fig. 18.** The distribution of isentropic efficiency along the blade height at the exit of the rotor blade in the reversible turbine under single calculation and joint calculation.

Under the ahead condition, the reversible turbine is in abnormal working state, and its pressure distribution is similar to that of the four-stage turbine under the reversing condition, which will not be repeated in this paper.

Compared the operating conditions of the reversible turbine between single calculation and joint calculation, when the reversible turbine is working with the four-stage turbine, the reversible turbine is greatly affected by the four-stage turbine, the loss is larger, and the efficiency is reduced. In order to reduce the influence of windage loss, the inlet and outlet parameters of the blade were selected to calculate turbine efficiency.

The change of the reversible turbine is similar to the four-stage turbine under the single calculation and the joint calculation. The total pressure ratio, from the inlet to outlet, and the isentropic efficiency decreases under the joint calculation. Since the reversible turbine has only one stage, the degree of variation of parameters is more severe than the four-stage turbine. The total pressure ratio is reduced by 0.14, the isentropic efficiency is decreased by 2.37% and the turbine power is decreased by 1.47 MW.

**Conclusions**

The gas turbine is redesigned and achieves the goal that the gas turbine can reverse directly by introducing a reversible turbine in this paper. The reversible turbine is external to the four-stage turbine, and they share a common turbine shaft to work. The rotor of reversible turbine is connected with the last stage rotor of the four-stage turbine, and the double-layer blade is formed by an intermediate ring. The upper and lower blades of double-layer blades have opposite blade shape, which allows the redesigned turbine to rotate positively and reversely. The internal flow field in the reversible turbine and the four-stage turbine when working alone and together is analyzed in this paper; the main factors that affect its efficiency are found; and some important parameters are obtained when they work normally.

1. In this paper, the internal flow fields of reversible turbines and four-stage turbines working separately and together are analyzed. In the reversible turbine, the loss is mainly caused by airflow separation and supersonic loss, in which airflow separation accounts for a large proportion.
2. When the turbine redesigned is rotating forward, the introduction of the reversible turbine has less impact on the working operation of the four-stage turbine. In the overall operation, the turbine efficiency can reach 93.16% and the output power is 24.78 MW.
3. When the turbine redesigned is reversing, the introduction of the four-stage turbine makes the separation of the airflow in the reversible turbine further deteriorated; the total pressure loss increases. The efficiency of the reversible turbine drops rapidly, reaching 53.37%, and the output power is 7.94 MW, which is 32% of the turbine power of the four-stage turbine.
4. The cascades in reversible turbine should be further adjusted, such as adjusting the distribution position of the maximum thickness of the rotor blade, and using tandem cascade. The better cascades would improve its internal flow field and reduce air separation loss.

<table>
<thead>
<tr>
<th>Nomenclature</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>S</td>
<td>The static blade of the four-stage turbine under single calculation</td>
</tr>
<tr>
<td>R</td>
<td>The rotor blade of the four-stage turbine under single calculation</td>
</tr>
<tr>
<td>RS</td>
<td>The static blade of the reversible turbine under single calculation</td>
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<tr>
<td>RR</td>
<td>The rotor blade of the reversible turbine under single calculation</td>
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<tr>
<td>WS</td>
<td>The static blade of the four-stage turbine under joint calculation</td>
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<tr>
<td>WR</td>
<td>The rotor blade of the four-stage turbine under joint calculation</td>
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<td>WRS</td>
<td>The static blade of the reversible turbine under joint calculation</td>
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<tr>
<td>WRR</td>
<td>The rotor blade of the reversible turbine under joint calculation</td>
</tr>
<tr>
<td>1,2,3,4</td>
<td>The stages of turbine</td>
</tr>
<tr>
<td>2500</td>
<td>The rev is 2500 rpm</td>
</tr>
<tr>
<td>3270</td>
<td>The rev is 3270 rpm</td>
</tr>
<tr>
<td>H</td>
<td>Relative blade height [-]</td>
</tr>
<tr>
<td>α</td>
<td>Airflow outlet angle [°]</td>
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Acknowledgments
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