Applied Thermal Engineering 105 (2016) 646–658

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Research Paper

Numerical investigation of the flow and heat transfer characteristics for a pre-swirl rotor–stator system with center inflow

Feng Zhang, Xinjun Wang *, Jun Li

Institute of Turbomachinery, Xi'an Jiaotong University, Xi'an 710049, China

HIGHLIGHTS

- We study the pre-swirl rotor–stator system with center inflow.
- The influences of the center inflow on flow dynamics have been studied.
- A theoretical expression of adiabatic effectiveness with center flow has been derived.
- The center inflow effects on heat transfer characteristics have been studied.

ARTICLE INFO

Article history:
Received 4 September 2015
Accepted 12 March 2016
Available online 19 March 2016

Keywords:
Pre-swirl rotor–stator system
Center inflow
Adiabatic effectiveness
Discharge coefficient
Heat transfer

ABSTRACT

The effects of the center inflow on the flow and heat transfer characteristics in a pre-swirl rotor–stator system are numerically investigated using the CFD software ANSYS-CFX. The investigation of the pre-swirl system without center inflow is also conducted to be served as a contrast and these results are compared with experimental measurements to verify the computational method. In this paper, five mass flow rate ratio of the center flow to the pre-swirl flow, *mf*, four dimensionless mass flow rate of pre-swirl flow, C_w, and five rotation speeds, *n*, are selected to study the flow dynamics of the pre-swirl system in terms of the flow structure, the air swirl ratio, the total pressure loss coefficient, the discharge coefficient of the receiver holes and the adiabatic effectiveness. It shows that the maximum value of the discharge coefficient occurs as swirl ratio in the core at the receiver-hole radius equals 1 and *mf* reaches as a high value as possible, and the adiabatic effectiveness decreases as *mf* increases. In addition, the relation between the Nusselt number, *Nu*, and *mf*, C_w as well as *n* is also illustrated and discussed.

1. Introduction

The cycle thermal efficiency and power output of advanced gas turbines can usually be improved by increasing the temperature at entry to the turbines. This temperature has already reached 2000 K, which exceeds the melting point temperature of many components materials. The advanced cooling method is consequently adopted to prevent these high temperature components from a failure. Note that the cooling air bled from compressor is guided through the secondary air system to the thermally loaded components (such as first stage blades). The pre-swirl rotor–stator system is always used to deliver cooling air to rotor blades as shown in Fig. 1. It can be seen that the pre-swirl rotor–stator system is a wheel space which is consist of a rotor disk and a stationary casing. Also note that the inlet referred as pre-swirl nozzle including angled nozzle and vane is located at the stator wall, while the outlet referred as receiver holes is mounted on the rotor disk. Additionally, two gaps between the stator wall and the rotor disk can be observed. In the pre-swirl rotor–stator system, the pre-swirl nozzles swirl the cooling air in the rotating direction, and this reduces the work done on the air by the rotor disk in accelerating the flow to the rotor disk speed. This consequently reduces the relative total temperature of cooling air supplied to the rotor blades. Therefore, the pre-swirl system has already been widely implemented in the gas turbine. For example, the direct-transfer pre-swirl system was used in the jet engine of Rolls-Royce PLC [1], and the result of the test program about vane type pre-swirl nozzle was instrumental in the design of the pre-swirl nozzles for the H-class SGT5-8000H gas turbine manufactured by Siemens Energy [2]. In addition, a radial pre-swirl system was applied to the gas turbine of Alstom Ltd [3].

It is recognized that the pre-swirl rotor–stator system is bounded by a rotating disk and a stationary casing. Inevitably, there are two seals placed between the rotating disk and stationary

* Corresponding author.

http://dx.doi.org/10.1016/j.applthermaleng.2016.03.060
1359-4311/

Nomenclature

a	cavity inner radius, m
b	cavity outer radius, m
C_D	discharge coefficient of receiver holes, m_b/m_i
c_p	specific heat capacity at constant pressure, J/(kg K)
$C_{p,t}$	total pressure loss coefficient, $(P_{0,p} - P_{0,0.9b})/(P_{0,0.9b} - P_{s,0.9b})$
C_w	non-dimensional mass flow rate, $m/(\mu b)$
h	heat transfer coefficient, W/m^2 K
k	thermal conductivity, W/m K
m	mass flow rate, kg
mf	mass flow rate ratio of center inflow to pre-swirl inflow
mr	the ratio of mass flow rate of receiver hole to that for $mf = 0$ and $C_w = 6600$
M	disk moment
n	rotation speed, rpm
Nu	Nusselt number, $(q_w r/k(T_w - T_{w,ad}))$
P_s	static pressure, Pa
P_0	total pressure, Pa
q_w	rotor wall heat flux, W/m^2
r	radius, m
r_p, r_b	radius of pre-swirl inlet and receiver holes, m
Re_ϕ	rotational Reynolds number, $\rho\Omega b^2/\mu$
S	rotor stator separation distance, m
$T_{0,p}$	total temperature at pre-swirl nozzles inlet in the stationary frame, K
$T_{0,b,rel}$	total temperature at receiver holes outlet in the rotating frame
V	absolute velocity, m/s
x	non-dimensional radius, r/b

Greek letter

β	swirl ratio, $V_\phi/(\Omega r)$
$\Theta_{b,ad}$	adiabatic effectiveness, $\Theta_{b,ad} = C_p(T_{0,p} - T_{0,b,rel})/\left(0.5\Omega^2 r_b^2\right)$
μ	dynamic viscosity, kg/s m
ρ	density, kg/m^3
Ω	angular velocity of rotor

Subscripts

ad	adiabatic
b	blade-cooling
i	isentropic value
0	total value in stationary frame
p	pre-swirl
rel	relative value in rotating frame
s	static value
ϕ, r, z	circumferential, radial and axial direction
∞	value in core at $z/S = 0.5$
0.9b	value at $r = 0.9b$ surface

casing. One of seals near turbine end-walls is called the rim seal which plays an important role to prevent the hot gas being ingested into the pre-swirl rotor–stator system and the cooling air being egressed to the mainstream. The other seal placed at inner radius is referred to as inner seal. In order to prevent cooling air being egressed through the inner seal, the center inflow is introduced to balance the pressure across the inner seal. It should be noted that the advanced gas turbine operates in high rotation speed and high temperature circumstance, which leads to a variation in the inner seal clearance. Hence, the pressure balance across the inner seal is broken, and then the center inflow would be ingested into the pre-swirl cavity through the inner seal. Clearly, the center inflow has significant effect on the performance of pre-swirl rotor–stator system. Therefore, it is essential to investigate the effect of center inflow on the flow and heat transfer characteristics of a pre-swirl rotor–stator system.

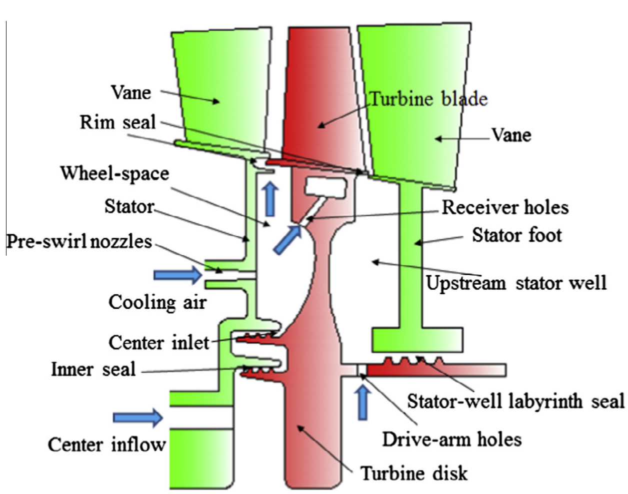

Fig. 1. Schematic view of pre-swirl rotor-stator system [29].

Researchers have investigated the flow and heat transfer characteristics of a pre-swirl rotor–stator system. Early experimental studies conducted by Meierhofer and Franklin [4] confirmed that swirling cooling air can significantly reduce the relative total temperature. Later on, El-Oun and Owen [5], Geis et al. [6], Chew et al. [7] and Farzaneh-Gord et al. [8] performed measurements, numerical calculations and theoretical analysis to investigate the relative total temperature drop of a pre-swirl rotor–stator system, and in their studies the relative total temperature drop was expressed by a dimensionless parameter in terms of adiabatic effectiveness. It should be noted that as the adiabatic effectiveness is above zero, the relative total temperature of the fluid feeding the blade is lower than the total temperature of the fluid through the nozzle, which indicates that the pre-swirl rotor–stator system can effectively reduce the relative total temperature. Consequently, to design a more effective cooling system, the engine designers strive to enhance the adiabatic effectiveness. Correspondingly, the adiabatic effectiveness is one of the significant parameters studied in this paper.

It is recognized that a high-performance pre-swirl system can not only provide low-temperature fluid, but also minimize the losses occurring within the system. In the pre-swirl system the losses occurs in the pre-swirl nozzle, the wheel space and the receiver hole. Note that the losses in the receiver holes are of interest to the designer, and it is quantified using the dimensionless parameter in terms of the discharge coefficient of the receiver hole. Consequently, many researchers have focused on the discharge coefficient of the receiver hole. Popp et al. [9] and Dittmann et al. [10] found that the discharge coefficient of the receiver hole was dominated by the relative tangential velocity between the fluid and the rotor disk, and a maximum value of discharge coefficient for receiver holes could be observed as the relative tangential velocity was close to zero. In the following years, Chew et al. [7], Yan et al. [11], Lewis et al. [12], and Cagan et al. [13] also measured