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Numerical investigation of flow and heat transfer characteristics in radial pre-swirl system with different pre-swirl nozzle angles

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ABSTRACT

The effect of pre-swirl nozzle angle (θ) on the flow and heat transfer characteristics of a radial pre-swirl system is numerically investigated using the CFD software ANSYS-CFX. In this paper, six pre-swirl nozzle angles are selected to study the flow dynamics of the radial pre-swirl system in terms of the flow structure, the nozzle exit flow angle, the air swirl ratio at nozzles outlet and receiver holes inlet, the total pressure loss coefficient in the nozzle and the cavity, the discharge coefficient of the nozzles and receiver holes and the adiabatic effectiveness. It is shown that the case of $\theta = 20^\circ$ exhibits the best performance in adiabatic effectiveness and total pressure loss in this system, while the case of $\theta = 60^\circ$ provides a maximum in discharge coefficient of receiver holes that is slightly higher than that of $\theta = 20^\circ$. Comparatively, the case of $\theta = 20^\circ$ provides the best performance of the radial pre-swirl system. In addition, the effect of pre-swirl nozzle angle on the heat transfer characteristics is also presented. Results show that the heat transfer on the rotor disk is dominated by the flow structure and tangential velocity differential, and the peak in heat transfer coefficient can be observed near the receiver hole.

1. Introduction

In order to manufacture a gas turbine with high efficiency, power output and low fuel consumption, it is necessary that the temperature at the entry to the turbine is as high as possible. However, this temperature rise is limited by the melting point of many component materials. Consequently, to surpass this ceiling it is necessary to provide cooling air to cool the high temperature components and prevent this material from failure. Note that feeds are bled from different stages of the compressor and guided through the secondary air system to provide high and low pressure air to different stages of the turbine. This air, however essential, is a source of work loss to the system, and thus it is important to minimize the volume of cooling air. It is recognized that swirling air can reduce the work must be done on the flow by the rotating disk accelerating the flow to the rotor disk velocity. As a result, the relative total temperature of air supplied to the rotor blades can be reduced. Hence, the pre-swirl system is always used to deliver cooling air to rotor blades. Correspondingly, many engine designers pay attention to the pressure loss, temperature reduction as well as heat transfer characteristics in the pre-swirl system.

Meierhofer and Franklin [1] were first to carry out an experiment to verify that swirling cooling air can significantly reduce the relative total temperature of cooling air delivered to the rotor blades, and they quantified the effectiveness of pre-swirl system in terms of swirl ratio which is defined as the ratio of cooling air tangential velocity to rotor disk speed. The pre-swirl system was classified into two categories by Karabay et al. [2], including direct-transfer system and cover-plate system. The first represented that the pre-swirl nozzles were located in the stator at nearly the receiver-hole radius, while the second implied that the pre-swirl nozzles were located in the stator radially inward of the receiver holes, and the air flowed through the cavity enclosed by a cover-plate and turbine disk. It should be noted that both pre-swirl systems have axial pre-swirl nozzles.

El-Oun and Owen [3] performed a theoretical model in terms of adiabatic effectiveness based on the Reynolds analogy, and this model indicated that the relative total temperature at receiver holes decreased monotonically as pre-swirl ratio increased. Geis et al. [4] also developed a theoretical model to evaluate the relative total temperature, and the predicted result is significantly lower than their measured values. Chew et al. [5] carried out an experimental and elementary method to calculate the adiabatic effectiveness. Chew et al. [6] also numerically investigated the pre-swirl systems based on both the 'Karlsruhe rig', used by Geis et al. [4], and a 'Sussex pre-swirl rig', and they found the predicted results

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Nomenclature

A	area, m^2
a	cavity inner radius, m
b	cavity outer radius, m
$C_{d,b}$	discharge coefficient of receiver holes, $m_b/m_{i,b}$
$C_{d,p}$	discharge coefficient of pre-swirl nozzles, $m_p/m_{i,p}$
c_p	specific heat capacity at constant pressure, J/(kg K)
$C_{p,t}$	total pressure loss coefficient in pre-swirl cavity, $(p_{1,t}-p_{plane1,t})/(p_{1,t}-p_{plane1})$
$C_{p,t,n}$	total pressure loss coefficient in pre-swirl nozzle, $(p_{0,t}-p_{1,t})/(p_{0_t}-p_1)$
D	nozzle diameter, m
h	heat transfer coefficient, W/m^2 K, $q_w/(T_w-T_{w,ad})$
k	thermal conductivity, W/m K
m	mass flow rate, kg
M	wall moment
n	rotation speed, rpm
p	static pressure, Pa
p_t	total pressure, Pa
q_w	rotor wall heat flux, W/m^2
r	radius, m
r_p, r_b	radius of pre-swirl outlet and receiver holes outlet, m
Re_ϕ	rotational Reynolds number, $\rho\Omega b^2/\mu$
$T_{0,t}$	total temperature at cavity inlet in the stationary frame, K
$T_{b,t,rel}$	total temperature at receiver holes outlet in the rotating frame
V	absolute velocity, m/s
r/b	non-dimensional radius

Greek letters

π	pressure ratio, $p_{0,t}/p_b$
β	swirl ratio, $V_\phi/(\Omega r)$
φ	nozzle exit flow angle, °
θ	pre-swirl nozzle angle, °
$\Theta_{b,ad}$	adiabatic effectiveness, $\Theta_{b,ad}=c_p(T_{0,t}-T_{b,t,rel})/(0.5\Omega^2 r_b^2)$
μ	dynamic viscosity, kg/s m
ρ	density, kg/m^3
Ω	angular velocity of rotor, rad/s
$\Delta\theta$	the difference between pre-swirl nozzle angle and nozzle exit flow angle, $\Delta\theta=\theta-\varphi$, °

Subscripts

ad	adiabatic
b	blade-cooling
i	isentropic value
in	inlet
0	inlet of cavity
1	outlet of pre-swirl nozzle
2	outlet of receiver holes
p	pre-swirl
$plane1$	value at plane1
rel	relative value in rotating frame
t	total value
ϕ, r, z	circumferential, radial and axial direction

were in good agreement with the experimental data of both rigs, and the adiabatic effectiveness of the Karlsruhe rig is lower due to the larger stator area of the pre-swirl cavity. Farzaneh-Gord et al. [7] also developed theoretical models to calculate the adiabatic effectiveness, taking account of the moment on the stator walls.

Popp et al. [8] numerically investigated the discharge coefficient of a pre-swirl system, and the results showed that a maximum value of discharge coefficient for receiver holes can be observed as the tangential velocity differential is at a minimum. Dittmann et al. [9] were the first to measure the discharge coefficients of receiver hole and confirmed the numerical results of Popp et al. [8]. In the following years, Chew et al. [5], Yan et al. [10], Lewis et al. [11] and Bricaud et al. [12] also measured and calculated the discharge coefficients. Ciampoli et al. [13] performed an unsteady numerical simulation to calculate the nozzle and receiver hole discharge coefficients, and they found that the predicted results in terms of discharge coefficient showed a good agreement with experimental data.

Lewis et al. [14] performed a CFD study to investigate the effect of the radial location of the pre-swirl nozzle on the performance of pre-swirl system. Didenko et al. [15] numerically investigated the effect of geometrical parameters in terms of the cavity width, pre-swirl nozzle location radius and way of air delivery on the adiabatic effectiveness, loss coefficient, and discharge coefficient. Liao et al. [16] conducted a numerical study to investigate the influences of geometrical parameters, including the nozzle angle, the fillet radius and the length to diameter ratio of pre-swirl nozzle, on the pre-swirl efficiency and total pressure loss. Zhang et al. [17] also investigated the effect for length to diameter ratio of nozzles on performance of pre-swirl system.

An experimental and numerical study on the heat transfer characteristic of the pre-swirl system was carried out by Wilson et al. [18] who found that the axisymmetric CFD model under predicted the measured Nusselt numbers by up to 25% near the outer shroud. Pilbrow et al. [19] and Karabay et al. [20,21] also performed experimental and numerical studies to investigate heat transfer characteristics in a pre-swirl system. Lock et al. [22] and Kakade et al. [23–25] adopted thermochromic liquid crystal technique to measure the heat transfer coefficient on the rotor disk. Corresponding numerical investigations are given by Farzaneh-Gord et al. [7], Lewis et al. [11], Karnahl et al. [26] and Javiya et al. [27]. Liao et al. [16] adopted two approaches to calculate the heat transfer coefficient on the rotating walls of a pre-swirl system, and they found that the value of heat transfer coefficient predicted by Formula Method is larger than that calculated by Superposition Method. Luo et al. [28–30] conducted experimental and numerical studies to investigate the flow and heat transfer characteristics of a rotor–stator cavity with an inlet at low radius and an outlet at high radius.

Above investigations paid attention to the axial pre-swirl system. This means that pre-swirl nozzles are axially arranged on the stator wall. However, there is a radial pre-swirl system where the cooling air is swirled by pre-swirl nozzles radially mounted on the stator wall and flows radially inward to the cavity. Granovskiy et al. [31] presented a comparison in flow structure and temperature distribution between a radial pre-swirl system and an axial pre-swirl system. The results showed that two pre-swirl systems have very similar characteristics despite the fact that the flow structure is significantly different. El-Sadi et al. [32] also compared the performance of the radial and axial pre-swirl system, and they found that compared with the radial pre-swirl system, the axial pre-swirl system had a positive impact on the parasitic work and