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Review

Effect of the dimple location and rotating number on the heat transfer and flow structure in a pin finned channel

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ABSTRACT

In this study, the Realizable $k-\varepsilon$ turbulent model combined with the wall enhanced function is used to evaluate the heat transfer and flow structure in a rotating pin finned channel with different dimple locations. The Re number is fixed at 7000. The Ro number is varied from 0 to 1.0. Three different dimple locations are introduced to investigate the effect of the dimple location on heat transfer and flow structure. The streamline, dissipative function and Nusselt number are shown and discussed in this study. A counter-rotating vortex, which is induced by the Coriolis force, is found at the back of the pin fin. It transports wake fluid from the trailing side to leading side. As the dimple moves close to the leading edge of the pin fin, the dissipative function and Nusselt number are increased remarkably. As the Ro increased, the Nusselt number is increased both at the leading side and trailing side. However, the Nusselt number at the trailing side shows a higher growth rate compared to that at the leading side. The lower Nusselt number region at the trailing side wake is also decreased as the Ro increased. The wall shear stress integral is decreased at the leading side while it is increased at the trailing side as the Ro increased.

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Nomenclature

d_p	pin fin diameter (mm)
d_d	dimple diameter (mm)
D_h	hydraulic diameter of the inlet (mm)
f	friction factor
f_0	friction factor in smooth channel ($0.316Re^{-0.25}$)
H	channel height (mm)
L	distance between the dimple and pin fin
h	heat transfer coefficient ($W \cdot m^{-2} \cdot K^{-1}$)
Nu	Nusselt number
Nu_0	Nusselt number in smooth channel ($0.023Re^{0.8}Pr^{0.4}$)
Pr	Prandtl number
P_{in}	inlet pressure (Pa)
P_{out}	outlet pressure (Pa)
q	wall heat flux ($W \cdot m^{-2}$)
Q	Q criterion about the vortex
R	mean radius (mm)
Re	Reylonds number
Ro	rotating number
S_x	pin fin spanwise spacing (mm)
S_y	pin fin streamwise spacing (mm)
T_{in}	inlet air temperature (°C)
T_w	endwall temperature (°C)
T_b	local bulk temperature (°C)
U	relatively mean velocity ($m.s^{-1}$)
u	velocity in x direction ($m.s^{-1}$)
v	velocity in y direction ($m.s^{-1}$)
w	velocity in z direction ($m.s^{-1}$)
δ	dimple depth (mm)
λ	thermal conductivity of air ($W \cdot m^{-1} \cdot K^{-1}$)
μ	dynamic viscosity ($Pa \cdot s$)
τ	wall shear integral (N)
Φ	dissipation function
ρ	density of air ($kg \cdot m^{-3}$)
Ω	rotating speed

Abbreviation

CFD	computational flow dynamics
Per	periodic wall
TKE	turbulent kinetic energy

1. Introduction

Due to the high demand for the efficiency and output power of the gas turbine, the most advanced turbine inlet temperature is more than 2000 K [1,2], which is far above the permissible of the metal materials. Therefore, the cooling air, which is from the compressor, is used to protect the blade surface. For all the region of the gas turbine blade, the trailing edge is a most difficult region to cool due to the thin thickness and narrow space [3–6]. Thus, it is worth paying more attention to enhance the cooling capacity to protect the trailing edge. At present, the pin fin [7], rib [8], impingement [9], swirl [10], dimple/protrusion [11] and lattice-work [12] are investigated to protect the trailing edge. Compared all the cooling structure, the pin fin is widely used in practical gas turbine. This is because the pin fin can increase the heat transfer and structural strength simultaneously [13].

Since the 1980s, numerous studies had been published on the heat transfer and flow structure in a pin finned channel. The horseshoe vortex and vortex shedding, which were induced by the pin fin, contributed to the heat transfer augmentation [14,15]. Recently, a great number of the literatures mainly tried to find the optimal parameter of the pin fin cross section shape, arrangement, length and main flow parameter to increase the heat transfer with small pressure loss. As for the shape, the circular [14–20], diamond [16,18–22], ellipse [16,20,23], triangular [19,20], and so on were investigated. In general, the pin fin which had the sharply edge, i.e., diamond and triangular, provided a higher heat transfer enhancement compared to the circular ones. This was because the horseshoe vortex wrapped and spread much wider laterally near the pin fin. The pin fin arrangement also affected the flow structure and heat transfer [24,25]. Actually, it was hard to decide which arrangement was better in the heat transfer. Not only the regular arrangement (in-line and staggered), but some irregularity arrangement was investigated [26–28]. The Nusselt number dependence on the varying stagger spacing and streamwise spacing was illustrated by Allan et al. [26] and Ostanek et al. [27], respectively. It was found that the Nusselt number was sensitive to the stagger spacing and streamwise spacing simultaneously. Bianchini et al. [28] compared to the heat transfer characteristics between the pentagonal displacement and staggered arrangement. Results showed that the staggered arrangement provided more uniform heat transfer distribution. Some literatures also investigated the effect of the pin fin length- to- diameter [19,29,30] on heat transfer. A general conclusion was that the endwall had a notable effect on heat transfer in a low pin fin length- to- diameter, corresponding to a larger proportion horseshoe vortex. A detailed summary about the pin fin had been done by Ligrani et al. [31,32] and Han et al. [33,34]. It was found that the maximum of Nu/Nu0 in a pin fined channel was about 5.5, which cannot meet the requirements of the most advanced gas turbine. Therefore, the dimple was used to enhance the heat transfer at the trailing edge. The advantage of the dimple was that it increased the heat transfer with slight pressure drop penalties [31,35–38]. The auto oscillations generated by the dimple contributed to the heat transfer enhancement [35]. According to the previous studies, the dimple shape had a significant effect on the heat transfer and flow structure. The dimple shape with circular, tear [39,40], ellipse [41,42] and so on were investigated extensively. Generally speaking, the tear shaped dimple provided a better thermal performance with respect to the circular ones. Although the dimple attracted the attention from the gas turbine industry and academia, the disadvantages of the dimple was notable. Firstly, dimple only induced a relatively lower heat transfer augmentation ratios [43]. The maximum heat transfer augmentation ratio in dimpled channel was only approximately 3.5 [31]. Secondly, the dimple reduced the blade thickness and exacerbated the structural damage in the gas turbine. To keep the advantages and remove the disadvantages of the dimple, many literatures combined the pin fin with dimple to enhance the heat transfer [44–47]. Results showed that the Nusselt number was increased remarkably after adoption of the dimple in a pin finned channel.

At present, the studies of the heat transfer and flow structure in a pin finned channel with dimple was in a static condition. However, the centrifugal force and Coriolis force played a very important role in the heat transfer and flow structure at the rotating blade for the gas turbine. Some literatures had already investigated the heat transfer and flow structure in a pure pin finned channel under stable rotating condition [48–52]. It reported that the Coriolis force made a visible difference for the heat transfer between the leading side and trailing side. The Ro number, which accounted for the ratio between the Coriolis forces to the inertia of the fluid, was used to quantifiable evaluate the effect of the rotating. The

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