COMPARATIVE ANALYSIS OF TURBINE BLADE TRAILING EDGE COOLING CONFIGURATIONS

Grazieli L. C. Carosio, grazieliglcc@iae.cta.br
Marcio T. Mendonca, marciomtm@iae.cta.br
Departamento de Ciência e Tecnologia Aeroespacial, Instituto de Aeronáutica e Espaço, São José dos Campos, SP, 12228-904, Brazil

Abstract. *In the present investigation, numerical simulations are used to compare two turbine blade trailing edge cooling configurations. The classic circular pin fin configuration is compared to a configuration composed with short plates. In the first configuration increased heat transfer is achieved through higher turbulence levels, which also increases pressure drop along the passage. In the second configuration increased heat transfer is achieved on the plates due to the thin developing boundary layer. The results show that a higher number of plates are necessary to achieve the same or higher cooling rates. However, even with a grater number of plate fins a lower pressure drop is achieved. The proposed configuration allows a higher cooling with a smaller pumping power, therefore lower losses through the cooling system.*

Keywords: *gas turbine, convective cooling, computational fluid dynamics, compact heat exchangers, pin-fin*

1. INTRODUCTION

High performance gas turbines require blade cooling due to the high temperature and pressure used in the thermodynamic cycle. Cooling air is employed in different ways to protect the blade. Convection cooling air is usually discharged through film cooling holes and at the trailing edge of the blade. Due to the small thickness of the cooling air passages at the trailing edge, cooling is critical process. Usually the designer faces the problem of increasing the heat transfer rate at the cost of higher pressure drop through the cooling passage. The most effective way to increase convective cooling coefficients is through turbulence, which also increases pressure drop significantly. The pressure drop through the trailing edge is sometimes used as a way to control the air flow through the secondary air system. Nevertheless, this is not always desirable and the designer may use the fin geometry as another control tool when smaller pressure drops are necessary.

Heat transfer and pressure loss characteristics of different array configurations with circular pin fins are commonly found in previous studies in pin fin cooling investigations (Fossen, 1982; Simoneau and Fossen, 1984; Lau *et al.*, 1989; Ligrani and Mahmood, 2003; Won *et al.*, 2004). Armstrong and Winstanley (1988) presented a review of heat transfer and pressure loss data for staggered arrays of circular pin fins in turbine cooling applications. A comparison of the various heat transfer augmentation techniques used in internal coolant passages, including pin fins, is performed by Ligrani *et al.* (2003). Different pin fin shapes and concepts have been studied as alternatives to circular fin, for example, cube- and diamond-shaped pin fins (Chyu *et al.*, 1998), elliptical pin fins (Uzol and Camci, 2005) and partial pins (Arora and Abdel-Messeh, 1989; Chi *et al.*, 2011).

Chyu *et al.* (1998) used cube- and diamond-shaped pins to enhance heat transfer coefficient from a surface. They measured mass transfer from a naphthalene surface and used the mass/heat transfer analogy to infer heat transfer results. The general trend of mass-transfer enhancement does not change by changing the shape of the pins. There is an initial increase in mass-transfer coefficient with increasing row number, and then the mass-transfer coefficient subsides to its fully developed value. In general, cube-shaped pins show higher mass-transfer coefficients near the inlet than that with diamond pins. It also shows that the cube-shaped pins have the highest mass-transfer coefficients among the shapes considered; round pins have the lowest mass-transfer coefficients. Corresponding pressure loss coefficients are higher for the cube and diamond shaped pins relative to the circular pins.

Results of an experimental investigation on the endwall heat transfer enhancement, total pressure loss, and wake flow field characteristics of circular and elliptical pin fin arrays were presented by Uzol and Camci (2005). Differences between the local enhancement patterns of the circular and elliptical pin fin arrays were observed. The elliptical fins have a weaker Reynolds number dependency compared to the circular pin fins, possibly due to boundary layer and separation characteristics. It was determined that, in terms of heat transfer enhancement performance, the elliptical fins not only have a lower performance compared to the circular fins, but also seem to be the least effective device among some of the other pin fin shapes that have been investigated by previous researchers.

Arora and Abdel-Messeh (1989) studied the effect of half pins on the heat-transfer coefficient. The friction factor for different geometries of partial pins is lower for partial pins compared to full-length pins. The overall surface Nusselt numbers for different configurations were compared, showing that, in general, heat-transfer coefficient decreases when partial pins are used.

In the present investigation a proposal is made where, based on compact heat exchanger configurations, the usual pin-fin heat transfer enhancers are substituted by a staggered arrangement of short plates. The heat transfer enhancement is then obtained by a constant restart of a thin boundary layer on each short plate and not by increased turbulence levels. The proposed configuration results in lower pressure drop across the length of the trailing edge region. Comparisons are presented for a model configuration between the classic pin-fin and the proposed array of short plates. Both heat transfer and pressure drop are compared. The study was conducted numerically using an open source CFD code (OpenFOAM), where the Navier-Stokes are solved and a SST $k - \omega$ turbulence model was used to close the Reynolds Average equations. The present results reveal that a better thermal and hydrodynamic performance can be obtained with the proposed configuration.

2. DESCRIPTION OF THE PROBLEM

The research presented in this paper was performed using OpenFOAM [1], an open source C++ colection of libraries for computational fluid dynamics. The compressible Navier-Stokes equations are solved numerically based on a cell centered unstructured finite volume scheme. The turbulent stress terms were closed using a two equation turbulence model based on the Boussinesq turbulent viscosity hypothesis.

The solution methodology is based on a segregated, compressible version, pressure based SIMPLEC algorithm and a steady state solver. An algebraic multigrid solver with preconditioning is used for pressure and velocity whereas a preconditioned, bi-conjugate gradient solver is used for the turbulent quantities and the energy equation. The vector field is interpolated using a Gauss linear schemes or a combination of Gauss linear scheme and first order Gauss upwind scheme.

The study considers a model problem where cooling air flows through a channel with circular pin fins or plates as shown in Figs. 1 to 3. The geometry is based on the work of Moon *et al.* (2011), where the circular pin diameter is $D = 6.345 \times 10^{-3}$ m, the plate thickness is $0.1D = 0.6345 \times 10^{-4}$ m, the plate length is $L = D$, the channel height is $H = 2D = 12.69 \times 10^{-3}$. The horizontal spacing between pins (or plates) is $X = 2D = 12.69 \times 10^{-3}$ m. A straight channel $2X$ long is used upstream of the pin fin/plate region in order to allow the flow to develop from the inlet boundary condition. The channel pin fin/plate region length is $L = 2X$ and the spanwise spacing is $Y = S/2 = D$. At the exit a straight channel with length $4X$ is used to avoid outflow boundary condition effects on the domain of interest. The pin fin/plate section has 2 rows of staggered pin fin/plates. In the spanwise direction a symmetry condition is imposed and a larger number of plates is simulated by reducing the spanwise domain.

Velocity and temperature profiles are imposed at the inlet. These profiles were obtained from a previous simulation on a $8X$ long entrance adiabatic straight channel. A fixed temperature of 429.8 K is imposed on the walls. At the exit a pressure level of 101320 Pa is imposed. Turbulence parameters are imposed at the inlet corresponding to a level of 10%.

3. RESULTS

The experiments were performed on an i7 2600 Intel processor, 3.4Ghz, 8GB RAM, under Ubuntu Linux 10.04 64bits.

[1] http://www.openfoam.com/

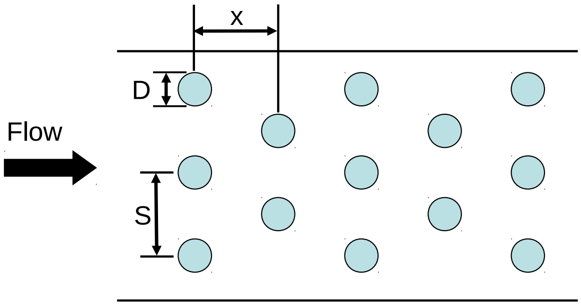

Figure 1. Circular pin fin cooling configuration.

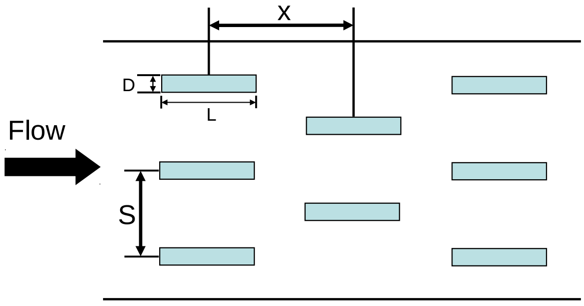

Figure 2. Plates cooling configuration.

Figure 3. Computational domains.

Table 1 shows a grid dependency study. A comparison was performed considering the average temperature at the exit, pressure drop through the channel, total heat flux through the walls and the turbulent wall distance of the first grid point $y+$. Four different grid densities were considered, 119,624 grid cells, 244,992 grid cells, 481,536 grid cells and 999,216 grid cells. All cells are composed of hexahedral volumes. The grid was created using blockMesh, a grid generator available in OpenFOAM.

The first grid shows results that compare well with the results of the other grids, even presenting a higher $y+$, which ideally is around $y+ = 1$ for the $k - \omega$ SST turbulence model used. For the rest of the simulations, the second and third grids were used.

Table 2 shows a similar grid dependency study, but considering the geometry with plates. The following grid densities where considered: 123,240; 249,600; 485,000 and 972,160. The following simulations were performed with the second and third grid sizes.

Table 1. Grid dependency test. Circular pin fins.

	T(K)	dp(Pa)	q(W)	y+
case 1	339.984	98	7.837124	4.71881
case 2	340.720	100	7.717022	2.69111
case 3	340.165	102	7.633663	1.67256
case 4	339.333	104	7.643072	1.21896

Table 2. Grid dependency test. Plate fins.

	T(K)	dp(Pa)	q(W)	y+
case 1	336.152	12	5.32592	4.10926
case 2	335.759	12	5.24026	3.43931
case 3	335.513	11	5.22613	2.75469
case 4	335.267	11	5.24936	2.06016

Figure 4 shows the velocity distribution on a longitudinal cut through the center of the channel. Flow separation behind the pins and flow acceleration on the minimum area throat are clearly noticed on the pin fin configuration. They are more severe than the separation and acceleration observed on the plate configuration. These flow separation and acceleration are associated with higher turbulence levels. Figure 5 shows that the turbulent kinetic energy on the pin fin configuration is 5 times higher than the turbulence levels found on the plate configuration. This higher turbulence level increases heat transfer through pin fin walls as well on the bottom and top walls. Most of the turbulence is generated downstream of the pins and on the reattachment regions. Be aware of the scales presented on these figures which are not adequate for direct visual comparison between configurations.

Figure 6 shows a comparison between the heat flux on the walls for the two configurations. For the plates configuration, the spanwise distance between plates is such that it corresponds to a channel with the same width of the pin fin channel but with four plates. The maximum heat flux on the plate configuration is higher than the heat flux on the pin fin configuration.

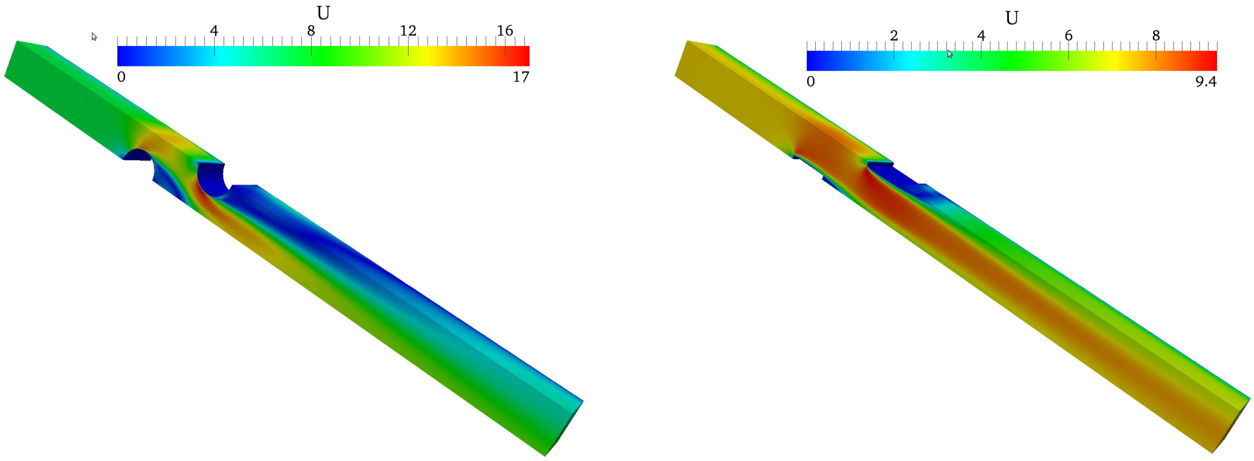

Figure 4. Velocity magnitude distribution.

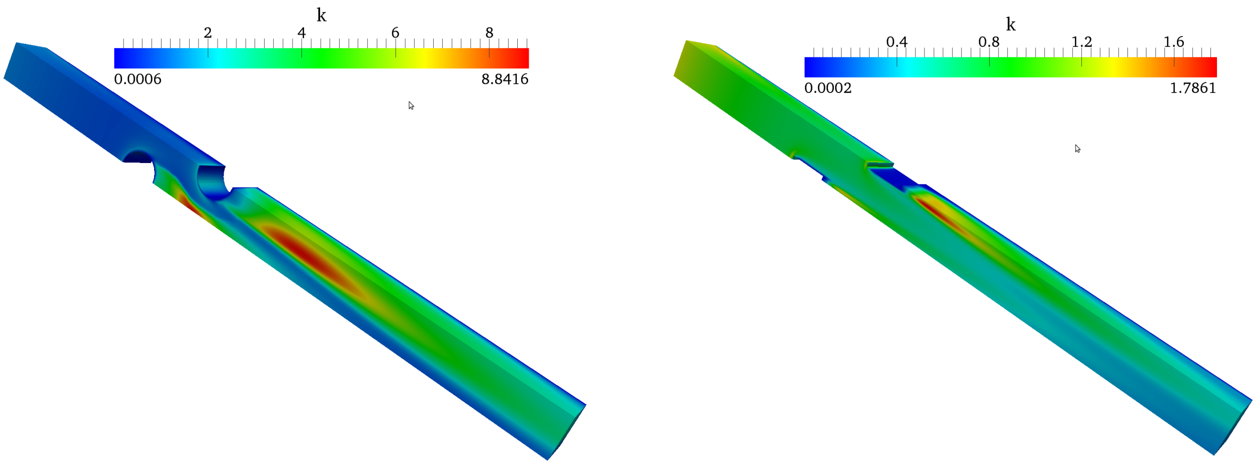

Figure 5. Turbulent kinetic energy distribution.

The strongest heat flux takes place at the leading edge of the fins, either circular pins or plates.

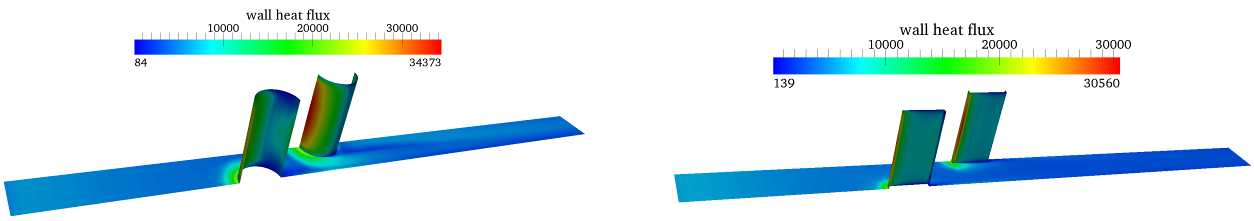

Figure 6. Wall heat flux on the walls.

Figure 7 shows streamlines colored by turbulent kinetic energy for the configuration with two pin fins. Detail of the flow structure about the pin fins can be seen. Two different groups of streamlines are shown in Figs. 7 (a)-(c), one for the flow over the first (upstream) pin and another for the flow over the second (downstream) pin. Figure 7 (a) shows the streamlines on the main stream, while Figs. 7 (b) and (c) show the streamlines that form the separation bubble. The three dimensional separation bubble contains fluid that comes from the top and bottom wall boundary layer. This fluid recirculates inside the bubble in a spiral streamline towards the center of the pin fin, leaves the bubble and mix with the main stream.

On the second pin, turbulent kinetic energy is generated on the shear layer form on the interface between the flow on the bubble and the main flow. In turn, the first pin turbulent kinetic energy is generated on the region at the trailing edge of the bubble where the main flow from the upper side of the bubble meets the flow from the lower side of the bubble. In this case, the mechanism is similar to a impinging jet mechanism. The simulation shows that the first mechanism has a much stronger turbulence generation.

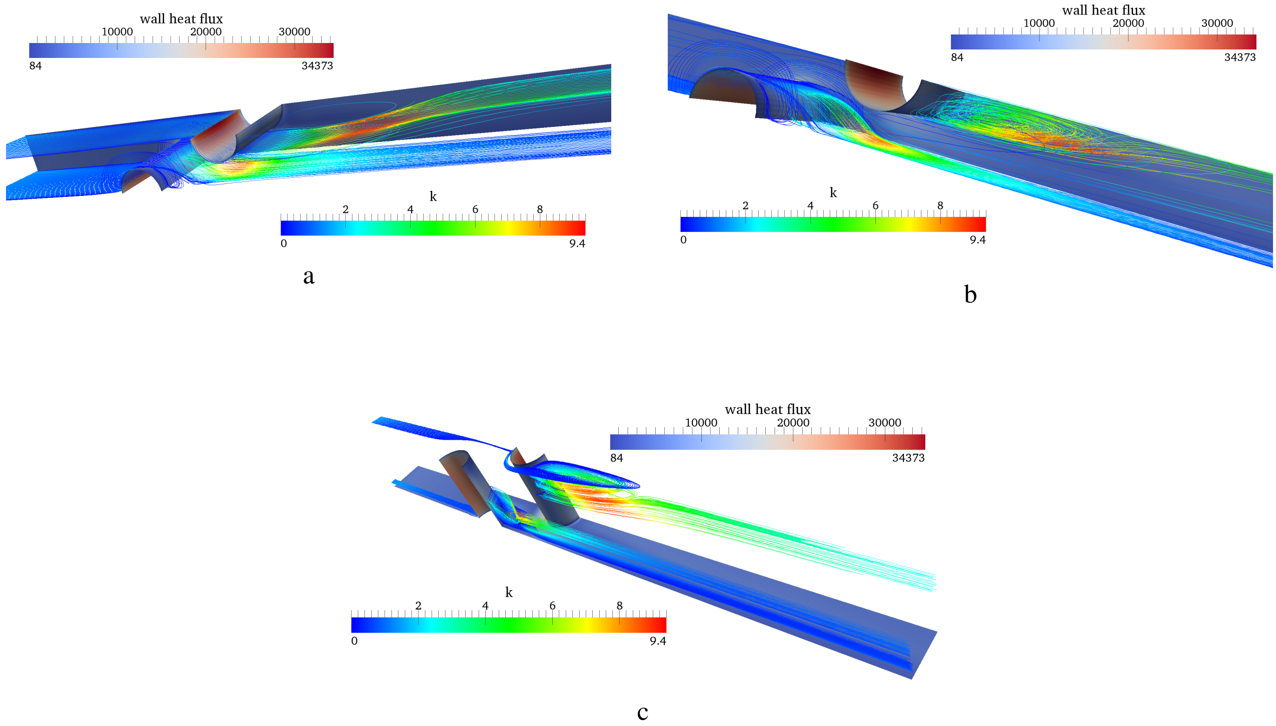

Figure 7. Streamlines, turbulent kinetic energy and wall heat flux distributions for the pin fin configuration.

Figure 7 also shows the wall heat flux distribution. The heat flux is larger at the leading edge of the pin as expected. The turbulence generation is away from that region and from the side walls. Therefore, this turbulence does not contribute directly to heat transfer increase, but has a significant effect on pressure drop on the passage.

On the other hand, the heat exchange on the wall boundary layer upstream of the pin fin warms up the air which is later drawn to the center of the channel when this fluid enters the bubble recirculation zone. This fluid is further warmed up by the greater residence time on the bubble zone. This effect can be clearly seen on Fig. 8 (a), where behind the second pin fin higher temperatures are observed on the upper part of the spanwise plane. As will be presented in Fig. 9 the separation bubbles behind the plates are much smaller and the temperature distribution on the spanwise plane shown in Fig. 8 (b) is more uniform.

Similarly, Fig. 9 shows the streamlines colored by turbulent kinetic energy for the configuration with two plates. This figure shows details of the flow structure about the plates. The differences between the pin fin and the plate configurations are in the magnitude of the velocities, the sizes of the separation bubbles, and some of the complexities of the flow structure.

The pin fin configuration predicted a larger bubble than the bubble observed behind the plate. The turbulent kinetic energy generation behind the plate is much lower than that on the pin fin configuration resulting in a much lower pressure drop. Again, the separation bubble is three dimensional and the fluid on the bubble comes from the top and bottom wall boundary layers. The fluid from the boundary layer is drawn to the center where it leaves the bubble and mix with the

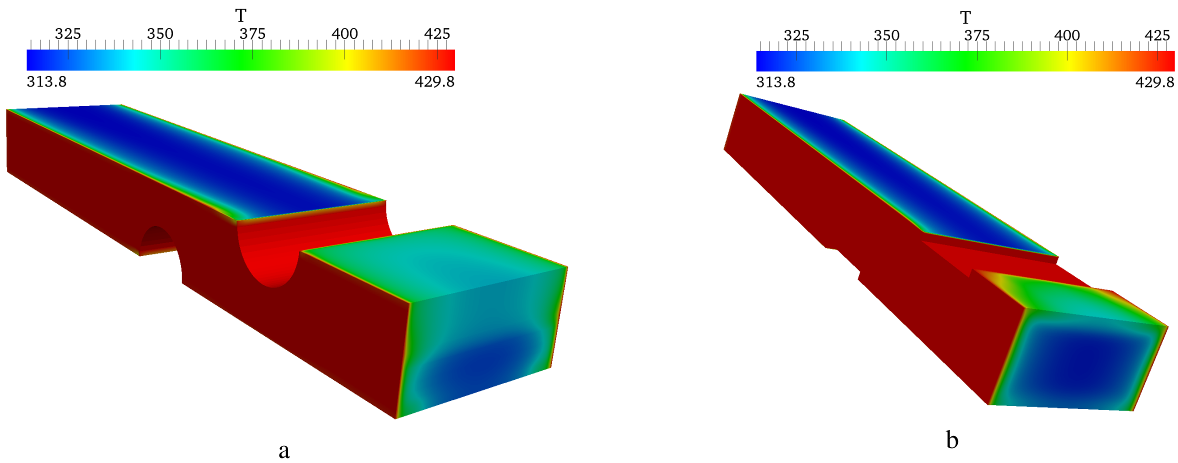

Figure 8. Spanwise plane temperature distribution for the pin fin and plate configurations.

main flow.

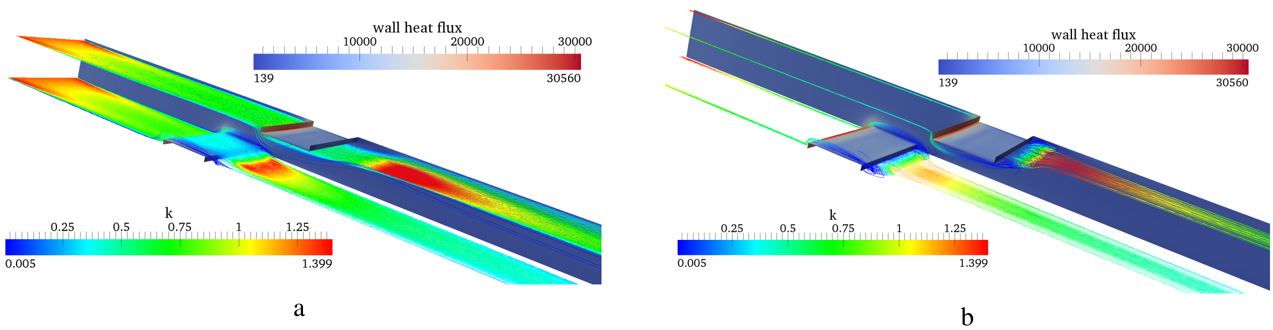

Figure 9. Streamlines, turbulent kinetic energy and wall heat flux distributions for the plate configuration.

The maximum heat flux on the plate leading edge is about the same as that on the pins and the heat transfer on the plates depends on the thin boundary layer over the plate wall. In order to increase the total heat transfer, a larger number of plates is needed. A comparison between the average temperature, wall heat flux and pressure drop for the circular pin fin and plate configurations is presented in Fig. 10. For this comparison, a configuration with only two pins was considered as the reference configuration and compared with configurations containing two to five plates. The objective of this analysis is to evaluate whether the use of plates instead of pins can be a viable alternative to achieve greater heat transfer and lower pressure drop.

Figure 10 (a) shows the average exit temperature, which is a direct measurement of the cooling capacity of each configuration. When more than three plates are used, one can conclude that the flow extracts more heat from the walls since the cooling air exit temperature is higher. The total heat flux from all walls (plates and fins) is presented in Fig. 10 (b) confirming the conclusion that when more than three plates are used there is a significant increase in heat transfer when compared to the pin fin configuration. For three plates, the exit temperature and the total heat flux is about the same for both the pin fin and plate configurations. On the other hand, as shown in Fig 10 (c) the pressure drop is considerably less. The higher the number of plates on the channel, the higher is the heat removed, but without considerable increase in the pressure drop through the channel, which becomes higher than that on the pins only for five plates or more.

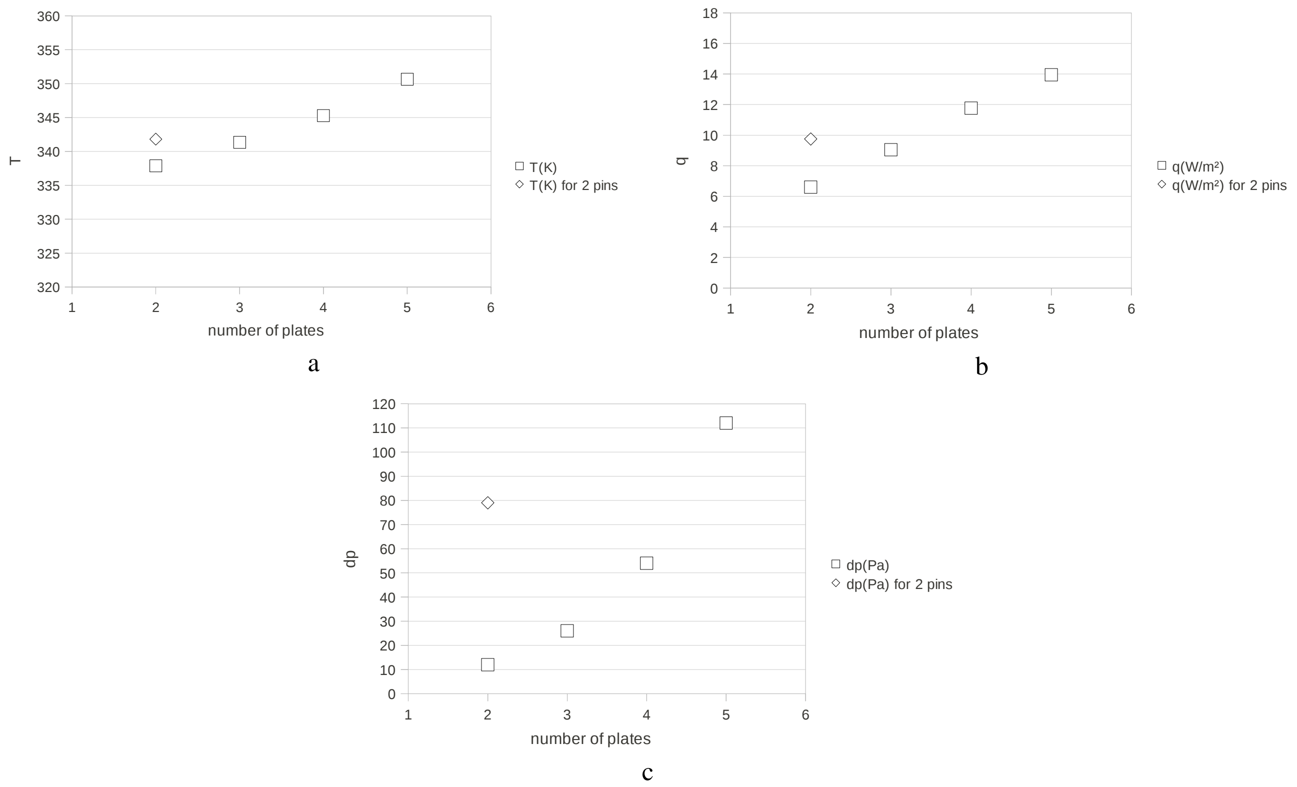

Figure 10. Cooling and pressure drop comparison.

4. CONCLUSIONS

This work proposes a configuration consisting of plate fins as heat transfer enhancement device on turbine blade trailing edges. In the simulation results, the comparison between pin fins and plates shows that the separation bubbles and the turbulence generation behind the pins are much larger than that on the plates. Therefore, the resulting pressure loss on the pin fin configuration is higher than that on the plate fin configuration when less than five plates are used. Regarding the total heat transfer, the plate fin configuration is better when more than three plates are used. The use of plates instead of pins may be a promising technique to achieve greater heat transfer and lower pressure drop. The results are encouraging and further investigations are under way in order to better understand the flow and heat transfer physics and characteristics of the proposed configuration.

5. ACKNOWLEDGEMENTS

The authors would like acknowledge the support from National Counsel of Technological and Scientific Development (CNPq) and from State of São Paulo Research Foundation (FAPESP).

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