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EXPERIMENTAL INVESTIGATION ON THE EFFECTS OF A LARGE RECIRCULATING AREA ON THE PERFORMANCE OF AN EFFUSION COOLED COMBUSTOR LINER

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ABSTRACT

An experimental analysis of a realistic engine cooling scheme was performed on a test article replicating a slot injection and an effusion array with a central large dilution hole. Test section consists of a rectangular cross-section duct with a flat plate comprised of 270 effusion holes arranged in 29 staggered rows (D = 1.65mm, Sx/D = 7.6, Sy/D = 6, L/D = 5.5, α = 30deg) and a dilution hole (D = 18.75mm) located at the 14th row. Both effusion and dilution holes are fed by a channel replicating a combustor annulus, that allows to control cold gas side cross-flow parameters, especially in terms of Reynolds number of both annulus and effusion holes. Upstream the first row, a 6mm high slot, ensure the protection of the very first region of the liner. In order to simulate the combustor flowpath, a backward facing step was installed upstream the slot to generate a large recirculating area. Adiabatic effectiveness, heat transfer coefficient and Net Heat Flux Reduction were evaluated and compared with non recirculating experiments. Measurements were performed by means of a steady-state Thermo-chromic Liquid Crystals (TLC) technique with a thin Inconel heating foil for the heat transfer measurements. A data reduction procedure based on a Finite Element approach has been developed to take into account the non uniform heat generation and conduction due to the large amount of holes. Experiments were carried out considering the combined effects of slot, effusion and dilution holes. Three different effusion blowing ratios (BR = 3-5-7) are investigated, keeping constant the slot flow parameters (BR = 1.3). Results highlight that the presence of the step leads to a general reduction of effectiveness while does not have effects on the heat transfer coefficient.

OVERVIEW

Over the last ten years, there have been significant technological advances toward the reduction of NO_x emissions, strongly aimed at meeting stricter legislation requirements. Some very encouraging results have already been obtained but the achieved solutions have created other technical problems.

To satisfy future ICAO standards concerning NO_x emissions, in the last years main engine manufacturers updated the design concept of combustors. Future aeroengines combustion devices will operate with very lean mixtures in the primary combustion zone, switching as much as possible to premixed flames. Whatever detailed design it will be selected (LPP, PERM or LDI), primary zone air amount grows significantly, while liner cooling air has to be decreased. Consequently, important attention must be paid in the appropriate design of liner cooling system; in addition, further goals need to be taken into account: reaction quenching due to cool air sudden mixing should be accurately avoided, whilst temperature distribution has to reach the desired levels in terms of both pattern factor and profile factor (see Lefebvre [1], Ballal and Zelina [2]).

In recent years, the improvement of drilling capability has allowed to perform a large amount of extremely small cylindrical holes, whose application is commonly referred to as effusion cooling. Even if this solution determines, at least in early part of the liner, a slight reduction of wall protection with respect to film cooling, the most interesting aspect is the significant effect of wall cooling due to the heat removed by the passage of coolant inside the holes [3–5]. In fact, a higher number of small holes, uniformly distributed over the whole surface, permits a significant improvement in lowering wall temperature. Even if early effusion cooling schemes were developed to be an approximation of transpiration cooling (several short normal injection holes, with reduced coolant momentum [6]), it is important to observe that present combustor effusion cooling design is usually based on very shallow injection holes (less than 30°) with high coolant jet momentum. This updated angled effusion cooling allows to greatly increase the heat sink effect (higher holes Reynolds number, and higher exchange areas) without excessive detriment to film effectiveness. With this design approach it becomes fundamental to analyze and to quantify the extent of the increase of the heat transfer coefficient due to the injection of coolant, that could dramatically affect the performance of the entire cooling system.

The main part of full coverage film cooling studies have been focused on the effectiveness of the protection due to the injection of coolant. This is true for experimental studies as well as for numerical investigations.

Scrittore et al. [7] studied the effects of dilution hole injection on effusion behavior; they found relevant turbulence levels downstream dilution holes, thus leading to an increased spreading of coolant jets. Scrittore et al. [8] measured velocity profiles and adiabatic effectiveness of a full coverage scheme with blowing ratios from 3.2 to 5.0, finding the attainment of a fully-developed effectiveness region at the 15^{th} row and a very low effect of blowing ratio on cooling performance.

To the author's knowledge there are very few data, in literature, dealing with experimental heat transfer evaluation of effusion cooling.

Metzger et al. [9] studied the variation of heat transfer coefficient for full-coverage film cooling scheme with normal holes, they found an augmentation in the local heat transfer with blowing ratio 0.1 and 0.2. Crawford et al. [10] experimentally determined Stanton number for an effusion cooling geometry. Martinez-Botas et al. [11] measured heat transfer coefficient and adiabatic effectiveness of a variety of geometries in a flat plate to test the influence of the injection angle varying blowing ratio from 0.33 to 2.0. They measured the variation of the heat transfer coefficient h respect to the reference case h_0; main result was a maximum of h/h_0 close to the hole and further downstream with highest heat transfer augmentation for 30° injection angle. Kelly and Bogard [12] investigated an array of 90 normal holes finding that the largest values for h/h_0 occur immediately downstream of the film cooling holes and the levels of h/h_0 are similar for the first 9 rows. They explained that this could be due to an increase in the local turbulence levels immediately downstream of the holes, created by the interaction between the cooling jet and the mainstream flow. Another reason could be the creation of a new thermal boundary layer immediately downstream of the cooling jets.

In the open literature none of the previous studies investigated the effect that an high blowing ratio has on both heat transfer coefficient and net heat flux reduction. As reported by Kelly and Bogard [12], increases in heat transfer coefficient due to high blowing ratios could potentially be superceded by increases in heat transfer coefficient due to high mainstream turbulence, it is important to investigate the effects of high blowing ratios on the heat transfer performance of a cooled liner. More recently Facchini et al. [13, 14] measured adiabatic effectiveness and heat transfer coefficient at variable blowing ratios on a real engine cooling scheme to evaluate the combined effects of slot, effusion and a large dilution hole.

Most of effectiveness and heat transfer studies were conducted in presence of uniform flow conditions at inlet. Actually gas turbine combustors operate with swirled flow which helps to promote better air-fuel mixing and to induce a recirculatory flow in the primary zone. Therefore becomes interesting to investigate the effects of non-uniform flow conditions on wall liner convective heat transfer. Ekkad et al. [15, 16] performed experimental and numerical investigations on liner wall convective heat transfer coefficient in a can and annular combustor respectively, in presence of an inlet swirled flow-field. They observed that the peak on the local heat transfer is related to the swirled flow that impinges on liner wall close to the inlet of the combustor.

In the present study, further to preliminary CFD simulations on the real engine scheme, a backward facing step was installed in order to simulate the point of impact of the swirled main flow over the liner. The presence of the step leads to a non-uniform inlet velocity profile and a recirculating area in the first part of the cooled surface.

The effects of a separation region behind a backward facing step was studied both on flowfield and on the heat transfer. B. F. Armaly et al. [17] performed Laser Doppler measurements of velocity distribution downstream a single backward facing step, they found a strong dependence of the length of recirculating area with Reynolds number and cross-section ratio of the channel before and behind the step. Lee and Mateescu [18] investigated experimentally and numerically the length of flow separation and reattachment downstream a backward facing step using a hot wire anemometry and multi-element-hot-film sensors array (MHFS) finding good agreement between experimental and numerical results.

Several authors studied the local convective transfer in the case of the separated and reattaching flows. Shiskov et al. [19]

and Vogel and Eaton [20] studied the convective transfer in the recirculation zone. They showed, in the case of a turbulent flow, that the thickness of the boundary layer upstream of the separation plays a very significant role in the maximum of the heat transfer value: the maximum Nusselt number Nu_{max} increase with the thickness of the boundary layer. Pozarlik et al [21] studied the heat transfer in a recirculating zone at steady state and oscillating conditions. Heat transfer coefficient was founded minimum in the recirculating area and maximum after around five times of step height for steady state measurements.

In the present study, adiabatic effectiveness, heat transfer coefficient and NHFR in presence of a recirculating area generated by a backward facing step are investigated and compared with the experimental results performed on the same cooling geometry, but without the step [13, 14].

The present work was developed within the European Integrated Project NEWAC (NEW Aero engine Core concepts), which focuses on the definition of innovative core engine concepts in order to meet the 2020 ACARE (Advisory Council of Aeronautical Research in Europe) emissions targets [22].

NOMENCLATURE

A	Cross section	$[mm^2]$
BR	Blowing Ratio	$[-]$
d	Effusion hole diameter	$[mm]$
D	Dilution hole diameter	$[mm]$
H	Step height	$[mm]$
HTC, h	Heat transfer coefficient	$[W/(m^2K)]$
k	Thermal conductivity	$[W/(mK)]$
L	Hole length	$[mm]$
$\dot{m}$	Mass flow rate	$[kg/s]$
p	Pressure	$[Pa]$
$plain$	Test without recirculating area	$[-]$
$\dot{q}$	Heat flux	$[W/m^2]$
s	Slot lip thickness	$[mm]$
$step$	Test with recirculating area	$[-]$
S_x	Streamwise pitch	$[mm]$
S_y	Spanwise pitch	$[mm]$
t	Slot height	$[mm]$
T	Temperature	$[^\circ C, K]$
VR	Velocity Ratio	$[-]$
x	Abscissa along the plate	$[mm]$
y	Spanwise location	$[mm]$

Greeks

α	Effusion holes injection angle	$[deg]$
η	Effectiveness	$[-]$
θ	Dimensionless temperature	$[-]$
ρ	Density	$[kg/m^3]$

Subscripts

0	Reference
aw	Adiabatic wall
c	Coolant
$conv$	Convection
eff	Effusion
$main$	Mainstream flow
max	Maximum value
ov	Overall
sl	Slot
w	Wall

Acronyms

FEM	Finite Element Method
$ICAO$	International Civil Aviation Organization
LDI	Lean Direct Injection
LPP	Lean Pre-mixed Pre-vaporized
$NEWAC$	NEW Aeroengine Core concepts
$NHFR$	Net Heat Flux Reduction
$PERM$	Partially Evaporated & Rapid Mixing
PIV	Particle Image Velocimetry
$PMMA$	Poly-Methyl Methacrylate
PVC	Polyvinyl Chloride
TLC	Thermo-chromic Liquid Cristal

EXPERIMENTS

Experimental apparatus

The experimental survey was performed at the Energy Engineering Department "Sergio Stecco" of the University of Florence. Final aim of this activity is the evaluation of the heat transfer coefficient, the adiabatic effectiveness and the NHFR over a specific cooled combustor liner in presence of a recirculating area generated upstream the slot coolant injection by means a backward facing step. These results are compared with measurements performed on the same test sample without the backward facing step and reported by Facchini et al. [14]. The test rig (depicted in the scheme of Fig. 1) consists of an open-loop suction type wind tunnel which allows the complete control of three separate flows: the hot mainstream, the slot coolant and the effusion coolant flows.

The vacuum system is composed by four rotary vane vacuum pumps: two with a capacity of $900\,m^3/h$ each and two of $300\,m^3/h$ each. The two large pumps are dedicated to the extraction of the mainstream mass flow, while the small ones to the annulus recirculating flow. The mainstream flow rate is set up by guiding the large pumps speed and using a calibrated orifice located before the heater, at the beginning of the wind tunnel.

Slot and effusion flow rates are set up throttling two separated valves, while the annulus internal crossflow is set up guiding the small vacuum pumps speed.

The mass flow rate is measured in four different points of the flow system: according to the standard EN ISO 5167-1, two

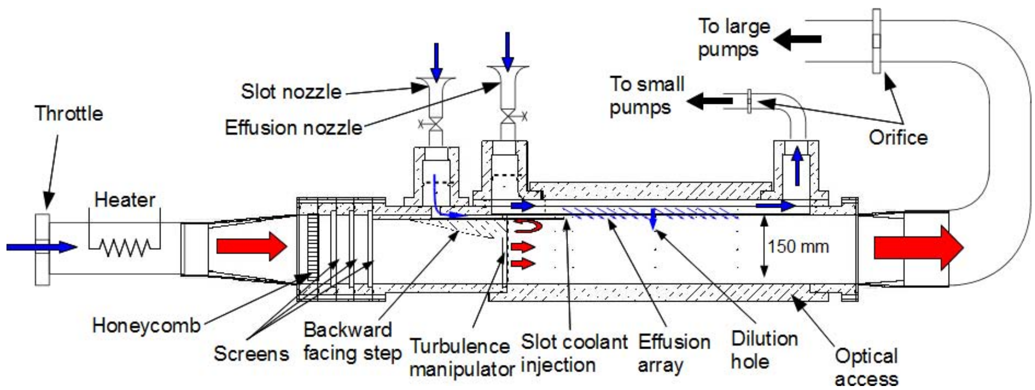

Figure 1. TEST RIG SCHEME

orifices measure the flow rate blown by each couple of pumps, while two nozzles measure the slot and the effusion mass flow rates. Slot coolant flow rate was set up at $0.007 kg/s$ whereas effusion flow rate was varied from $0.025 kg/s$ to $0.090 kg/s$ to reach the desired blowing ratio values.

Wall temperature was measured by means wide band TLC *30C20W* supplied by Hallcrest and active from $30°C$ to $50°C$. Crystals are thinned with water and sprayed with an airbrush on the tested surface after the application of a black background paint. TLC have been calibrated in the same optical condition of a real test, moreover the calibration has been checked directly on the test article before each experiment.

Two pressure scanners Scanivalve® DSA 3217 with temperature compensated piezoresistive relative pressure sensors measure the pressure in 32 different locations with a maximum accuracy of $6.9\,Pa$. Several T type thermocouples connected to a data acquisition/switch unit (HP/Agilent®34970A) measure the mainstream temperature, the coolant temperature and other temperatures inside the model. A digital camera (Sony®DFW-X710) records a sequence of color bitmap images ($1024 \times 768 pixel - 15\,fps$) from the TLC painted surface on a PC (IEEE-1394 standard). The illuminating system (Shott-Fostec®KL1500 LCD) uses an optical fiber goose-neck to ensure a uniform illumination on the test surface and it allows to keep both color temperature and light power constant.

The main channel has a constant cross section of $150 \times 100\,mm^2$ and is $1000\,mm$ long in correspondence of the measurement zone. In the first part of the channel the mainstream flow passes through an honeycomb and then three screens that allow to set an uniform velocity profile. A $50.0\,mm$ high backward facing step was installed $125mm$ upstream the slot, in order to generate a recirculating area approximately on the 6^{th} row of effusion holes. The height and the length of the step was chosen after a series of preliminary CFD simulation in order to simulate the point of impact of the swirled main flow over the liner. A $6.0\,mm$ square hole grid (hole pitch $7.6\,mm$, plate thickness $0.7\,mm$) is placed immediately downstream the step as to set turbulence level at $x/S_x = 0$ around 5% with a macroscopic length scale of $2.8\,mm$, according to correlations proposed by Roach [23].

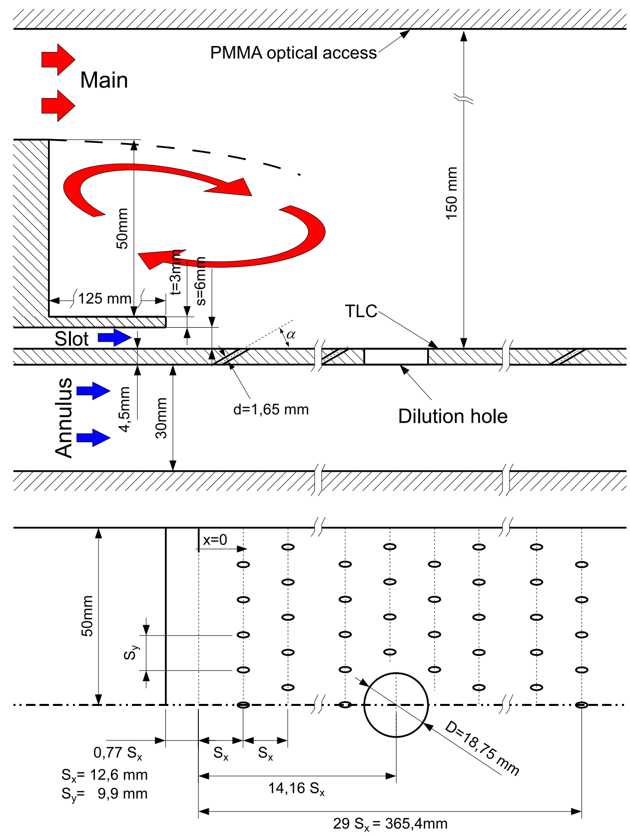

Figure 2. LINER GEOMETRY

The test article is completely made of transparent PMMA, thus allowing the required optical access for TLC measurements and for future PIV measurements, the effusion plate only was made of PVC.

Geometry

Figure 2 reports a sketch of the scaled test article, with the main dimensions of the liner cooling system. The slot coolant is injected in the mainstream from a 6.0 mm height channel, with a lip thickness of 3.0 mm. The effusion array and the dilution hole are fed by an annulus with a rectangular cross-section 30.0 mm high and 120 mm wide; in the end of the annulus there is a plenum connected to the vacuum so as to control the desired rate of recirculating flow.

The effusion geometry consists of a staggered array of 270 circular holes ($d = 1.65\,mm$), with an inclination of $\alpha = 30°$, drilled in a 4.5 mm thick PVC plate and with a length to diameter ratio of $L/d = 5.5$. The spanwise and the streamwise pitches are respectively $S_y = 9.9\,mm$ and $S_x = 12.6\,mm$. The first row is located $22.25\,mm$ $(1.77 S_x)$ after the slot injection, while the last row 375 mm downstream. In order to improve readability of the final results, the origin of the coordinate system ($x = 0$) was set in order to have $x/S_x = 1$ at the first row and than $x/S_x = 29$ in the last row; so, the slot injection is located at $x/S_x = -0.77$. The dilution hole, with $D = 18.75\,mm$, is located immediately after the 14^{th} row, at $x/S_x = 14.16$.

Measurements and test conditions

Two different experimental procedures were done to evaluate heat transfer coefficient and effectiveness over the effusion plate.

Heat transfer measurements Heat transfer coefficients were determined by a steady state technique, measuring wall temperatures from an heated surface, using TLC paint.

The surface heat flux is generated by Joule effect with a 25.4μm thick Inconel Alloy sheet, fed by a DC power supply (Agilent®N5763A) connected to the Inconel sheet through two copper bus bars fixed on lateral extremities of the test plate.

The heat transfer coefficient of the mainstream flow is defined as:

$$HTC_{main} = \frac{\dot{q}_{conv}}{T_w - T_{aw}} \tag{1}$$

where the adiabatic wall temperature T_{aw} represents the surface temperature of a perfectly insulated wall. As mainstream Mach number is low, recovery effects can be neglected hence mainstream temperature, measured with three thermocouples inside the main channel, is used as adiabatic wall temperature. T_w is measured by means of TLC while $\dot{q}_{conv}$ represents the heat rate exchanged by convection between the effusion plate and the mainstream flow. The evaluation of $\dot{q}_{conv}$ was obtained implementing a procedure based on a complete 3D FEM simulation of both the surface heat generation and the heat conduction through the plate that has been already explained in a previous work [14].

Heat transfer experiments have been carried out with coolant and mainstream flows at ambient temperature. The mainstream absolute pressure was kept constant at about $p_{main} = 50000\,Pa$, while coolant pressure was varied in order to ensure the desired values of coolant velocity. Pressure ratio between coolant and mainstream was varied from $p_c/p_{main} = 1.03$ to $p_c/p_{main} = 1.27$.

Main investigation parameters, BR and VR for both slot and effusion flows, are defined as follows:

$$BR_{eff} = \frac{\dot{m}_{eff}/\pi\left(270\,d^2/4 + D^2/4\right)}{\dot{m}_{main}/A_{main}} \tag{2}$$

$$VR_{eff} = BR_{eff}\,\frac{\rho_{main}}{\rho_{eff}} \tag{3}$$

$$BR_{sl} = \frac{\dot{m}_{sl}/A_{sl}}{\dot{m}_{main}/A_{main}} \tag{4}$$

$$VR_{sl} = BR_{sl}\,\frac{\rho_{main}}{\rho_{sl}} \tag{5}$$

where $\dot{m}_{eff}$ is the mass flow rate that flows through the effusion plate and the dilution hole; A_{main} is the mainstream channel cross-section ($150 \times 100\,mm^2$); A_{sl} is the slot cross-section ($6 \times 100\,mm^2$).

Eight different experiments have been performed, starting with the first one without coolant injection and then varying both slot and effusion blowing ratio, in order to simulate values close to engine operating conditions. For a better comprehension of slot and effusion influence on the protection performance, some tests have been performed activating the only slot coolant at $BR_{sl} = 1.3$ or the only effusion flow at $BR_{eff} = 3-5-7$. Then the two cooling systems have been tested together keeping constant the slot mass flow while varying the effusion one.

Adiabatic effectiveness measurements The measurement of the effectiveness over the effusion plate consists of a steady state test realized with assigned flow conditions. Starting from the definitions:

$$\eta_{aw} = \frac{T_{main} - T_{aw}}{T_{main} - T_c} \tag{6a}$$

$$\eta_{ov} = \frac{T_{main} - T_w}{T_{main} - T_{c,in}} \tag{6b}$$

Three thermocouples, normal to the flow, located one pitch upstream the slot injection acquire mainstream temperature T_{main}. Three more probes are dedicated to coolant flow and are inserted

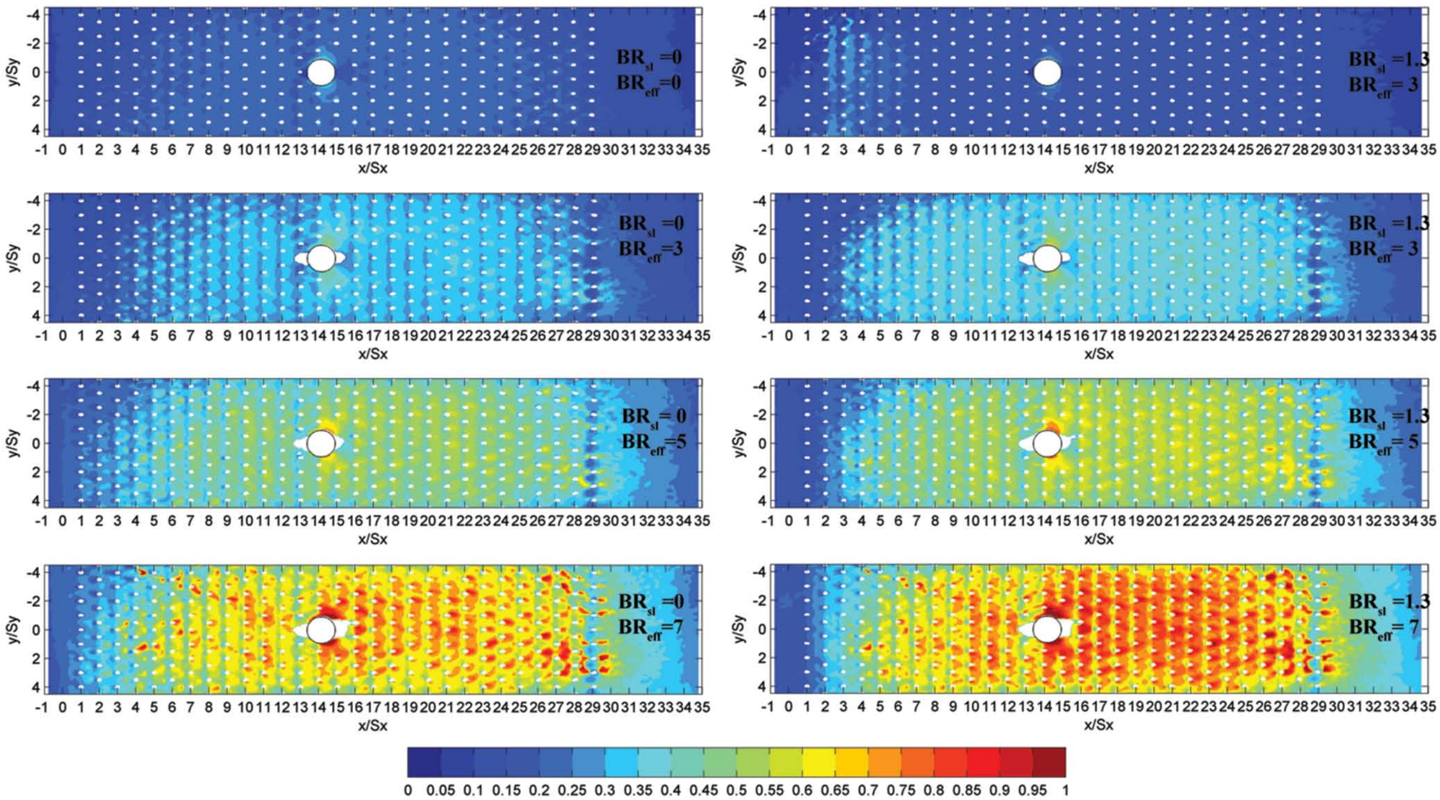

Figure 3. HEAT TRANSFER COEFFICIENT MAPS

into the annulus, at $x/S_x = 0, 14, 29$ and one probe is located inside the slot channel at $x/S_x = -1$.

Likewise heat transfer measurements, T_w is measured by means of TLC, while T_{aw} was evaluated taking into account the thermal fluxes across the plate, due to the coolant flow inside the holes and on the cold side of the effusion surface, with a post-processing procedure based on a 1-D approach, already described in [13].

Effectiveness measurements have been carried out with coolant flows at ambient temperature and mainstream one at about 350K. Seven different experiments have been performed varying both slot and effusion velocity ratio, in particular pressure ratio between coolant and mainstream was varied from $p_c/p_{main} = 1.06$ to $p_c/p_{main} = 1.25$. The recirculating flow of the annulus was set at the same value of the effusion coolant one, so a half of the inlet coolant is delivered from the nozzle to the perforated plate, while the other half is blown by the two small vacuum pumps.

All the tests are run after steady conditions are reached by all the measured quantities: flow rates, pressures and temperatures. These measurements are performed all at once while recording with the camcorder for $10\,s$, afterward, an average value is used for the calculations.

The uncertainty analysis was performed following the standard ANSI/ASME PTC 19.1 [24] based on the Kline and McClintock method [25]. Temperature accuracy is $\pm 0.5\,K$, differential pressure $\pm 6.9\,Pa$, mass flow rate $\pm 2-3\%$ and consequently the maximum relative error referred to the heat transfer coefficient calculation is $\pm 10\%$ and ± 0.05 in measuring the effectiveness. In order to keep uncertainties as low as possible, a great attention was payed to TLC calibration that is the main source of uncertainty in evaluating T_w.

RESULTS

Heat transfer

Figure 3 shows heat transfer coefficient maps for the eight performed experiments.

In the white areas close to the dilution hole, HTC was not measured because of low heat generation which leads to a wall temperature below the activation range of TLC. Low HTC values located at $x/S_x = 29$ are due to a manufacturing defect on the Inconel heating foil that causes a local higher heat generation. HTC values on the maps are expressed in terms of (HTC/HTC_{max})

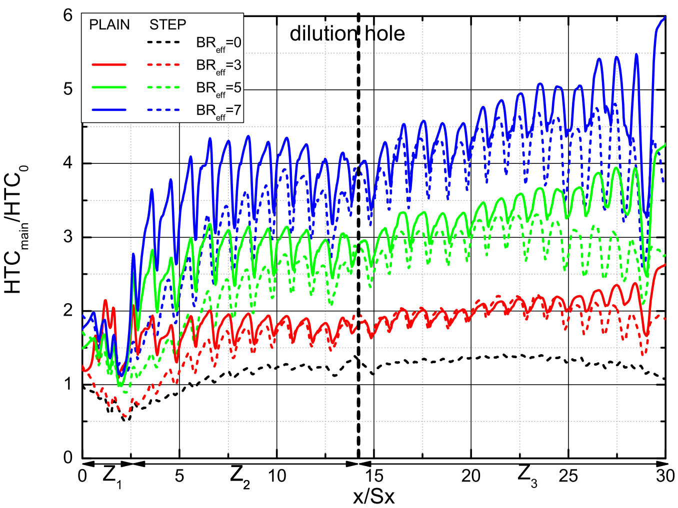

Figure 4. SPANWISE AVERAGED HTC WITH ONLY EFFUSION COOLANT

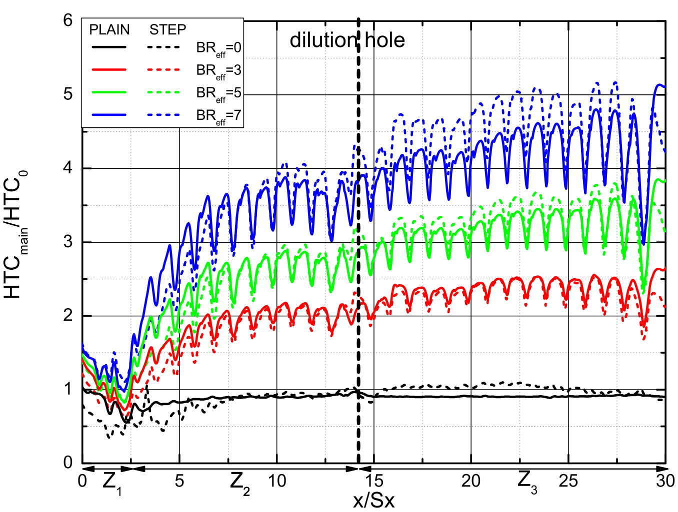

Figure 5. SPANWISE AVERAGED HTC WITH SLOT AND EFFUSION COOLANT

where HTC_{max} represent the maximum evaluated value (i.e. for $BR_{eff} = 7$). According to Facchini et al. [14], heat transfer maps show that HTC increase is mainly related to the effusion blowing ratio, so maximum values of HTC are measured at $BR_{eff} = 7$. In general, there is a rapid growth of the heat transfer up to the 14^{th} row, than HTC values slight increase due to the coolant injected in the main channel that increases the Reynolds number. The effect of coolant flow coming out from effusion holes is the generation of a peak of HTC immediately downstream the holes, as shown in spanwise averaged values.

Figures 4 and 5 report the comparison between spanwise averaged values of HTC_{main}/HTC_0 respectively with only effusion coolant and in presence of slot and effusion coolant injection with and without the recirculating area. HTC_0 represents the heat transfer measured in total absence of coolant injection from slot and effusion in plain configuration, so without the backward facing step. This experiment was used as reference to compare plain and step measurements, with and without slot coolant flow. To better point out different trends of HTC between *plain* and *step* (i.e. without and with backward facing step respectively), the whole investigation area was divided in three zones: the slot affected zone Z1 ($x/S_x < 3$), the increasing zone Z2 ($3 < x/S_x < 14$) and the fully developed zone Z3 ($x/S_x > 14$), located downstream the dilution hole.

Looking at Figure 4, in the experiment without effusion coolant injection ($BR_{sl} = 0 - BR_{eff} = 0$) is clearly visible the effect due to the step. Up to the 5^{th} there is an HTC reduction, then a slight increase up to the 10^{th} row. This point corresponds to the theoretical reattachment point of the recirculation generated by the backward facing step as the x/H ratio (i.e. streamwise abscissa over step height) is about 5.5.

With the effusion coolant injection only ($BR_{sl} = 0 - BR_{eff} = 3 - 5 - 7$) the effect of the recirculating area is a decrease of HTC up to the dilution hole (Zone 1 and Zone 2), while there are no significant effects in the downstream area (Zone 3) up to $x/S_x = 25$. Further downstream the HTC reduction which is present also in the only main flow experiment is probably due to a bias error in measurement chain. Looking again to the first rows, it is important to underline that without the step the absence of slot coolant flow generates local recirculating area. This small recirculation strongly increases heat transfer coefficient in Zone 1 and 2 especially in comparison to the large recirculation generated by the backward facing step.

Looking at the experiment with slot coolant only ($BR_{sl} = 1.3 - BR_{eff} = 0$ - Fig. 5) the presence of the recirculating area leads to lower HTC values only in the first rows, while more downstream there are no significant differences with the *plain* test. Finally, the HTC increase generated by the combined injection of slot and effusion coolant ($BR_{sl} = 1.3 - BR_{eff} = 3 - 5 - 7$) is not affected by the mainstream flow recirculation as there are no large differences between *plain* and *step* experiments. Only at the higher BR_{eff} values slightly higher HTC were measured in *step* case; such difference is visible starting from the 10th row only. The interaction between the recirculating main flow and the coolant from the dilution hole can be a cause of augmentation of the HTC, too. However such results highlight that in presence of slot and effusion coolant injections, the large recirculating area does not significantly influence the heat transfer distribution except that in Zone 3.

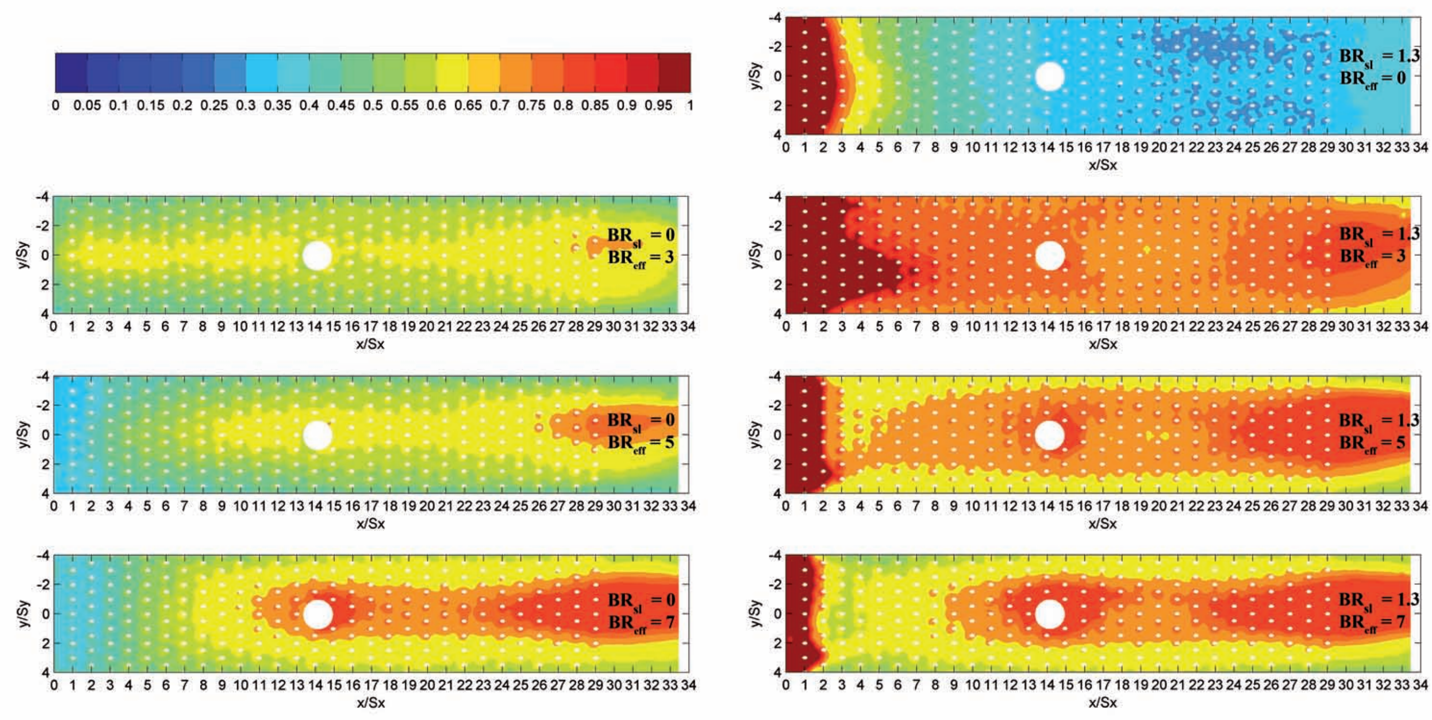

Figure 6. OVERALL EFFECTIVENESS MAPS

Overall and Adiabatic Effectiveness

Figure 6 reports the overall effectiveness maps, calculated with equation 6b.

The comparison between the maps with and without coolant from the slot clearly shows the significant effect of the slot in-

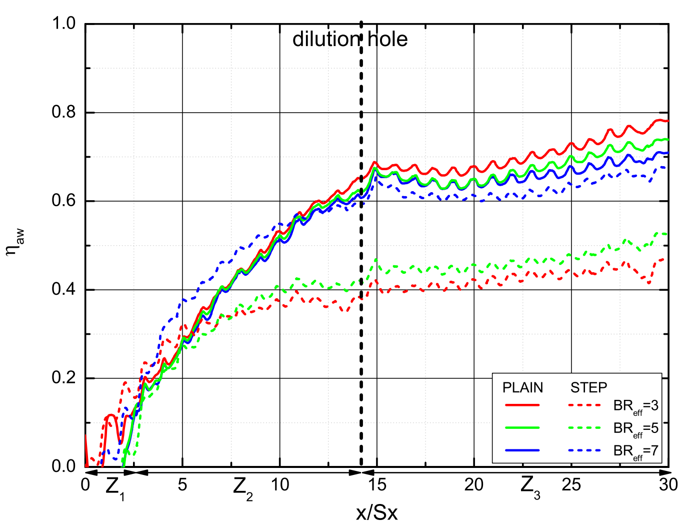

Figure 7. SPANWISE AVERAGED η_{aw} WITH ONLY EFFUSION COOLANT

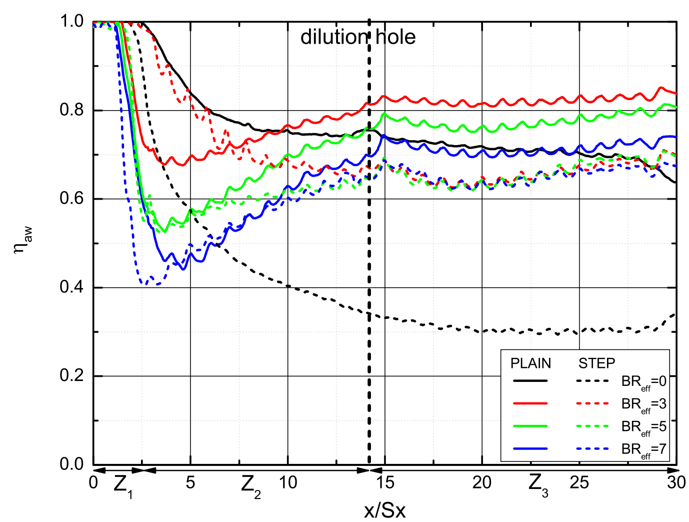

Figure 8. SPANWISE AVERAGED η_{aw} WITH SLOT AND EFFUSION COOLANT

jection. This very important feature do not fail in presence of the backward facing step, in particular in Zone 1 where effusion coolant does not guarantee a sufficient protection from hot mainstream flow.

Figures 7 and 8 report adiabatic effectiveness spanwise averaged values for *plain* and *step* cases without and with slot coolant injection respectively.

Starting from Figure 7, the effect of the recirculation is negligible in Zone 1. In Zone 2, from the 5^{th} row of effusion holes which is the theoretical reattachment point of the separated flow, adiabatic effectiveness trends are different between *plain* and *step* cases. For $BR_{eff} = 3-5$ the main flow recirculation reduces the protection capability of the effusion cooling. This trend is extended also over the whole Zone 3, where the interaction between mainstream and coolant flow causes an adiabatic effectiveness drop of about $0.20-0.25$. At $BR_{eff} = 7$, the recirculating area has no effects, hence effectiveness values are comparable with *plain* case. So, at lower effusion blowing ratios, the step generates a dependence of the effectiveness by the BR_{eff}, with lower values than in *plain* configuration.

In Figure 8 the effects of the recirculating area in presence of coolant from the slot are shown. With only slot coolant injection ($BR_{sl} = 1.3 - BR_{eff} = 0$), the $eta_{aw} = 1$ area is slightly reduced in Zone 1 between *plain* and *step* cases. In Zone 2 there is a faster effectiveness decay because the recirculation promotes the mixing between the hot main flow and the coolant, then in Zone 3 it slightly decays and it is about 0.4 lower than in the plain case.

Activating the effusion cooling, different trends can be observed varying the blowing ratio. For *plain* tests, up to the 5^{th} row, an increase of the BR_{eff} leads on a shift of the effectiveness values. This feature is nullified in presence of the recirculating area that interacts in a different way depending on the effusion blowing ratio in Zone 2. In particular at $BR_{eff} = 3$, the recirculation seems to have beneficial effect on the protection up to the 6^{th} row, then coolant and main flow begin to interact producing a reduction of the effectiveness. For this BR the curve decrease mono-tonically as the "only slot" one. $BR_{eff} = 5$ and 7 step experiments are more similar to the plain ones, where a minimum is reached around the 4^{th} row and an increase of BR leads to an effectiveness decrease. A final difference with the plain case regards the Zone 3. After the dilution hole, the spanwise averaged adiabatic effectiveness values are not affected by the Blowing Ratio, while in the plain case there was an effectiveness shift also in this final part of the plate.

NHFR

As already done for heat transfer and effectiveness measurements, in this section plain (reprted in [14]) and *step* cases are compared in terms of heat flux reduction.

NHFR (Net Heat Flux Reduction) is a commonly used parameter to evaluate the reduction of heat flux across a cooled surface. This parameter was defined by Sen et al. [26] as:

$$NHFR = 1 - \frac{\dot{q}}{\dot{q}_0} = 1 - \frac{HTC_{main}}{HTC_0}(1 - \eta_{aw}\theta) \tag{7}$$

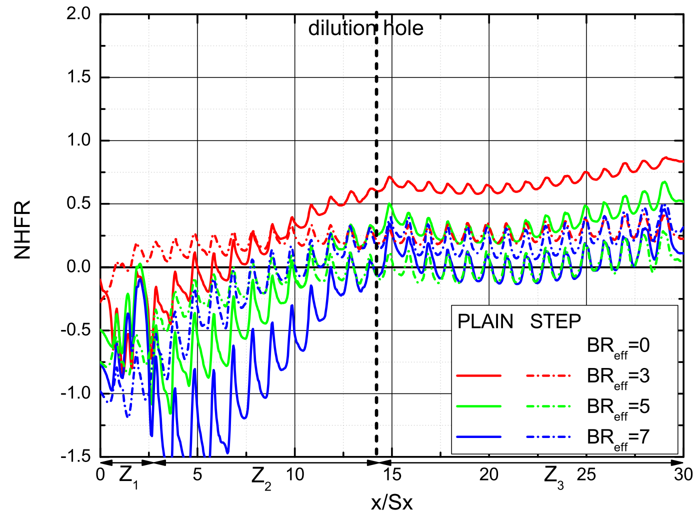

Figure 9. SPANWISE AVERAGED NHFR WITH ONLY EFFUSION COOLANT

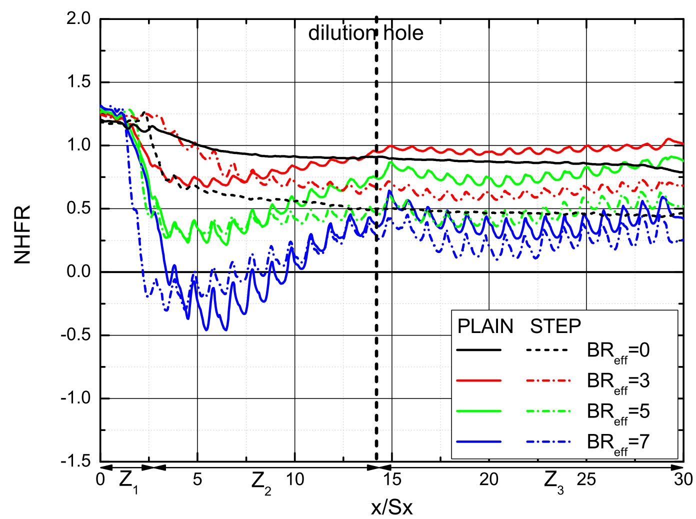

Figure 10. SPANWISE AVERAGED NHFR WITH SLOT AND EFFUSION COOLANT

where θ represent the dimensionless temperature defined as:

$$\theta = \frac{T_{main} - T_c}{T_{main} - T_w} \tag{8}$$

In the evaluation of NHFR, HTC_{main}/HTC_0 values are combined with the adiabatic effectiveness values η_{aw} defined in Eq. 6a.

About the dimensionless temperature θ, in the open literature [26–29] NHFR was mainly used to evaluate turbine endwall and blades cooling systems where $\theta = 1.5-1.6$. In the present study the dimensionless temperature has been calculated using temperature values representative of a combustor, obtaining $\theta = 1.2$ [14].

Figures 9 and 10 reports the NHFR spanwise averaged values of plain and step measurements, without and with slot coolant respectively.

With only effusion coolant injection, the recirculation seems to be beneficial on the NHFR up to the 10^{th} row, in particular for $BR_{eff} = 3$, there is even a positive NHFR values from the first rows despite low effectiveness values. On the contrary, in Zone 3 presence of the step is detrimental for the NHFR, with the exception of $BR_{eff} = 7$ where there is slight higher heat flux reduction in *step* case.

In presence of the slot coolant (Figure 10), that represents the engine configuration, the effect of the recirculating area is negligible only at $BR_{eff} = 7$ on the whole cooled surface. Without effusion coolant injection $BR_{eff} = 0$, despite of very low effectiveness value in *step* case, slot coolant affords to guarantee an heat flux reduction. For $BR_{eff} = 3$ there is a benefit up to the $6^{th} - 7^{th}$ row, then recirculation causes a decrease of NHFR anyway with positive values. This trend is strongly related to the adiabatic effectiveness values. For $BR_{eff} = 5$, no noticeable effects are observed up to the 8^{th} row then NHFR values are lower than in the plain case and quite constant at about 0.5. Finally, the effect of the recirculation downstream the dilution hole (i.e Zone 3) is to shift down NHFR values, without changing the slope of the curves.

Conclusions

An experimental investigation on a real combustor liner cooling system in presence of a recirculating area has been performed.

The cooling scheme consists of a slot injection, followed by a flat plate with 29 effusion rows and a single large dilution hole located at the 14^{th} row. A backward facing step was installed to generate the recirculating area close to the inlet region.

Experiments have been done in order to measure local heat transfer coefficient, adiabatic effectiveness and NHFR, as well as to compare results between measurements with and without the backward facing step. Three different effusion blowing ratios ($BR_{eff} = 3-5-7$) were tested to cover typical engine values while slot coolant was activated at $BR_{eff} = 1.3$. Measurements have been carried out in a scaled model using a steady state technique with wide band thermo-chromic liquid crystals.

About the heat transfer coefficient measurements, the most relevant result of the experimental survey is that the effect of the backward facing step is negligible as there are no measurable differences with the plain configuration. Such result highlights that the wall heat transfer phenomena are driven by the slot and the effusion cooling flow, and not by the mainstream flow.

Regarding the effectiveness measurements, without slot coolant injection, the performance of the effusion cooling are heavily reduced by the backward facing step at $BR_{eff} = 3-5$ while at $BR_{eff} = 7$ the recirculation has not effects. In presence of slot coolant injection the main effects of the step is downstream the dilution hole where adiabatic effectiveness curves converge all to a quite constant value of $\eta_{aw} = 0.65$. So the recirculation nullifies the effects of the effusion blowing ratio on the effectiveness. In general, for the *step* case, seems that the recirculation promotes hot main flow and coolant mixing leading to lower values of effectiveness.

Finally, combining HTC_{main}/HTC_0 with η_{aw} the net heat flux reduction (NHFR) has been evaluated. Without slot coolant injection the presence of the step seems to be beneficial on the NHFR up to the 10^{th} row, in particular for $BR_{eff} = 3$, there is even a positive NHFR values from the first rows. On the contrary, downstream the dilution hole the presence of the step is detrimental for the NHFR, with the exception of $BR_{eff} = 7$ where there is slight higher heat flux reduction in *step* case. In presence of the slot coolant, the effect of the recirculating area is negligible only for $BR_{eff} = 7$ on the whole cooled surface. At $BR_{eff} = 3$ the trend is strongly related to adiabatic effectiveness values; there is a benefit up to the $6^{th} - 7^{th}$ rows, then the recirculation causes a decrease of NHFR anyway with positive values. At $BR_{eff} = 5$, no noticeable effects are observed up to the 8^{th} row then NHFR values are stabilized at about 0.5. Finally, the effect of the step downstream the dilution hole is to shift down NHFR values without changing the slope of the curves.

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