INVESTIGATION OF CONJUGATED HEAT TRANSFER FOR A RADIAL TURBINE WITH IMPINGEMENT COOLING

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Abstract. Radial turbine is widely used in micro-turbines, turbochargers, small jet engines and expanders, and the pursuit of high system efficiency has resulted in elevated turbine inlet temperatures for some of its applications, threatening its reliability. There are, however, few cooling studies on radial turbines. This paper studies the jet impingement cooling of a turbocharger radial turbine. A small amount of air (coolant), which could come from compressor discharge cooled by an intercooler, is injected through a few jet holes on the heat shield of the turbine onto the upper part of turbine backdisc, to cool the rotor blades and the backdisc. Parameters that may affect the cooling were studied by a Conjugated Heat Transfer (CHT) numerical simulation using steady flow calculations. The influences to the cooling effects by different coolant-to-turbine mass flow ratios, Coolant-to-turbine inlet temperature ratio, number of the jets etc. were analysed by a steady flow simulation. The simulation results show that, when four jet holes are placed at blade leading edge radius, using 1.0% ~ 3.0% of the main gas mass flow of coolant, the average temperature on leading edge, inducer hub and backdisc surface is reduced by 2K ~ 17K, 27K ~ 65K and 51K ~ 70K respectively. Turbine efficiency is mostly reduced little over 1% point.
Nomenclature

| Symbol | Description                        |
|--------|-----------------------------------|
| CHT    | Conjugated heat transfer          |
| T      | Temperature [K]                   |
| A      | Area                              |
| n      | Rotor speed [rpm]                 |
| N      | Jet hole number                   |
| r      | Jet hole radius [mm]              |
| s      | Gap of backdisc cavity [mm]       |
| D      | Diameter [mm]                     |
| g      | Tip width of rotor [mm]           |
| ṁ      | Mass flow [kg/s]                  |
| ṁco      | Coolant-to-turbine mass flow ratio |
| φave   | Area averaged cooling efficiency  |
| 0      | Total state                       |
| 1      | Inlet                             |
| 2      | Outlet                            |
| ave    | Averaged                          |
| c      | Coolant                           |
| d      | Backdisc                          |
| m      | Mainstream gas                    |
| h      | Jet holes                         |
| w,e    | The conjugate interface wall on the rotor |
| c      | Coolant                           |

Subscripts

| Symbol | Description                        |
|--------|-----------------------------------|
| 0      | Total state                       |
| 1      | Inlet                             |
| 2      | Outlet                            |
| ave    | Averaged                          |
| c      | Coolant                           |

1. Introduction

Radial turbines are widely used in micro gas turbines and turbochargers due to their high efficiency in small sizes, compact structure and high reliability. Radial turbines have high efficiency potential, higher maximum gas inlet temperature, and a bright future of micro gas turbines for nuclear emergency cooling and miniature generators [1, 2]. To pursue higher thermal efficiency and specific work, turbine inlet temperature is gradually increased. This subjects the inlet section of blades and the backdisc of turbines to great thermal stress. Xie et al. [3] found that the temperature of the rotor changes relatively smoothly, the highest temperature occurs at the inlet. And heat transfer from turbocharger turbine to compressor seriously impacts the compressor [4].

However, it’s difficult to cool the blades of radial turbines because of their small size, thin blades and cost concern. Although the cooling technology has not been applied to radial turbines in industry, it has been subjected to many studies, and some progresses made. Rodgers [5] proposed a method of film cooling by setting jet holes at the hub some distance outside the inlet of the turbine blades. The cold jet air mixed with the hot gas flow rapidly to form a film barrier. The problem with this cooling method is, however, that the cooling efficiency is relatively low. Ma et al. [6] numerically investigated the conjugated heat transfer of radial turbine backdisc cooling. A new way was innovatively proposed to cool the turbine by cooling the turbine backdisc. The results show only 0.5%~3% of main mass flow ratio may be sufficient. The disadvantage of this technology is that the turbine efficiency and expansion ratio may be slightly affected.

Although the internal cooling of the radial turbine is not easy to achieve by traditional technology, and the cost is high, some researches have still been done. R.W. Vershure et al. [7] bonded together photolithographic laminates to form a complete wheel with internal cooling channels and demonstrated the technology needed to make small cooling turbines. P.H. Snyder et al. [8] designed an advanced air-cooled metal rotor with a combination of series and parallel branch internal flow channels, carrying 4.3% of the coolant airflow to fully cool the rotor at an inlet temperature of 2500°F. Ewing et al. [9] introduced the mechanical design and process development of high temperature radial turbine, especially the internal cooling of the turbine. Steinthorsson et al. [10] developed a numerical code for calculating the three-dimensional, turbulent, compressible flow in the radial turbine blade coolant passage. Hamed et al. [11] investigated temperature distribution in a cooled radial inflow turbine rotor. Lizet et al. [12] compared the aerodynamic performance and heat transfer characteristics of a radial-inflow turbine with solid and cooled rotors.
In recent years, with the development of additive manufacture technology, manufacturing of radial turbines with complex internal cooling structure has been introduced. Arifin et al. [13] discussed the manufacturing process of radial turbine rotor using additive manufacturing method through Selective Laser Melting (SLM) machine. Zhang et al. [14, 15] used additive manufacturing technology to design an internal cooling passage within blades and hub of a radial turbine. They used conjugate heat transfer (CHT) numerical simulations and experiments and compared the turbine with the non-cooled baseline wheel. Although the development of new manufacturing technologies in recent years has made the internal cooling radial turbine possible, high manufacturing cost has limited its commercial applications. Considering the technology limitations and economic cost, for small radial gas turbines, the most practical method at this stage is still external cooling.

To cool the blades and backdisc, a small amount of compressed air from the compressor outlet, cooled by the intercooler of turbocharger, is injected through holes in the heat shield of the turbine to impinge on the blades and the backdisc of the turbine rotor with a scalloped backdisc. Scalloped backdisc is most commonly employed in the high temperature applications of radial turbines to reduce the combined thermal and centrifugal stress of the rotor. A numerical simulation of the conjugate heat transfer for the turbine has been carried out with steady flow calculations.

2. Numerical Method

2.1 Radial turbine and computation domain

We inject a small amount of air, which could come from compressor discharge uncooled or be cooled by an intercooler, through a few jet holes on the heat shield of the turbine, onto the upper part of turbine backdisc, to cool the rotor blades and the backdisc. The radial turbine of a marine turbocharger was chosen as the research object. The structural parameters of the turbine are shown in Table 1.

| Parameters (units)                  | Values | Parameters (units)                  | Values |
|------------------------------------|--------|------------------------------------|--------|
| Tip diameter \(D_1\) (mm)          | 129.5  | Outlet shroud diameter \(D_{2s}\) (mm) | 115    |
| Number of blades                   | 13     | Housing tongue Diameter \(D_t\) (mm) | 160    |
| Backdisc diameter \(D_d\) (mm)     | 93.5   | Gap of backdisc cavity \(s\) (mm)   | 0.9    |
| Inlet blade height \(g\) (mm)      | 21.8   | Jet hole inlet diameter \(D_h\) (mm) | 123.5  |
| Outlet hub diameter \(D_{2h}\) (mm) | 38.3   | Diameter of jet holes \(d\) (mm)   | 6      |

The turbine rotor has a scalloped backdisc with a tip diameter of 93.5mm. The cooling study was mainly carried out under the design condition of the turbine, that is, rotating speed of 75,200rpm, expansion ratio of 2.5, inlet total temperature \(T_{01}\) of 973K and the mainstream gas mass flow of 0.789kg/s. A CHT study was employed for this purpose. The computational domain includes the mainstream and coolant flow domains, as well as the solid metal domain. A full 360° channel of rotor flow passages with backdisc cavity and entire solid rotor are included. The computation domain of the turbine is shown in Figure 1, and the meridional channel in Figure 2.

| Parameters (units)                  | Values                  |
|------------------------------------|-------------------------|
| Coolant-to-turbine mass flow ratio \(\dot{m}_{re}\) | 0, 1.0% ~ 3.0%          |
| Jet hole number \(N\)              | 0, 2, 3, 4, 5, 6        |
| Jet hole radius \(r\) (mm, \(d = 2r\)) | 0, 2.5, 2.75, 3, 3.5, 4.25 |
| Coolant-to-turbine inlet temp. ratio \(T_{re}\) | 0.332, 0.353, 0.373, 0.383, 0.435, 0.486, 0.538 |
The location of jet holes is on the heat shield of the turbine opposite to the blades and backdisc of the turbine rotor. The low temperature, high-speed jet air coolant from the jet holes impinges on the rotor blades and backdisc surface, and conducts heat exchange with the solid rotor surface, thereby achieving the purpose of cooling. Parameters that may affect the cooling effect were studied by a steady simulation, they included the number of jet holes, jet holes position, jet angles and jet-to-turbine mass flow ratio etc. Effects of the cooling on the aerodynamic performance of turbine were also studied. Range of the investigated parameters in the study is listed in Table 2. The configuration of jet holes relative to the rotor is shown in Figure 3, and cylindric holes were chosen and distributed evenly.

2.2 Performance definitions

Key performance parameters and the formulas are shown blow:

Turbine expansion ratio, \( \pi = \frac{P_{01}}{P} \)  

Efficiency of turbine without counting cooling power consumption,  
\[
\eta_t = \frac{P_w}{mC_pT_{m0}[1 - (P_{02}/P_{w0})^{\gamma - 1}]}
\]

Where \( k \) is specific heat ratio, \( P_w \) is shaft power of turbine.

Area averaged cooling efficiency,  
\[
\Phi_{ave} = \int_0^A \Phi_{cool} dS / A
\]

Average temperature,  
\[
T_{ave} = \int_0^A T dS / A
\]

Coolant-to-turbine inlet temperature ratio, \( T_c / T_{01} \)  

Coolant-to-turbine mass flow ratio, \( \dot{m}_{cw} = \dot{m}_c / \dot{m}_m \)  

Where \( \dot{m}_c \) is the coolant mass flow ratio, and \( \dot{m}_m \) is the mainstream mass flow ratio of the turbine.
2.3 Numerical setup
Commercial software CFX 18.2 was used to solve the 3D steady Navier-Stokes equations, and Shear Stress Transport (SST) turbulence model was employed. The Frozen-rotor interface and the Sliding interface were implemented in the steady flow calculations respectively. The coolant released from jet holes may impinge on the target surface first, but always mixes with the mainstream gas. Therefore, there is only one outlet of the whole fluid domain where a static pressure was given. At the mainstream flow inlet, total pressure and total temperature as well as two flow directions were specified. At the coolant inlet, coolant mass flow, flow directions and total temperature were specified.

The conjugate interface boundary conditions were imposed on all the fluid-solid interfaces of the rotor. The boundary conditions of the other solid walls were defined as non-slip, smooth and adiabatic. The coolant was regarded as an ideal gas, and its temperature varied from 50°C to 250°C.

2.4 Mesh and mesh independence
Due to the complex model structure, the hexahedral meshes for the fluid domain with 10 boundary-type of layers and solid domain were generated by the commercial software ICEM CFD. The meshes for the jet hole, the backdisc of rotor and the backdisc cavity were densified to better solve the flow and heat transfer. The rotor meshes of fluid domain and solid domain are shown in Figure 4.

![Meshes of jet holes, cavity and rotor](image1)
![Solid rotor mesh](image2)
![Jet hole mesh](image3)

Fig. 4 Mesh for the CHT model with 4 jet holes

The grid independence was first verified with four jet holes. Shown in Figure 5, the average cooling efficiency of the rotor and turbine efficiency are compared for five different meshes with 10.5 million, 16.1 million, 19.2 million, 23.2 million, 26.5 million and 28.8 million grid numbers respectively. The average cooling efficiency changes little when the mesh density reaches and exceeds 23.2 million, so this mesh density was selected.

![Graph of calculated average cooling efficiency and turbine efficiency](image4)

Fig. 5 Calculated $\phi_{ave}$ on the surface of solid rotor and $\eta$ with different mesh grids
3. Results and discussion

3.1 Effects of number of jet holes

Different numbers of jet holes were compared for the metal temperature of the rotor surfaces under the same total coolant mass flow ratio and same total jet hole area at the design condition of the turbine (rotating speed \( n = 75,200 \text{rpm} \), expansion ratio \( \pi = 2.5 \)). The relative coolant mass flow \( \dot{m}_c/\dot{m}_m \) was fixed at 3.0%, and relative coolant temperature \( T_c/T_01 \) equals to 323.15K/973.15K = 0.332. The jet hole was centered at \( D_h = 123.5 \text{mm} \), giving a dimensionless jet hole location \( D_h/D_1 \) of 0.954.

Figure 6 compares the temperature distribution at the rotor surface without and with cooling. Different numbers of jet holes are also compared. Without jet cooling, the circumferentially uneven temperature distribution is caused by the uneven volute discharge. For blades, the maximum temperature of 900K is located on the blade inducer hub and the leading edge of blades. The minimum temperature of the backdisc is above 867K, and the temperature of backdisc tip region is high. It can be seen that the blade inlet section and the edge of the backdisc are the places that need cooling most. Due to the gas expansion, the temperature of rotor continues to drop along the flow direction, and the lowest temperature is about 829K recorded at the turbine outlet.

![Figure 6](image)

When the cooling is introduced, the temperature of the blades and the rotor backdisc near jet holes has decreased significantly. The minimum blade temperature and the backdisc near the jet hole is below 800K. The low temperature zones of the rotor blades and the backdisc are not at the positions directly facing the jet holes, but at the downstream positions of the holes (the rotor rotates clockwise in the figure). This is mainly due to the spreading effect of the rotor rotation. In addition, the number of low temperature zones on the solid rotor wall is equal to the number of jet holes. When the number of jet holes is increased from 2 to 6, the temperature distribution on the solid rotor surface is more even, and the area of the low temperature zone is larger, indicating that the more jet holes can improve the cooling and temperature distribution, thereby reduce the stress caused by temperature difference.
However, one disadvantage of more jet holes is that the leading edge of the blades cannot be effectively cooled. This is further illustrated in Figure 7. The reason for this is that when the total area of jet holes and the total coolant mass flow are fixed, the more jet holes are, the smaller diameter of each jet hole and the less coolant mass flow ratio per hole will be, and the kinetic energy carried by the coolant per hole will be smaller, so the coolant is less able to penetrate the mainstream flow and to cool the leading edge. This problem may be solved by opening some jet holes on the housing shroud wall near the leading edge shroud as well.

![Fig. 7 Average temperature of each blade leading edge at design condition (circles are the position of jet holes, housing tongue is located at 0/360°.)](image1)

![Fig. 8 Average temperature of each blade inducer hub at design condition (circles are the position of jet holes.)](image2)

It can be seen from Figure 8 that the temperature of the blade inducer hub (the hub exposed by the backdisc scalloping) near the jet holes is significantly reduced, and the cooling effect on the blade inducer hub is better than that on the leading edge. The main reason for the better cooling of the inducer hub is that the hub is closer to the jet holes than the leading edge. The lowest temperature of the blade leading edge and the blade induce hub both appear near 360°/0° (location of the volute tongue). This is because that at the design condition, the static pressure is lowest approaching the volute tongue, hence the coolant can penetrate deeper, and produce the best cooling effect.

![Fig. 9 Effects of the number of cooling holes on average temperature and average cooling efficiency at design condition. (Here Rotor refers to the averaged results of entire rotor including leading edge, backdisc etc.; Blades refers to the suction and pressure surfaces of the blades, excluding the leading edge, inducer hub and shroud of the blades.)](image3)
Figure 9(a) shows that the average temperature of the turbine wheel can be significantly reduced with impingement cooling, but the temperature reduction of each part of the wheel varies with the number of jet holes. The average temperature of blade leading edge is less affected by the number of the jet holes, with 2 jet holes, the temperature of the blade leading edge is reduced by 20K, and with 6 jet holes by 12K. For other parts of the wheel, cooling effect in general increases with the number of the jet holes. The jet cooling has the best cooling effect on the backdisc. Average cooling efficiency in Figure 9(b) confirms the findings from average temperature.

![Figure 9(b)](image)

Fig. 10 Effects of jet holes number on turbine efficiency at design condition

The effects of the jet cooling and the number of the jet holes on turbine efficiency are shown in Figure 10. The impingement cooling has a negative impact on the efficiency. This impact reduces with the increment of the jet holes. The coolant increases the weight of the turbine wheel and consumes some of the rotating torque of the wheel when it impinges on to the wheel. The cold coolant also mixes with the hot gas in the mainstream flow passages, generating mixing losses. Increasing the number of jet holes makes the impingement gentler and more evenly spread, and less disturb the mainstream gas in the rotor, therefore reduces the losses generated by the coolant.

3.2 Effects of jet coolant relative mass flow ratio

The effects of coolant-to-mainstream mass flow ratio on temperature of turbine solid rotor are given in Figure 11. The ratio ranges from 1.0% ~ 3.0%, and Jet hole position is set at $D_h/D_i = 0.954$. The temperature of blade leading edge decreases in a convexly type curve with the increases of coolant mass flow ratio. More cooling air than 3% can result better cooling of the leading edge. On the other hand, the temperature of the blade inducer hub and others decreases in a concavely fashion. As the mass flow ratio of the cooling air increases, the average temperature of the blade inducer hub decreases the most, followed by the backdisc, and the blade surfaces, the average temperature on the blade leading edge decreases the least.

The relationship between the relative coolant mass flow ratio and turbine performance, expansion ratio is shown in Figure 12. Figure 12 (a), Turbine efficiency decreases with the increase of the relative coolant mass flow ratio. With 6 jets, the efficiency penalty is about 1% point at 3% relative coolant flow. For this turbine rotor with a scalloped backdisc, the impingement cooling has little effect on the turbine expansion ratio, in Figure 12 (b). The maximum change happens with the 2 jet holes, it is less than 0.5%.
3.3 Effects of jet coolant temperature and operating temperature

In Figures 13 the effects of jet coolant inlet temperature on the average temperature of the solid rotor are shown. At the turbine design condition and 3% relative mass flow of coolant, the temperature of different parts of the solid rotor surface increases with the coolant temperature, except the blade leading edge where the metal temperature remains constant when the relative coolant temperature varies between 0.332 \sim 0.538 (50^\circ C \sim 250^\circ C). So, considering the associated cost, reducing coolant temperature is not the ideal way to enhance the cooling effect of the blade leading edge.
Figure 14 shows the effects of jet coolant inlet temperature and turbine inlet temperature on the average temperature of the solid rotor. The temperature of every part of the solid rotor increases linearly with the turbine inlet temperature from 873k to 1173k (600°C to 900°C). Note that the temperature of the blade leading edge has only increased by 245k while the inlet gas temperature raises by 300k, that is, the cooling of the leading edge becomes more effective with the increment of turbine inlet temperature. At 1173k inlet temperature, the leading edge temperature is more than 140k lower than the gas inlet temperature, and this offers an effective protection to the leading edge.

4. Conclusions
A steady CHT simulation was carried out to study the effects of impingement cooling of the rotor of a turbocharger radial turbine. Four cooling jets were installed on the heat shield of the turbine and impinged through the backdisc-heat shield cavity on to the rotor.

And found that
1. The jet cooling has a good effect on the cooling of the scalloped backdisc rotor. Leading-edge shroud is least cooled while significant cooling can be achieved for inducer hub and backdisc of the rotor.
2. The more jet holes, the more even the distribution of the low temperature area on the rotor surface. However, the cooling effect on the leading-edge shroud area is worse.
3. As the coolant-to-turbine mass flow ratio increases, the blade inducer hub temperature decreases the most, followed by the backdisc, and the leading edge decreases the least.
4. As coolant-to-turbine inlet temperature ratio decreases, the average temperature on the rotor surface decreases. Coolant-to-turbine inlet temperature ratio is decreased from 0.538 to 0.332, with the four-hole jet, the average temperature of the leading edge, inducer hub and backdisc is reduced by about 7K, 35K and 33K, respectively.
5. Impingement cooling has a small negative impact on efficiency an expansion ratio of the turbine.

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