PREDICTION AND AUGMENTATION OF NOZZLE GUIDE VANE FILM COOLING HOLE PRESSURE MARGIN

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ABSTRACT

To ensure adequate cooling and avoidance of hot gas ingestion, a high pressure turbine nozzle guide vane must maintain a safe pressure margin by which the film coolant static feed pressure exceeds the hot gas total pressure. This pressure margin is lowest for cooling holes near the stagnation region, especially near the coolant inlet. This study investigates an insert device which increases the pressure margin in these ingestion risk regions by altering the coolant passage geometry, potentially allowing engine performance gains via reduced combustor pressure loss requirements. Seven parametrically varied inserts are compared within a pressure-tapped, cooled vane model, and design rules are suggested. For correctly designed inserts, pressure margin results show significant improvement in the ingestion risk region, without causing ingestion risks elsewhere. An analytical model of combined converging, diverging and prismatic coolant channels is validated by experimental data to be capable of accurately predicting coolant pressure margins.

KEYWORDS

FILM COOLING, PRESSURE MARGIN, HOT GAS INGESTION

NOMENCLATURE

CTI Combustor Turbine Interactions
\( \dot{m} \) Mass Flow Rate
NGV Nozzle Guide Vane
PM Normalised hole Pressure Margin
PS, SS Pressure Side and Suction Side
\( \bar{m}_{c}, \bar{m}_{T} \) Channel and passage total \( \bar{m} \)
\( J \) Momentum flux ratio
\( n, N \) Covered and total cooling holes
\( L, H \) Coolant passage and insert lengths
\( A, A_{T} \) Channel and passage areas
\( P_{i} \) Normalised internal pressure
\( \rho, \rho_{c}, R \) Coolant density, specific gas constant
\( \nu_{c}, \nu_{\text{throat}} \) Mainstream NGV inlet, throat velocity
\( u_{c}, u_{c}^{*} \) Coolant velocity
\( n_{c, i/e} \) Internal/external pressure
\( p_{0/sc}, T_{0/sc} \) Coolant total/static pressure, temperature
\( D_{H} \) Hydraulic diameter
\( \%m_{NGV} \) NGV coolant percentage of throat \( \dot{m} \)
\( s \) Fractional spanwise distance
\( K_{L}, f \) Loss coefficient, pipe friction factor
\( x, C_{x} \) Axial distance, axial chord
INTRODUCTION

Film cooling of high pressure turbine nozzle guide vanes (NGVs) is essential to maintaining their structural integrity at the extreme combustor exit temperatures of any modern jet engine gas turbine. The effusion of film coolant without potentially destructive ingestion of hot mainstream gases requires that the feed coolant total pressure be adequately in excess of the NGV external static pressure at all engine conditions. This excess is commonly known as the ‘pressure margin’, and has significant influence on engine performance and efficiency, via its consequences for parasitic combustor pressure loss and NGV coolant mass flow rate ($\dot{m}$). For a given NGV film coolant $\dot{m}$, a pressure margin increase permits a reduction in combustor pressure loss, which has historically been limited by NGV pressure margin considerations. Using the wind tunnel shown in Fig. 1., the present study investigates the performance of a device which alters the aerodynamic geometry of the NGV leading edge coolant feed passage so as to increase the margin independently of the coolant $\dot{m}$, potentially allowing a reduction in this rate, in the combustor pressure loss, or a combination of both. The tested device is referred to as an ‘insert’, since it is intended to be attached to the inside of an airfoil coolant passage (rather than cast as part of the airfoil), in order to improve the pressure margin at the cooling holes most at risk of ingestion. Such a device could also potentially be employed to allow reshaping of an airfoil external profile, where this may otherwise have been compromised by the need to contour the coolant passage while

Figure 1: Experimental wind tunnel showing bellmouth inlet, combustor simulator, test section, tailboards and central vane with cooling hole locations and an insert at coolant inlet

Figure 2: NGV leading edge, with and without a half-span insert, with internal pressure contours from CFD (%$\dot{m}_{NGV} = 9.9$). A static pressure rise inside the insert is evident. Pressures relative to the stagnation line ($P_{st}$=0)
maintaining allowable metal thickness. One of the seven insert geometries investigated in this study is shown in the lower image of Fig. 2 in its operational position within the NGV leading edge model. The inserts could also find application in other cooled parts. As further discussed in the theory section, their underlying principle is to slow the coolant flow in the region of the ingestion risk hole inlets, increasing the local recovery of dynamic pressure and hence the hole feed static pressures.

In practice, slowing coolant flow with an insert will reduce the local internal heat transfer. This could be partially offset by increased turbulation on the coolant passage wall to augment heat transfer coefficient. Similarly, the coolant temperature at hole exits will be slightly increased, reducing film cooling effectiveness. This will be partially offset by higher hole pressure margins than would have been achieved without the insert. The use of one or more inserts may represent a calculated design choice to improve engine cycle efficiency at the expense of a local increase in NGV temperature.

**Literature Survey**

There appears to be no previous academic literature proposing increases to airfoil pressure margins, by geometric manipulations or otherwise. Film cooling performance studies usually concern the effects of variations in airfoils, holes, and jet or hot gas flows. While some studies have investigated the performance and cooling jet physics of various coolant supply conditions (e.g. Gritsch et al. (1997), Thole et al. (1997), Walters and Leylek (1997)), little consideration is given explicitly to hot gas ingestion concerns. The studies most closely related to the present one appear to be those of internal cooling schemes which consider the associated coolant total pressure losses. In regions of low ingestion risk, internal losses may be used to reduce film blowing rate. Total pressure loss is inherent to the turbulent friction required to increase solid-to-fluid internal heat transfer, and as such the pressure loss characteristics of conventional techniques (impingement jets, rib turbulators and pin-fins) are well studied. Some studies are particular to leading edges, such as that by Domaschke et al. (2012), who also reviewed the prior studies of rib-turbulated cooling in prismatic passages of triangular cross-sections. There is a recent trend in airfoil internal cooling research to combine and integrate the various existing technologies (Ligrani, 2013). A promising example is the impingement of coolant into wall cavities bridged by pin-fin arrays. In these ‘double-wall’ configurations, there is little additional pressure loss concomitant with the extra heat transfer surface area provided by the pin-fins (Bamba et al., 2008). The complex internal flow field, combined with the use of low blowing ratio, high hole density film cooling, mimics the operation and performance of a porous material undergoing transpiration cooling (Murray et al., 2017). The insert concept of this study is investigated in a purely film-cooled geometry, though it has potential to be integrated with double-wall cooling designs, which could be permitted more aggressive impingement pressure losses as a result.

**PRESSURE MARGIN INSERT THEORY**

**Engine Coolant Mass Flow Rate Matching**

The NGV coolant \( \dot{m} \) in this study is expressed as a percentage of overall throat flow \( \%\dot{m}_{NGV} \), a value which will vary between engines. The percentages calculated in this study are not engine-representative because of the absence of downstream PS film cooling holes and slots. True engine-matching in this respect would also require testing of geometry with a more engine-realistic coolant flow path with varying coolant passage area and curvature, and potentially a variety of internal cooling features. The primary purpose of varying \( \%\dot{m}_{NGV} \) in this study is to examine its influence on the inserts’ cooling and flow attributes.

**Inviscid Analysis**

The following inviscid analysis shows that a sufficiently broad insert can increase its internal static pressure, thus increasing the pressure margin of enclosed holes. The fundamental mechanism is to slow the coolant, increasing the local recovery of dynamic pressure. Consider an inviscid and incompressible analysis (constant coolant total pressure \( p_{0c} \) and density \( \rho_c \)) in which a prismatic coolant passage of constant cross-sectional area \( A_T \) and internal length \( L \) feeds \( N \) cooling holes from
a single end. Let \( s \) represent the fractional spanwise distance from the inlet (0 \( \leq s \leq 1 \)). If the coolant mass flow rate at the inlet is \( \dot{m}_T \) and the coolant is expelled evenly along the span, then the passage mass flow rate at a position \( s \) can be approximated by \( \dot{m}_c(s) \approx (1-s)\dot{m}_T \), so the coolant velocity \( u_c(s) \approx \frac{(1-s)\dot{m}_T}{\rho_c A_T} \). Then the coolant static pressure \( p_{sc}(s) \) within the passage simplifies to eqn. (1).

\[
p_{sc,\text{passage}}(s) = p_{0c} - \frac{1}{2} \rho_c [u_c(s)]^2 \quad \Rightarrow \quad p_{sc,\text{passage}}(s) = p_{0c} - \frac{1}{2} \rho_c (1-s)^2 \left( \frac{\dot{m}_T}{A_T} \right)^2
\]

Consider also the two adjacent channels of length \( H \) with 0 \( \leq H \leq L \), formed by an insert of the same length \( H \), resembling those pictured in Figs. 1 and 2. The passage mass flow rate within the converging channel can be approximated by \( \dot{m}_c(s) \approx (1 - \frac{L}{H})s \frac{\dot{m}_T}{H} \) where \( n \) is the number of holes covered by the insert (i.e. whose inlets are within the converging channel). If the channel's cross-sectional area is \( A(s) \), then the static pressures in this channel \( p_{sc,\text{conv}}(s) \) can be expressed as eqn. (2). Employing eqns. (1) and (2), the condition for \( p_{sc,\text{conv}}(s) > p_{sc,\text{passage}}(s) \) simplifies to inequality (3). At the inlet (\( s = 0 \)), the static pressure within the insert converging channel increases above the prismatic passage static pressure when the proportion of the total passage inlet area occupied by the insert's cross-section exceeds the proportion of included cooling holes. In the results section, an extension of the preceding analysis including total pressure loss estimations is described, which enables hole pressure margins to be predicted with good agreement with the experimental data.

### Insert Design

To determine a profile of minimum allowable cross-sectional area \( A_{min}(s) \), the inviscid analysis above can be repeated with inclusion of a desired minimum pressure margin \( PM_{\text{design}} = p_{sc} - p_{se,max} \), where \( p_{se,max} \) is the maximum (stagnation line) external airfoil static pressure. Since the external pressure around the airfoil varies, the pressure margins between covered hole rows will be different even if the static pressure inside the insert is uniform, but all will be greater than \( PM_{\text{design}} \).

\[
p_{0c} - \frac{1}{2} \rho_c [u_c(s)]^2 \geq p_{se,max} + PM_{\text{design}} \Rightarrow p_{0c} - \frac{1}{2} \rho_c \left[ \left(1 - \frac{L}{H} \right)s \frac{\dot{m}_T}{H} \frac{n}{N} \right] \geq p_{se,max} + PM_{\text{design}}
\]

\[
A_{min}(s) = \left(1 - \frac{L}{H} \right)s \frac{\dot{m}_T}{H} \frac{n}{N} \equiv \sqrt{2\rho_c [p_{0c} - (p_{se,max} + PM_{\text{design}})]}
\]

In this study, the \( PM_{\text{design}} \) was three NGV mainstream inlet dynamic pressures as measured in the test rig (estimated as representative of the equivalent pressure margin in a modern civil jet engine). The other parameters in eqn. (4) were measurable from rig tests on the film-cooled NGV model with no insert attached. These rig pressure margin results demonstrated a clear need to include within the inserts all but the two rearmost suction surface (SS) rows. There is only a slight recovery of dynamic pressure observed as the coolant velocity decreases along the NGV span, suggesting that the ingestion risk has little spanwise dependency. This would also likely be the case for the same coolant passage under engine conditions, since the recoverable coolant dynamic pressure is smaller than the rig case relative to the respective estimated mainstream NGV dynamic pressures. Therefore, inserts should cover all holes of rows at ingestion risk. Nevertheless, since many engines feature double-end fed leading edge coolant passages, inserts of lengths 0.3\( L \), 0.5\( L \) and \( L \) were tested. Tab. 1 summarises the geometric details of the tested designs. For each of the shorter two lengths, three inserts were designed on the basis of \( A_{min}(s) \) profile predictions according to eqn. (4), for \%\( \dot{m}_{NGV} = 4.7, 7.1, 8.4 \), employing the respective NGV \( \dot{m} \) and pressure data collected during rig operation with no insert. Since testing on these designs indicated higher enclosed hole pressure margins for higher inlet areas, only a single full-span insert of generous inlet area was tested. All \( A(s) \) profiles of the tested inserts exceed all three of their relevant \( A_{min}(s) \) profiles at all span positions. All of the inserts

<table>
<thead>
<tr>
<th>Insert</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
<th>G</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A(0)/A_T )</td>
<td>0.22</td>
<td>0.33</td>
<td>0.41</td>
<td>0.34</td>
<td>0.53</td>
<td>0.60</td>
<td>0.89</td>
</tr>
<tr>
<td>( H/L )</td>
<td>0.3</td>
<td>0.3</td>
<td>0.3</td>
<td>0.5</td>
<td>0.5</td>
<td>0.5</td>
<td>1</td>
</tr>
</tbody>
</table>
were tested at $\%m_{NGV} = 4.7, 7.1, 8.4, 9.9$. The risk of diverging channel flow separation increases as the inlet area of the insert increases, or the length decreases, as these changes increase the diffusion rate. The cooling holes just downstream of the end of an insert are at particular risk of ingestion, as they may be located within a flow separation from off this end. Fig. 3 shows a summary of the experimental results more thoroughly conveyed in the results section, which illustrates the beneficial effects of well-designed inserts. Non-dimensional pressure margins $PM$ across a near-stagnation line cooling hole row are shown for all seven inserts plus the datum case of no insert. While experimental data are only available at $s = 0.15, 0.30, 0.75$, it is clear that half-span inserts E and F vastly improve pressure margin near the inlet, while full-span insert G somewhat improves margin for the entire span.

**EXPERIMENTAL APPARATUS AND METHOD**

**Combustor Turbine Interactions Rig**

The Oxford Combustor Turbine Interactions (CTI) rig features a two-passage cascade test section with linear extrusions of a modern engine NGV midspan geometry, and an upstream combustor simulator (Fig. 1). The rig-to-engine scale is 4.45. The mainstream flow is induced by a downstream centrifugal fan, while the plena for combustor simulator dilution ports and endwall film cooling holes are supplied by an additional blower, via a branching pipe system with butterfly valves permitting individual control of the coolant mass flow rates, measured by orifice plate meters. The plenum coolant temperatures are recorded by thermocouples, due to the coolant temperature rise and density reduction through the blower. The core into the simulator is inferred using an upstream pitot tube, which was referenced to the detailed NGV mainstream inlet velocity distribution measured by Cresci et al. (2015a). The NGV throat Reynolds number based upon the chord varies between approximately $6.5 \times 10^5$ for no NGV coolant flow and $7.2 \times 10^5$ for the maximum investigated NGV coolant $m_{NGV}$. Increased NGV coolant $m_{NGV}$ very slightly reduces the induced mainstream $m_{core}$. Rather than compensating by increasing the mainstream fan speed, the vane mainstream inlet dynamic pressure is determined separately for each run, and used to normalise the pressure results from that run. The mainstream $m_{core}$ change is insufficient to significantly affect $\%m_{NGV}$ or coolant momentum flux ratios ($J = \frac{\rho_{NGV} \dot{m}_{NGV}^2}{\rho_{core} \dot{m}_{core}^2}$). The latter are determined at the dilution ports ($J = 9$) and outer and inner endwall film cooling holes ($J = 7$ and 11, respectively), and are representative of those in a modern Rich-Burn, Quick-Mix, Lean-Burn (RQL) annular combustor. This parameter is well established as the most crucial in determining jet-in-crossflow penetration and mixing, and hence the fluid property profiles at the NGVs (Hatch et al., 1992, Holdeman et al., 1996, Barringer et al., 2002).

**Figure 3: Measured SS1 hole row pressure margins for $\%m_{NGV} = 7.1$, all seven tested inserts**
NGV Leading Edge Test Pieces

The central cascade vane features a removable leading edge section, allowing interchange of test pieces suited to different measurements. These are fed coolant from a single end by attachment to the same plenum as feeds the inner endwall film cooling. The coolant supply passage is prismatic along the span and feeds 9 rows of cooling holes, labelled in Fig. 1. All NGV film cooling holes eject parallel with the mainstream flow. The hole inlets and exits are uniformly positioned along the whole span at midspan engine-representative positions. The cooling holes per row, hole diameter and hole pitch are also engine-representative. As the main body of the central NGV is uncooled, it lacks several downstream pressure side (PS) cooling hole rows and a trailing edge slot, which would be present in an engine. Cresci et al. (2015a, 2015b, 2015c) provide more details about the rig and the flow properties approaching the NGV cascade. The two pressure-tapped leading edges used were 3D printed. One features coolant passage internal pressure tappings at fractional span positions \( s = 0.15, 0.30 \), while the other features internal and external surface tappings at \( s = 0.75, 0.50 \), respectively. A prior study on the same cooled geometry found negligible variation between external pressure measurements at \( s = 0.50 \) and \( s = 0.75 \), so the external pressure distribution is considered uniform along the span. The pressure margin inserts were also 3D printed, and their edges were sealed to the inside of the coolant passage with flowable silicone, as shown in Fig. 2. Occasionally, the coolant passage tappings were obstructed by the inserts or by excess silicone, and these were removed from the data as necessary. Additional \( s = 0.50 \) tappings in the downstream central vane section permit external pressure measurements around the rest of the vane. \( s = 0.50 \) tappings in the two half-vanes allow the tailboards downstream of the cascade to be adjusted so that the external pressure distributions mimic those of a periodic, annular cascade.

Experimental Uncertainty Analysis

The uncertainties in the derived quantities mentioned in Tab. 2 are calculated with those propagated by orifice meter, pressure tapping, pitot tube and thermocouple measurements. \( \% \hat{m}_{\text{NGV}} \) values are highly sensitive to adjustment of the relevant butterfly valve, with a maximum variation from the stated target quantities across all runs of 0.09 (which is added to the uncertainty based on orifice plate calculations). Since \( J \) goes with the square of the jet mass flow rates, its adjustment was particularly sensitive, with a maximum variation from the target values of approximately 0.5. These variations are accounted for since all vane pressure data are normalised by simultaneously recorded NGV mainstream dynamic pressures. Calculation of the non-dimensional hole pressure margins \( PM \) requires interpolation of the tapping data to the centres of the hole inlets and exits. The hole exit pressure uncertainty for SS5 and SS6 is considerable, as they reside in the region of greatest pressure gradient on the NGV exterior. This additional uncertainty in \( PM \) is estimated as 0.1, based on the maximum residual of a linear fit through the pressure data points in this region. The normalized internal pressures \( P_i \) are not subject to this additional uncertainty as they represent the pressures measured at tappings. The main contributor to uncertainty in both \( PM \) and \( P_i \) is the cumulative mass flow uncertainty in the estimation of the normalising mainstream dynamic pressures. Their largest percentage uncertainties apply only to low NGV coolant mass flow cases for which some internal pressures approach external pressures.

Table 2: Estimated maximum uncertainties for the derived flow quantities of interest

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Symbol</th>
<th>Range</th>
<th>Maximum uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>NGV coolant ( \hat{m} ) ( % )</td>
<td>%( \hat{m}_{\text{NGV}} )</td>
<td>4.7 - 9.9</td>
<td>0.26 (2.6 - 5.5%)</td>
</tr>
<tr>
<td>Momentum flux ratio</td>
<td>( J )</td>
<td>7 - 11</td>
<td>0.9 (8.2 - 12.9%)</td>
</tr>
<tr>
<td>Vane internal pressure</td>
<td>( P_i )</td>
<td>-0.56 - 50.8</td>
<td>0.18 (0.3 - 32.1%)</td>
</tr>
<tr>
<td>Non-dimensional hole pressure margin</td>
<td>( PM )</td>
<td>2.5 - 61.7</td>
<td>0.28 (0.5 - 11.2%)</td>
</tr>
</tbody>
</table>
EXPERIMENTAL AND COMPUTATIONAL RESULTS

External Pressure Distributions

Fig. 4 shows experimentally measured NGV external pressures for the case of no insert and for all $\%\dot{m}_{NGV}$ values. The external pressures with inserts exhibited negligible departure from these results. Unlike the margin results, these pressures are normalised by the more usual throat dynamic pressure $(\frac{1}{2}pu^2)_{throat}$. This is calculated in two different ways: by excluding the NGV coolant $\dot{m}$ (empty circles) and by including it (filled circles). The small gap where the SS of the removable leading edge NGV piece meets that of the downstream piece causes a small separation region, around which the flow accelerates, causing a pressure drop. For all experiments, tape was sealed over the gap, though little improvement was yielded. For higher NGV coolant $\dot{m}$, the film thickness increases, generating a greater effective NGV blockage, which increases the velocity of the mainstream flow, reducing the NGV external surface absolute pressures, as indicated by the empty circles (the effect of NGV coolant $\dot{m}$ on the mainstream $\dot{m}$ is very small). The effect is most profound on the SS where there are more cooling holes, but it is also present to a minor degree on the PS, except for $0.3 < x/C_x < 0.8$, where there is no clear trend. The trend appears to reverse when normalising by $(\frac{1}{2}pu^2)_{throat}$ including NGV coolant, which increases with the NGV $\dot{m}$. A series of uncooled vane pressure distribution measurements with $(\frac{1}{2}pu^2)_{throat}$ values matching those of each cooled case could be compared to the respective cooled distributions in order to examine the effects of film-mainstream mixing total pressure loss on the pressure distribution. However, this is beyond the scope of the present study.

![Figure 4: Experimental pressures on NGV exterior, with no insert present. Filled/empty circles are normalised by $(\frac{1}{2}pu^2)_{throat}$ including/excluding the NGV coolant $\dot{m}$](image)

Internal Pressure Distributions

The measured internal pressures relative to the maximum measured external static pressure and normalised by the mainstream inlet dynamic pressure $(\frac{1}{2}pu^2)_w$ are shown in Fig. 5. $x/C_x$ describes positions around the cross-section of the coolant passage inner surface (PS negative, SS positive, and $x = 0$ is the vane’s geometric stagnation line). The internal pressure tends to increase with coolant $\dot{m}$ because higher velocity coolant recovers more dynamic pressure as it slows. Considering the experimental data with no insert, a spanwise increase in static pressure due to progressive effusion of coolant is only just apparent for $\%\dot{m}_{NGV} = 9.9$. 

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Figure 5: Experimentally measured static pressures inside the NGV coolant passage for seven pressure margin inserts. $x/C_x < 0$ and $x/C_x > 0$ are the coolant passage wall regions internal to the NGV pressure and suction sides, respectively. Note the vertical axis scale variations in each graph.

**Effects of Overly Broad or Narrow Inserts**

The tested $H/L = 0.30$ inserts illustrate the need to design inserts for low diffusion rate in the diverging channel. The $s = 0.30$ tappings lie just downstream of the inner ends of these inserts (many were prone to blockage by adhesive, causing the irregular pressures just downstream of insert A). Those just downstream of the portion of inner surface covered by insert B experience the same uniform pressure as those farther around the passage surface. By contrast, the pressures measured just behind insert C are substantially lower than those farther around the passage surface, and indeed much lower.
lower than with no insert. This is because insert C has the widest inlet area of the three $H/L = 0.30$ candidates and hence the greatest diffusion rate in the diverging channel, causing a flow separation and low pressure region near its end. The pressure inside insert A is slightly less than in the absence of an insert, indicating that the converging passage is too narrow and accelerates the flow. The effective cross-sectional area will always be less than the ideal inviscid one, because of boundary layers within the converging passage. The restriction of mass flow into the converging channel explains why insert A exhibits the highest static pressures at $s = 0.30$; the associated increase in $\dot{m}$ to the diverging channel increases the recoverable dynamic pressure. Compared to no insert, inserts B and C succeed in significantly increasing the pressures within the inserts, while reducing the pressures outside at $s = 0.15$. Coolant pressures at $s = 0.75$ are only very slightly reduced by the three $H/L = 0.30$ inserts. The $H/L = 0.50$ inserts make this reduction significant, especially for larger inlet areas, because slowing the coolant flow at more holes in the converging channel must increase $\dot{m}$ per hole in the diverging channel. This trend is continued by the $H/L = 1$ insert G, for holes within the diverging passage. At $s = 0.15$, insert D increases coolant pressure outside of the insert, and affects little change inside it. At $s = 0.30$, it yields an equal but small static pressure gain both inside and out. These changes are not useful; it is preferable to realise a significant gain within the inserts, at the expense of the static pressures outside them (at the inlets of low ingestion risk holes).

Effects of Successfully Designed Inserts

Both inserts E and F produce much increased static pressures inside, with only slight reductions outside at $s = 0.15$ and $s = 0.30$, compared to no insert pressures. The trade-off between them is that the even higher pressures within F cause lower static pressures outside at $s = 0.75$. The choice between them in a double-end feed arrangement would therefore depend upon the relative static pressure requirements at the ingestion risk film cooling holes and any (internal or film) cooling features fed from the channels outside of the inserts. As mentioned previously, a full-span insert such as G may be required for application of the concept in a single-end feed NGV leading edge. This device also succeeds in increasing its internal static pressures at $s = 0.15$ and $s = 0.30$. The static pressure results for $s = 0.75$ are increased inside the insert, but appear to decrease towards the stagnation region ($x/C_x = 0$). This indicates a leak where the insert is not fully sealed by the silicone adhesive, which proved difficult to apply at locations deep within the passage. If the two channels were properly isolated in this region, the pressure within the insert would further increase, while the pressure outside it would further decrease. Moreover, only a single $H/L = 1$ profile has been tested. Therefore, the results show that the insert concept is viable for both single- and double-end fed NGVs.

Cooling Hole Pressure Margins

The internal and external pressure data at each of the three measurement spanwise positions can be interpolated to estimate the static pressures at the centres of each hole inlet ($p_{5,i}$) and exit ($p_{5,e}$), respectively. Fig. 6 shows the normalised differences in static pressure (margins) implied by this process, across all hole rows. The external pressure distribution causes SS5 and SS6 margins to be much higher than others. To facilitate comparison of insert performance for all rows, the hole pressure margin results for each row are normalised once more by the highest margin achieved in that row by any insert. For insert A, the margins of covered holes decrease at $s = 0.15$, in accordance with the decreased internal pressures noted previously. Inserts B and C increase these margins at the expense of the SS5 and SS6 margins. The effect is more significant at higher $\%\dot{m}_{NGV}$, for which insert C almost equalises the margins of all rows. The inserts slightly reduce the downstream margins at $s = 0.30$ and $s = 0.75$, except for the separation region behind insert C, which suffers significantly reduced margins (note the change of vertical axis scale for $s = 0.30$). Insert D is too narrow to produce a significant effect. Inserts E and F significantly improve the hole margins within them for all $\%\dot{m}_{NGV}$, and at high $\%\dot{m}_{NGV}$ F essentially equalises the margins at all rows. Both tend to reduce the pressure margins at $s = 0.75$, though this is only significant for high $\%\dot{m}_{NGV}$. Insert G increases the at-risk row margins for all $\%\dot{m}_{NGV}$ and experimental $s$. At-risk margins appear unaffected at $s = 0.75$, but this is a relic of the leakage from this region of the diverging channel due to poor sealing.
Viscous Modelling for Prediction of Coolant Pressure Margin

The coolant passage internal pressures (Fig. 2) generated by ANSYS Fluent RANS CFD were significantly overestimated, independently of mesh density and turbulence model. This is attributed to the overly large vena contra i.e. underestimated discharge coefficient \( (C_d = \frac{(\text{mass flow})}{(\text{area}) \cdot \text{velocity}}) \) predicted within each cooling hole. The reduced effective flow area due to larger vena contra means that an unrealistically high driving pressure is needed in order to produce the mass flow rates equivalent to experiments. For long cooling holes (length-to-diameter ratios greater than two) such as those of this study, \( C_d \) underestimation is ubiquitous with RANS. The effect has been observed in comparisons to
experimental $C_d$ data, even when individual holes are simulated with a variety of airfoil film cooling inlet and outlet conditions and various turbulence models and mesh fidelities (Gupta et al., 2008). In the absence of easily obtainable, reliable CFD results, the following enhanced analytical method may complement or replace limited resolution experimental data. Total pressure losses in coolant channels of length $H$ are estimated by pipe flow Darcy friction factors $f$ in converging sections and diffuser loss coefficients $K_l$ in diverging sections. In this analysis, $Q(s)$ is derived from the definition of total pressure and $p_{0c} = \rho_c R T_0$, and $F(s)$ is the fraction of inlet dynamic pressure lost for a circular pipe of length $ds$, according to the Darcy-Weisbach equation. $R$ is the specific gas constant for air. $f(s)$ is estimated according to Serghides (1984), making it in fact a function of Reynolds number, hydraulic diameter $D_h$ (ultimately functions of $s$) and surface roughness (assumed constant).

$$Q(s) = \frac{p_{0c}(s)}{p_{0c}(s)} = 1 - \left[\frac{\rho_c(s)}{\rho_c(A(s))}\right]^2 \times \frac{1}{2 R T_0(s)} F(s) = \int_0^s \frac{f(s) \, ds}{\nu_H(s)}$$

Channel static pressure can be expressed with $Q(s)$ and the cumulative total pressure loss $p_{0,lost}(s)$:

$$p_{sc}(s) = Q(s) \times p_{0c}(s) \rightarrow p_{sc}(s) = Q(s) \times [p_{0c}(s = 0) - p_{0,lost}(s)]$$

A converging channel is treated as a series of pipes of infinitesimal length $dz$, yielding the following simplified, viscous form of eqn. 2, which applies also to prismatic sections. $z$ is a dummy variable.

$$p_{sc,conv}(s) = Q(s) \times [p_{0c}(s = 0) - \int_0^s F(z) \times \frac{1}{2} \rho_c[u_c(z)]^2 \, dz]$$

(5)

The static pressure in a diverging channel $p_{sc,dif}(s)$ is determined more simply, by assuming a linear $p_{0,lost}(s)$ amounting at the channel’s exit to $K_l$ times the dynamic pressure at its inlet. In practice, total pressure loss can be calculated at specific spanwise intervals which divide the channels into short, successive sections of diffuser or pipe. This avoids the need to solve exactly the integral in eqn. 5. Connected channels can be treated consecutively. For comparisons to the experiments of this study, inlet total pressures were estimated from static pressure measurements within inserts at $s = 0.15$, the channel velocities, and pipe flow losses. Area distributions $A(s)$ of the channels and channel mass flow reductions $\dot{m}_c(s)$ were approximated as linear. The results are quite insensitive to estimated surface roughness ($0.1 \text{ mm}$ was used). The $K_l$ values were estimated as 1.0 for insert F and 0.5 for the lower diffusion rate insert G. Little basis currently exists for more accurate prediction of performance of irregularly shaped diffusers with flow extraction. The pressure margin for a cooling hole row with spanwise constant exit static pressure $p_{se, row}$ is $PM_{row}(s) = p_{sc}(s) - p_{se, row}$. Despite these simplifications, accurate and useful hole pressure margin predictions were made, as shown in Fig. 7. SS1 and SS5 are typically the lowest margin hole rows inside and outside of the inserts, respectively. The near-inlet pressure margin in the diverging channel is revealed to be significantly reduced by the overly broad insert G. This suggests that for a full-span insert, special care must be taken near the coolant inlet to balance the coolant pressures inside and outside of the insert.

Figure 7: Viscous model hole pressure margin predictions for inserts F and G, $%\dot{m}_{NGV} = 4.7\%$
CONCLUSIONS
A large-scale, film-cooled nozzle guide vane in engine-representative combustor flow conditions has been used to experimentally measure the internal coolant and external surface pressures at various coolant mass flow rates. Stagnation region holes are at considerably greater risk of hot gas ingestion than rows far around the suction side. Seven coolant passage inserts were tested with the goal of reducing the ingestion risk by increasing the margin by which the static pressure at the entrances of at-risk holes exceeds that at their exits. This is achieved by slowing the flow to these holes by enclosing them within a converging channel with cross-sectional areas which are large relative to the number of holes supplied by the channel. The pressure margins for the unaltered prismatic leading edge coolant passage are compared to those in the presence of the inserts, in order to evaluate the efficacy of the concept and to identify design rules. The main conclusions drawn are:
1. In the diverging channel formed outside of an insert, the rate of diffusion must not be so large as to produce separations (which create localised low pressure zones where ingestion may occur)
2. Broader inserts increase/decrease the pressure margins for cooling holes inside/outside the inserts
3. Higher coolant flow rates exacerbate both of the effects in (2), resulting in more complete equalisation of the margins of holes within the inserts and outside
4. The insert concept can improve pressure margins, while potentially also controlling blowing ratios and/or internal cooling feature feed conditions, for single- and double-end fed airfoils
5. The data have validated an analytical method for predicting the pressure margin performance of combined converging, diverging and prismatic cooling passages

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REFERENCES
Cresci, I., Ireland, P. T., Bacic, M., Tibbott, I., Rawlinson, A., (2015b). Velocity and turbulence intensity profiles downstream of a long reach endwall double row of film cooling holes in a gas turbine combustor representative environment. ASME Turbo Expo 2015, Montreal, Canada
Cresci, I., Ireland, P. T., Bacic, M., Tibbott, I., Rawlinson, A., (2015c). Realistic velocity, turbulence and temperature profiles at the combustor-turbine interaction plane in a rig. International Gas Turbine Congress 2015, Tokyo, Japan


