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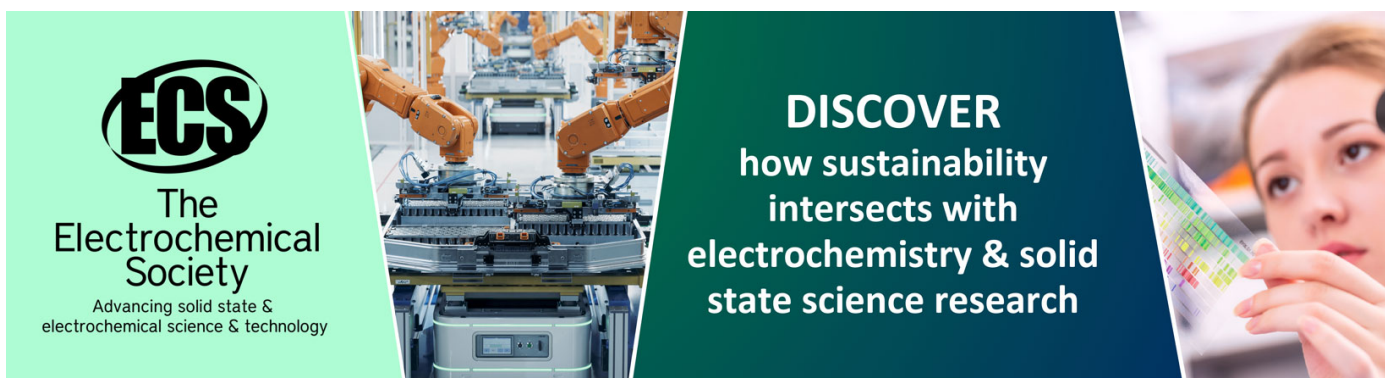
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To cite this article: Bing Ji *et al* 2021 *J. Phys.: Conf. Ser.* **1750** 012070

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Analysis of Unsteady Flow Field in Rotating Detonation Turbine Engine

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Abstract. The rotating detonation turbine engine is a research hotspot in recent years, but the research is mainly at the phase of theoretical analysis. In this paper, the influence of the circumferentially rotating pulsating flow field on the performance of the turbine is studied by numerical simulation. The results show that the uneven turbine inlet flow will increase the working load of the stator vane, but will produce greater flow loss. At the same time, the strength of the channel vortex and the tip leakage vortex inside the rotor is also increasing. The mass flow rate and work efficiency of the turbine decrease with the increase of the unevenness of the flow field.

1. Introduction

Rotating detonation is another mode that exists in the combustion process. It has significant advantages such as high thermodynamic efficiency, fast energy conversion, self-pressure gain, and small entropy. The concept of rotating detonation combustion was first proposed by Voitsekhovskii[1]. In recent years, the energy efficiency of the rotating detonation combustion and the propagation mechanism of detonation wave have been widely studied through experimental tests and numerical calculations [2, 3]. However, there are relatively few studies on the combination of rotating detonation combustor and turbine. Only simple experiments and numerical simulations are done[4, 5], and there are few studies on the flow mechanism inside the turbine under complex conditions. This paper studies the influence of the rotating pulsating flow field on the flow characteristics of the turbine stator, and analyzes the flow loss mechanism.

2. Materials and methods

2.1. Calculation model

The geometric model used in the numerical calculation in this paper is the first stage of the GE-E3 engine high-pressure turbine, due to the limitation of computing resources, this paper only studies 4 channels among them. The calculation model was meshed with NUMECA AutoGrid-5 software, and the grid independence was verified before the formal calculation. The details of computational domain and grid division are shown in Figure 1, and the total number of grid nodes is 5 million.



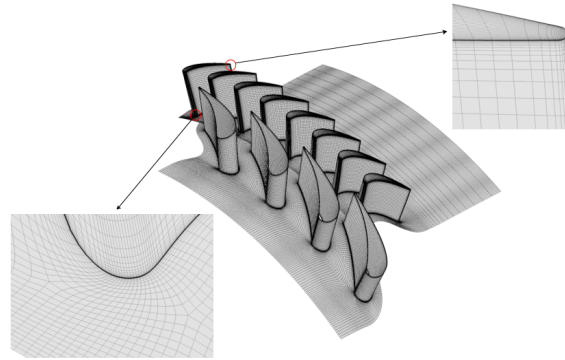


Figure 1. Computational domain and grid of turbine

2.2. Numerical Methods

In the unsteady calculation, based on the uneven circumferential rotation distribution characteristics of the combustion chamber outlet flow field, the total pressure and total temperature conditions at the turbine inlet are simplified to a sawtooth wave distributed in the circumferential direction and rotating at a specific linear velocity. Figure 2 shows the circumferential distribution of the turbine inlet conditions at the initial moment in the unsteady calculation. In the figure, P_{0m} is 2068kPa, and T_{0m} is 2100K. $k_{\theta i}$ expresses the circumferential pulsation coefficient of the import parameter, which is defined as the ratio of the fluctuation amplitude of the import parameter to the average parameter, reflecting the magnitude of the pulsation amplitude of the flow field. The circumferential rotational linear velocity of the inlet flow field $v_{\theta i}$ is 1500m/s.

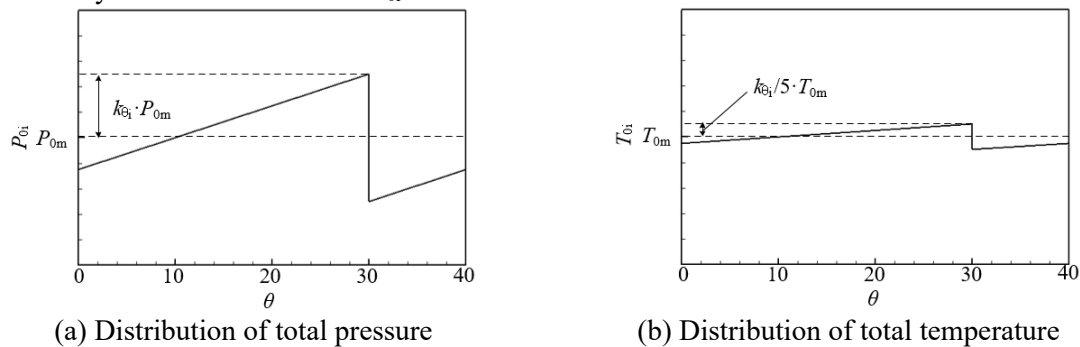


Figure 2. Turbine inlet boundary conditions at the initial moment

Under the same condition that the circumferential rotational linear velocity $v_{\theta i}$ of the pulsating incoming flow at the inlet is the same, four working conditions from a-1 to a-4 are calculated to study the influence of the uneven circumferential distribution of the inlet flow field on the turbine operating characteristics.

Table 1. Turbine inlet boundary conditions under different working conditions

Case number	$k_{\theta i}$	$v_{\theta i} / \text{m} \cdot \text{s}^{-1}$
a-1	0.1	1500
a-2	0.2	1500
a-3	0.3	1500
a-4	0.4	1500

3. Results and discussions

Figure 3 compares the same instantaneous pressure distribution at the 50% blade height section of the turbine under different circumferential pulsation amplitudes. It can be concluded that the larger the circumferential pulsation coefficient $k_{\theta i}$ of the incoming flow, the stronger the unevenness of the flow field in the turbine channel. Especially in the stator domain, the oblique pressure peak formed by the

rotating pulsating wave will form an obvious high pressure area after reaching the pressure surface of the stator blade, so there will be a stronger secondary flow in the cascade channel.

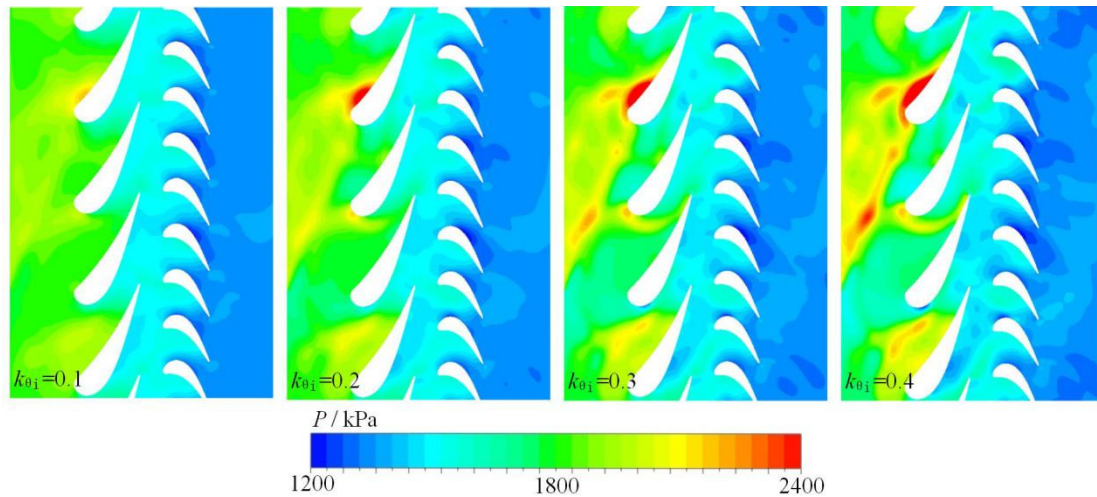
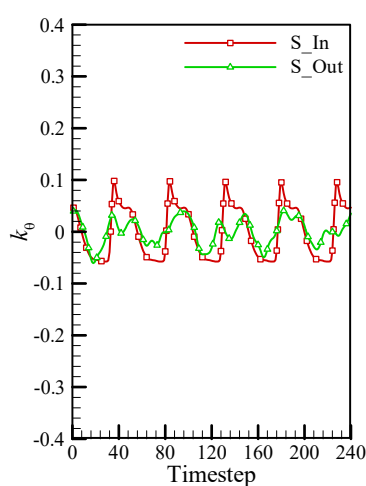
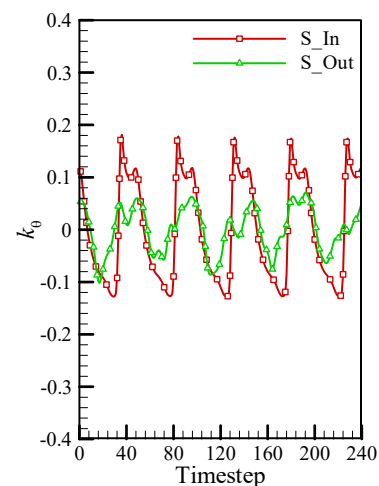


Figure 3. Instantaneous pressure distribution at 50% height of turbine

By comparing the pulsation of the pressure at the inlet and outlet of the stator channel in one cycle, the change characteristics of the pulsating flow field in the turbine channel can be quantitatively analyzed. As shown in Figure 4, under the currently set calculation conditions, the rotation speed of the turbine inlet rotating pulsation wave is about 5 times the turbine rotor speed, so there will be 5 pressure values in one calculation cycle (240 calculation time steps). In the figure, the ordinate k_θ represents the local pressure pulsation coefficient of the monitoring point. It can be seen from the figure that compared with the pulsation coefficient setting value $k_{\theta i}$ of the turbine inlet, the pulsation coefficient k_θ of the flow field at the inlet of the stator channel is slightly reduced but basically the same. After passing through the stator channel, the pulsation characteristics of the flow field are suppressed.



(a) $k_{\theta i}=0.1$



(b) $k_{\theta i}=0.2$

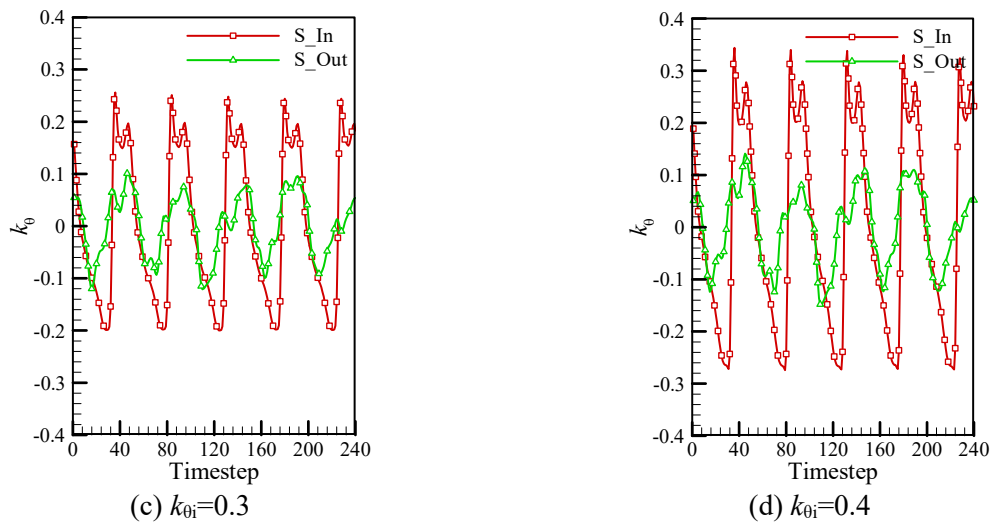


Figure 4. Distribution of pressure fluctuation coefficient at the monitoring point of the inlet/outlet section of the stator passage in a single cycle

To a certain extent, the attenuation of pressure pulsation is also accompanied by the total pressure loss of the flow field. Figure 5 shows the change trend of the average total pressure recovery coefficient at the exit section of the stator blade with the circumferential pulsation coefficient of the turbine inlet pressure. Since the increase of the circumferential pulsation coefficient of the inlet flow will strengthen the non-uniformity of the flow field of the turbine vane channel, the increase of the reverse secondary flow in the channel will aggravate the flow loss. Therefore, as the inlet circumferential pressure pulsation coefficient increases, the time-averaged total pressure recovery coefficient at the outlet of the turbine vane decreases slightly.

Figure 6 shows the distribution law of the isentropic efficiency of the stator exit section along the span. On the whole, the isentropic efficiency at the exit of the stator has a significant drop near 20% leaf height and 60% leaf height. This is because the channel vortex originating from the leading edge of the vane is constantly rolled up to the center of the channel under the influence of the secondary flow of the end wall in the cascade channel. After development, when the channel vortex reaches the vicinity of the trailing edge, a large separation zone has been created in the span direction. With the increase of the circumferential pulsation coefficient of the flow from the turbine inlet, the secondary flow in the stator channel and the channel vortex are enhanced to a certain extent, so more flow loss will be generated. Therefore, the isentropic efficiency of the stator outlet distributed along the span direction will gradually decrease with the increase of the circumferential pulsation coefficient of the incoming flow.

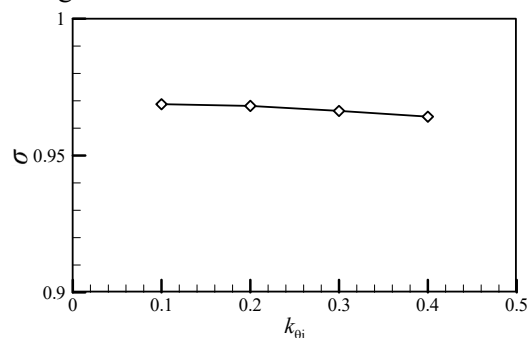


Figure 5. Change of the time-averaged total pressure recovery coefficient at the stator outlet

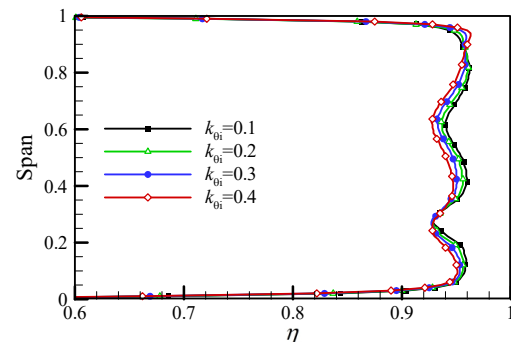


Figure 6. Distribution of the time averaged efficiency of stator exit along the span direction

Figure 7 compares the variation of the time-averaged pressure load distribution on the surface of anew with the circumferential pulsation coefficient of the incoming flow at different span positions. The analysis shows that the difference in the influence of the flow field under different circumferential pulsation coefficients on the stator blade load is mainly reflected in the area of the front 50% chord length of the vane. With the increase of the circumferential pulsation coefficient of the incoming flow, the time-averaged load on the pressure surface and suction surface of the vane increases. The main reason for the increase of the average load on the pressure surface is as follows. The rotating pulsating wave of the inlet flow will form an oblique high-pressure fluid group in the upstream passage of the vane. After the high-pressure fluid group develops to the leading edge of the vane, a high-pressure zone will be formed on the pressure surface side of the leading edge of the vane. The greater the circumferential pulsation coefficient of the incoming flow, the higher the pressure peak. Therefore, the time-averaged pressure in the high-pressure area of the pressure surface side flow field near the front edge of the vane will be greater, and the corresponding time-averaged load in the front 50% chord area of vane surface will be higher. The high-pressure fluid group on the pressure surface side of the vane will produce a reverse migration opposite to the swirling direction of the turbine inlet under the action of the reverse pressure gradient in the cascade channel. Therefore, after the high-pressure fluid group in the channel with increased pressure adheres to the suction surface side of the vane, it will naturally cause an increase in the average load on the suction surface.

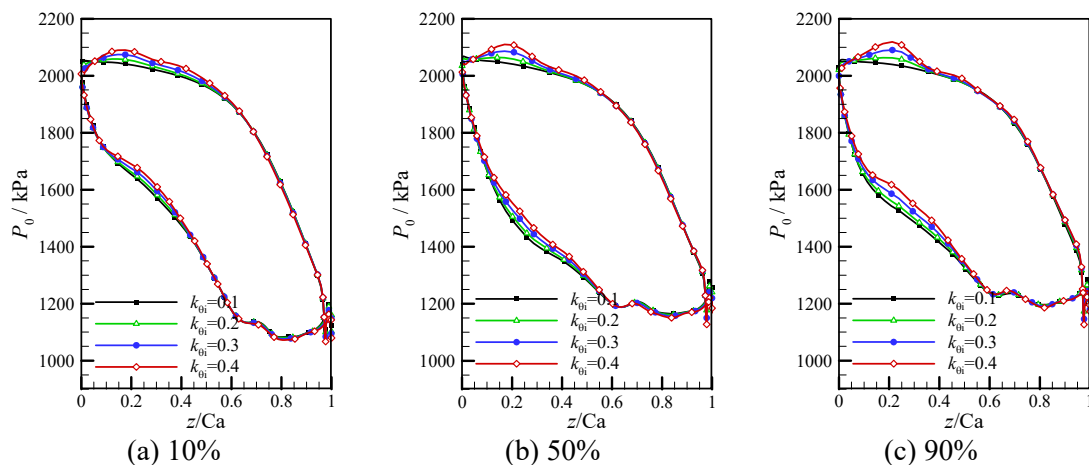


Figure 7. Time-averaged load distribution at different spanwise positions on the surface of the vane

Figure 8 compares the time-average entropy increase distribution on the exit section of the turbine rotor under different inlet flow conditions. It can be seen that the entropy increase of the turbine exit section basically originates from the rotor blade wake, the lower end wall channel vortex and the rotor blade tip leakage vortex. With the increase of the circumferential pulsation coefficient of the flow from the turbine inlet, the strength of the channel vortex and the tip leakage vortex is continuously enhanced, and spreads along both sides in the circumferential direction. This shows that more flow loss is generated in the bucket channel, and the leakage loss of the blade tip accounts for a large part.

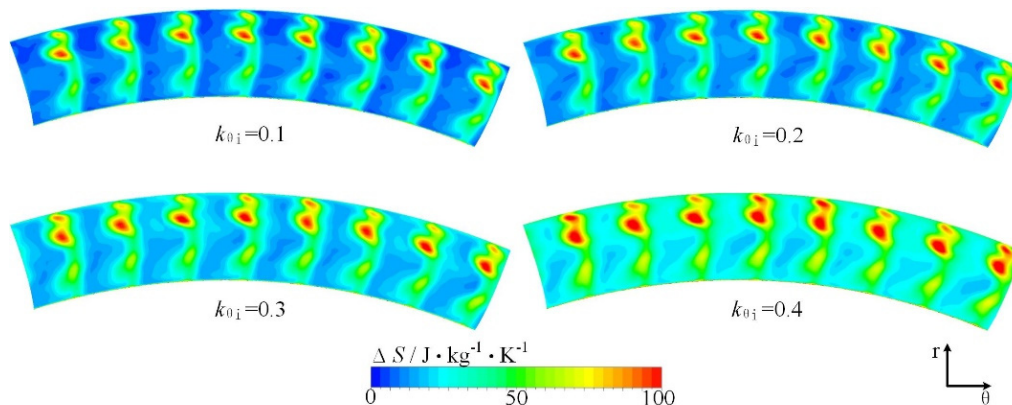


Figure 8. The time-average entropy increase distribution on the exit of rotor

It can be seen from Figure 9 and Figure 10 that with the increase of the circumferential pulsation coefficient of the inlet flow, the dimensionless converted mass flow rate and efficiency of the turbine show a downward trend. Compared with the turbine working under uniform inlet flow conditions, when the pulsation condition of inlet flow is $k_{\theta i}=0.4$, $v_{\theta i}=1500\text{m/s}$, the mass flow of the turbine is slightly reduced (-2%). Affected by the secondary flow in the stator blade channel and the increase of the leakage flow of the rotor blade tip, the efficiency has decreased significantly (-11.75%).

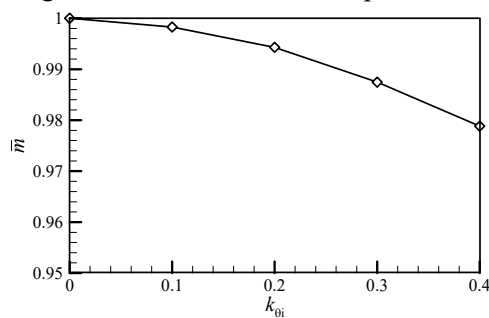


Figure 9. Change of the dimensionless converted mass flow rate

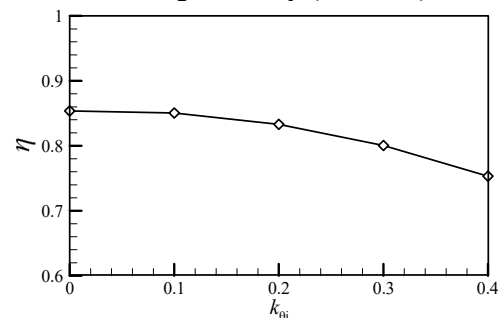


Figure 10. Change of the turbine efficiency

4. Conclusions

Through the numerical simulation, the flow mechanism inside the turbine during the operation of the rotating detonation turbine engine is studied.

- 1) The larger the value of the circumferential pulsation coefficient of the flow from the turbine inlet, the more obvious the pulsation attenuation of the flow field after passing through the vane channel.
- 2) Uneven flow from the turbine inlet will increase the working load of the vane, but at the same time it will produce greater flow loss.
- 3) With the increase of the circumferential pulsation coefficient of the incoming flow from the turbine inlet, the strength of the channel vortex and the tip leakage vortex inside the rotor continue to increase.
- 4) The mass flow rate and work efficiency of the turbine decrease with the increase of the unevenness of the flow field.

Acknowledgments

This paper is supported by Natural Science Basic Research Program of Shaanxi (Program No. 2020JQ-473).

References

- [1] Voitsekhovskii, B.V. (1959) Statsionarnaya dyetonatsiya. J. Doklady Akademii Nauk SSSR., 129: 1254-1256.
- [2] Fotia, M.L., Hoke, J., Schauer, F. (2017) Study of the Ignition Process in a Laboratory Scale Rotating Detonation Engine. J. Experimental Thermal & Fluid Science., S113985359.
- [3] Frolov, S.M., Aksenov, V.S., Gusev, P.A. (2015) Experimental studies of small samples bench rocket engine with a continuously-detonation combustors. J. Goren. Vzryv (Mosk.); Combustion and Explosion., 8: 151-163.
- [4] Ji, B., Li, J., Wang, Z.D. (2020) Effect mechanism of pulsating flow on unsteady aerodynamic performance of turbine. J. Journal of Aerospace Power., 35: 410-421.(In Chinese)
- [5] Wolański, PIOTR. (2015) Application of the Continuous Rotating Detonation to Gas Turbine. J. Applied Mechanics & Materials., 782: 3-12.