Influence of the Amplitude of Inlet Axial Pulsating Flow on Turbine Working Characteristics

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Abstract. The flow field at the outlet of the rotating detonation combustor has strong pulsation characteristics, and this strong impact will bring fatal damage to the downstream turbine. In this paper, the simplified sine wave model was used to simulate the rotating detonation wave to study the characteristics of the turbine working in a pulsating flow field. The results show that the greater the amplitude of the incoming flow pulsation, the stronger the non-uniformity of the aerodynamic parameters and heat transfer parameters of the flow field inside the turbine. The increase in the amplitude of the incoming flow pulsation will aggravate the separation of the end wall secondary flow and the cascade flow, resulting in greater flow loss and reducing the working efficiency of the turbine.

1. Introduction
Rotating detonation combustor was originally proposed and investigated by Voitsekhovskii, who selected a circular combustor, the fuel and oxidant were injected into the closed section of the combustor. The rotational detonation wave with a certain tangential velocity was generated by detonation combustion and propagated downstream. Rotating detonation combustor has significant advantages in high thermodynamic efficiency, fast energy conversion, self-pressure gain and low entropy etc[1,2]. Therefore, it garnered plenty of attention in recent years.

Presently, a great deal of current researches of rotating detonation engines mainly focus on combustor itself. Bykovskii[3], Frolov[4], Russo[5] and Thomas[6] et al. explored the generation and propagation mechanism of rotating detonation waves under different conditions. Rankin[7] analyzed the thrust performance of the rotating detonation engines. Eto[9], Jourdaine[9] and Yi[10] probed into the influence of nozzles on engine flow field characteristics by installing different geometric sizes nozzles at combustor downstream.

When the rotating detonation combustor is applied to a gas turbine engine, the generated pressure gain can operate a large effective thrust under a low pressure ratio, which will reduce the number of engine compressor stages and the manufacturing requirements of the gas turbine engine. The engine performance will be dramatically improved due to the simple and compact combustor and higher thrust-weight ratio. Therefore, scholars have carried out studies of the matching mode between the combustor and turbine.

In summary, few researches are focusing on the matching mechanism of rotating detonation combustor and turbine with its influence mechanism currently, and lacking for evaluation on the unsteady pulsation flow field and aerodynamic heat transfer performance. Therefore, according to the periodic pulsation characteristics of total pressure and total temperature at the outlet of rotating detonation combustor, this paper simplified it to sinusoidal wave with periodic fluctuation of specific
amplitude and frequency. The influence mechanism of the pulsating flow which from detonation combustor on turbine performance was studied through unsteady numerical simulations.

2. Calculation model and numerical method
NUMECA Autogrid-5 software was adopted to generate grids. The computational domain and local grids are shown in Fig.1. Firstly, the steady numerical calculation was conducted for the model, and the result was regarded as the initial field of unsteady computation. In the stationary calculation, the turbine's total inlet pressure was set as 344.74kPa, the total inlet temperature was 709K, the average outlet stator pressure was 150kPa, the rotor speed was 8,280r/min, and the inlet turbulence intensity was 10%. The turbulence model was SST γ-θ model, the wall surface was set as adiabatic without slip boundary condition, and the interface was selected as the Frozen Rotor mode.

In order to verify the influence of the grid nodes number on the numerical calculation accuracy, the grid independence verification of three sets grids was implemented through steady numerical calculation. The calculation results are shown in Tab.1. By comparison, from grid 1 to grid 2, the isentropic efficiency was changed by 0.26% and the outlet flow was varied by 0.09%. From grid 2 to grid 3, the isentropic efficiency was changed by 0.13%, and the outlet mass flow was varied by 0.04%, which means that the influence of grid nodes is relatively little. Therefore, grid 2 was selected for numerical study.

\[
p_{0i} = k \times p_{0m} \times \sin(2\pi f) + p_{0m} \\
T_{0i} = 2k/3 \times T_{0m} \sin(2\pi f) + T_{0m}
\]

Based on the outflow characteristics of typical rotating detonation combustor, the fluctuation rules of total pressure and temperature at turbine inlet were set in unsteady calculation. The fluctuation rule of inlet total pressure \( p_{0i} \) was donated by Equation(1), and the fluctuation rule of inlet total temperature \( T_{0i} \) was given by Equation(2), where the inlet average total pressure \( p_{0m} \) was 344.74kPa, and the average total temperature \( T_{0m} \) at the inlet was 709K. \( k \) is the pulsation coefficient of the inlet parameter, which was defined as the ratio of the inlet parameter fluctuation amplitude and the average parameter, which reflects the pulsation amplitude of the flow field, and \( f \) is the pulsation frequency of the inlet parameter. The Transient Rotor-Stator model was chosen for data transmission at the interface between stator and rotor. The time for a rotor blade to rotate a stator vane's pitch was defined as a computing cycle \( T_{com} \). The physical time step of each calculation cycle has a significant influence on the calculation results. Through the verification of unsteady calculation, 48 physical time steps were set for each calculation cycle, and the duration of each physical time step was 3.97e-6s.

3. Results and Discussions
Aiming to investigating the effects of fluctuating amplitude and frequency of inlet pressure and temperature on turbine working characteristics, 5 working conditions were set according to the different inlet conditions, as shown in Tab.2. Due to the differences of fuel composition, equivalent ratio and combustor geometry structure used in detonation combustion, the amplitude of the rotating detonation wave varies, and the propagation frequency may occur in the range of 1~100kHz[7]. The pulsation coefficient selected in this paper is 0~0.4, and the frequency is 5.244Hz.
In order to reveal the migration and developing process of pulsation flows in turbine, Fig.2 shows the distribution of instantaneous total pressure $p_0$ at the 50% turbine span under working case 3 within a flow cycle $T$.

In Fig.2, at the moment of $T/8$, the rotor blade B2 was located in the central of the stator path downstream. At this time, the pressure crest of the inlet pulsating flow was transferred to the leading edge of the rotor blade, and a high pressure area was formed at the leading edge point of B2. At the moment of $T/2$, B2 moved to the downstream of the trailing edge of the stator vane. However, due to the rotating effect of the rotor, there is a flow velocity component along the rotating direction in the flow field at the rotor-stator interface. The high pressure area in the flow field moved to the leading edge of B2, so that the leading edge of B2 was still in a high pressure area. At the same time, B1 moved to the central of stator path downstream, since the flow field at the outlet of the stator path was still a low pressure value at the moment, thus B1 was still in the low pressure area. At the moment of $T$, B2 moved to the center downstream of the next adjacent stator path. Meanwhile, the pulsation pressure in the next cycle reached the maximum at the throat of the stator path, and a high pressure area was composed at the leading edge of B2. Then the pressure crest of the flow field pushed downstream, and the rotor blades continued to shift along the steering direction and formed a new cycle.

Under the condition that the pulsation frequency of the inlet flow field is equal to the rotation frequency of the rotor blade, the initial position of B1 is in the mixing zone of the vane wake. Therefore, B1 is always in the lower pressure zone under the influence of the pressure trough during the entire motion cycle. While B2 is affected by the pressure peak at the initial moment, and is subjected to higher pressure shocks throughout the exercise cycle.

Fig.3(a) shows the time averaged pressure loading of two adjacent rotor blades B1 and B2 at the 50% span in Case 0. As for the inlet condition of the turbine under this working condition is a uniform flow field without pulsation, the load distribution of the two rotor blades of the turbine is consistent.

When the pulsation frequency of the inlet flow is fixed ($f=5,244Hz$), the averaged load of the two rotor blades is significantly different due to the interference between the rotor blades and the periodic pulsating airflow. With the increase of pulsation amplitude, the time-averaged pressure load at the height of 50% of rotor blade B1 declines gradually, while the time-averaged load at the adjacent rotor blade B2 increases progressively. Knowing from Fig.3(b) and Fig.3(c) that the greater the inlet
pressure pulsation amplitude is, the larger the time-averaged pressure load difference between the two adjacent rotor blades will be. This indicates that different rotor blades will bear different pressure loads at a certain pulsating incoming flow frequency.

Fig. 3 Effects of pulsation amplitude on time-average pressure loading of blade at 50% span

Fig. 4 compares the influence of inflow pulsation coefficient on the time-averaged heat transfer coefficient distribution around the upper and lower end-wall of the stator vane is basically consistent because of the symmetric flow state of the upper and lower end-wall. In general, the time-averaged heat transfer coefficient at the height of 50% stator vanes is more easily affected by the inflow pulsation coefficient, and the pressure surface changes more obviously than the suction surface. With the increase of the inflow pulsation coefficient, greater flow separation would happen on the pressure surface in the middle vane, therefore the time-averaged heat transfer coefficient of the vane wall would steeply decrease. The flow separation on the stator vane surface will be explained in the following part.

Fig. 5 contrasts the influence of the inflow pulsation coefficient on the time-averaged heat transfer coefficient of rotor blade B2 at different blade wall heights. In conjunction with Fig. 3(c), it can be seen that the greater the inflow pulsation coefficient of the incoming flow, the greater the load on the blade surface. It can be clearly seen from the pressure surface side of the rotor blade B2 that the corresponding time-averaged heat transfer coefficient will increase with the heat load. On the suction side of the rotor blade, the increase of the coefficient leads the turning point position on the blade surface to move forward.
Fluctuating characteristics of flow field cause uneven distribution of stator end-wall temperature. Fig. 6 shows the variation of the maximum time-averaged temperature $T_{\text{max}}^{ia}$ and the area-mean time-averaged temperature $T_{\text{aa}}^{ia}$ at the end-wall of stator under different inflow pulsation coefficients. It showed that the uneven distribution of end-wall temperature is affected significantly by the fluctuation coefficient of inlet flow. Although the average total temperature at the turbine inlet is the same under different working conditions, as the wall temperature is less sensitive to time changes than pressure, the area-mean time-averaged temperature and the maximum time-averaged temperature in the high-temperature region both increase following the increment of the inflow pulsation coefficient. According to the variation trend of the end-wall nonuniform coefficient $r_{T} = T_{\text{max}}^{ia} / T_{\text{aa}}^{ia}$, the growth of the inflow pulsation coefficient will aggravate the temperature inhomogeneity of the end-wall and cause the local area bear larger heat load.

Fig. 6 Effects of inflow pulsation coefficient on the Time-averaged temperature of rotor end-wall

Fig. 7 Effects of pulsation amplitude on turbine efficiency

Fig. 7 provides the variation curve of turbine efficiency following with inflow pulsation amplitude. It is observed that turbine efficiency decreases with the increase of pulsation amplitude. When the inflow pulsation frequency is 5,244Hz and the pulsation coefficient of the inlet flow field grows from 0 to 0.4, the turbine efficiency drops from 0.8898 to 0.7653, reducing by 13.99%.
Under the condition of incoming flow pulsation frequency of 5,244Hz, due to the existence of local adverse pressure gradient, the flow separation of pressure side and suction side appeared successively, the extent and range of the flow separation increase with the increase of flow pulsation amplitude, which causes increment of flow losses in turbine. At the same time, pulsating inflow may lead the relative flow angle of turbine rotor blade inlet to deviate from the designed value, and generate positive/negative attack angle at the leading edge of rotor blade, further increases flow losses.

4. Conclusions
Through numerical simulation, the influences of inlet flow pulsation amplitude and frequency on turbine operating characteristics are investigated, and the following conclusions are obtained:
1). The fluctuation of turbine flow field, the increment of rotor thermal load, and the turbine working efficiency reduction will be aggravated by the increment of the inlet flow pulsation amplitude. Premise of the fixed inlet flow pulsation frequency at 5,244Hz, when the pulsation coefficient is increased to 0.4, the turbine efficiency is declined by 13.99%. The aerodynamic performance differences between different blades will be more obvious with the increase of inflow pulsation amplitude.
2). Uneven pressure distribution in stator flow path will be exacerbated by the increase of the inflow pulsation amplitude and frequency, which will produce a larger secondary flow loss. In addition, it will escalate the flow separation of stator and rotor blade surface, and make the rotor inlet attack angle deviate from the designed state, which is the main reason for the turbine efficiency reduction.

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