Film Cooling Performance Predictions For Air And Supercritical CO₂

30 August 2021
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Cover Illustration: sCO2 coolant streamlines, surface temperature contours, coolant jet shape & penetration in to the freestream, and coolant iso-surface contours at 1300 K.

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Film Cooling Performance Prediction For Air And Supercritical CO₂

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Pete Strakey, Technical Portfolio Lead
## Table of Contents

1 EXECUTIVE SUMMARY ........................................................................................................ 1

2 INTRODUCTION .................................................................................................................. 2
   2.1 sCO2 TURBINE COOLING ........................................................................................... 2
   2.2 FILM COOLING ............................................................................................................ 2
   2.3 CROSSFLOW COOLANT INJECTION ............................................................................. 4
   2.4 FILM COOLING CFD .................................................................................................... 4

3 METHODS ............................................................................................................................. 6
   3.1 NUMERICAL DOMAIN ............................................................................................... 6
   3.2 MODEL ....................................................................................................................... 6
   3.3 TEST CASES ............................................................................................................... 7

4 OBSERVATIONS ................................................................................................................ 10
   4.1 GRID INDEPENDENT STUDY .................................................................................... 11
   4.2 COMPARISON WITH LITERATURE ........................................................................... 11
   4.3 EFFECT OF CROSSFLOW INJECTION & TURBULENCE MODEL ......................... 12
   4.4 EFFECT OF REYNOLDS NUMBER ........................................................................... 13
   4.5 EFFECT OF BLOWING RATIO & DENSITY RATIO ............................................... 14
   4.6 FILM COOLING EFFECTIVENESS: AIR VS. sCO2 ................................................ 16

5 CONCLUSIONS .................................................................................................................. 18

6 NOMENCLATURE ............................................................................................................... 19

7 REFERENCES ..................................................................................................................... 20
List of Figures

Figure 1: A generic numerical model used to study crossflow injection ........................................ 6
Figure 2: Sample tetrahedral mesh near the film cooling hole. Test case #5. ................................. 7
Figure 3: Fluid properties as a function of temperature: a) Air, b) sCO2 ................................. 9
Figure 4: Surface temperature Comparison: a) CFD (unstructured) and b) interpolated
  (structured, grid size = d/10). Test case #1 ........................................................................... 10
Figure 5: Laterally averaged adiabatic or film effectiveness comparison. Test case #5 .............. 11
Figure 6: Laterally averaged effectiveness: Grid independent study ........................................... 11
Figure 7: Comparison of laterally averaged effectiveness: literature vs. current study .............. 12
Figure 8: Effect of crossflow and turbulence model: a) Laterally averaged film effectiveness, b)
  Film effectiveness contours .................................................................................................. 13
Figure 9: Effect of freestream Reynolds number: a) Laterally averaged film effectiveness, b)
  Film effectiveness contours .................................................................................................. 14
Figure 10: Effect of blowing ratio and density ratio on laterally averaged film effectiveness: a)
  Air, b) sCO2 .......................................................................................................................... 15
Figure 11: Laterally averaged effectiveness at a) BR=0.5 & c) BR=1.0; Difference in laterally
  averaged values between air and sCO2 at b) BR=0.5 & d) BR=1.0 ........................................ 16
Figure 12: Adiabatic or film effectiveness distribution downstream of cylindrical hole exit. a) BR
  = 0.5 & DR = 1.5, b) BR = 0.5 & DR = 2.0, c) BR = 1.0 & DR = 1.5, and d) BR = 1.0 & DR
  = 2.0 ...................................................................................................................................... 17
List of Tables

Table 1: Numerical Test Plan ................................................................. 7
Table 2: Test Cases ............................................................................... 8
Table 3: Statistical comparison of surface temperature (K) .................. 10
## Acronyms, Abbreviations, and Symbols

<table>
<thead>
<tr>
<th>Term</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>BR</td>
<td>Blowing ratio</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational fluid dynamics</td>
</tr>
<tr>
<td>d</td>
<td>Film cooling hole diameter</td>
</tr>
<tr>
<td>DES</td>
<td>Detached eddy simulation</td>
</tr>
<tr>
<td>DNS</td>
<td>Direct numerical simulations</td>
</tr>
<tr>
<td>DR</td>
<td>Density ratio</td>
</tr>
<tr>
<td>l</td>
<td>Momentum flux ratio</td>
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<tr>
<td>LES</td>
<td>Large eddy simulations</td>
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<tr>
<td>RANS</td>
<td>Reynolds averaged Navier-Stokes</td>
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<td>RKE</td>
<td>Realizable k-epsilon ($k\epsilon$)</td>
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<td>SAS</td>
<td>Scale adaptive simulation</td>
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<td>sCO2</td>
<td>Supercritical carbon dioxide</td>
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<tr>
<td>SKW</td>
<td>Standard k-omega ($k\omega$)</td>
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<tr>
<td>SST-KW</td>
<td>Shear stress transport k-omega ($k\omega$)</td>
</tr>
<tr>
<td>$\nu$</td>
<td>Velocity</td>
</tr>
<tr>
<td>VR</td>
<td>Velocity ratio</td>
</tr>
<tr>
<td>$\delta$</td>
<td>Boundary layer thickness</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Film effectiveness</td>
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### Subscripts

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<th>Subscript</th>
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<tr>
<td>c</td>
<td>Coolant</td>
</tr>
<tr>
<td>ch</td>
<td>Channel</td>
</tr>
<tr>
<td>g</td>
<td>Hot gas</td>
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Acknowledgments

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The authors would like to acknowledge the significant contributions and guidance provided by James Black who recently retired from NETL.
1 EXECUTIVE SUMMARY

The current study explores the possibility of cooling the vanes and blades of a direct-fired sCO₂ turbine using film cooling geometries. The operating conditions of a direct-fired sCO₂ cycle and thermophysical properties of the fluid at those conditions can alter the flow field characteristics of the coolant jet and its mixing with the mainstream. Very little information is present in the literature regarding the performance of film cooling geometries employing a supercritical fluid. The objective of this study is to estimate the resulting film cooling effectiveness while also capturing the effects of the crossflow-to-mainstream velocity ratio on the coolant jet. A computational fluid dynamic model is used to study the coolant jet exiting a cylindrical hole located on a flat plate, with the coolant fed by an internal channel. Steady-state RANS equations were solved along with the SST κω model to provide the turbulence closure. The operating conditions for the direct-fired sCO₂ turbine are obtained using an in-house Cooled Turbine Model. Numerical predictions revealed that the crossflow effects and jet lift-off were more pronounced in the case of sCO₂ when compared to air. Spatial distribution of flow field and cooling effectiveness will be presented at different operating conditions. Recommendations for future studies will be given wherever applicable.
2 INTRODUCTION

2.1 sCO2 TURBINE COOLING

sCO2 power cycles have gained a lot of interest in the recent years. Some of the benefits include higher efficiencies, reduced fuel usage, carbon capture, and compact systems that will potentially reduce cost. While indirect cycles rely on heat transfer surfaces to heat the CO2 working fluid, direct fired cycles combust fossil fuels that results in a mixture of CO2 and steam to power the turbine. Weiland et al. [1] analyzed the performance of a baseline direct fired coal plant that produces ~ 600 MW of net power. The turbine inlet temperature and pressure were expected to be close to 1150 °C or 1423.15 K and 300 bar or 30 MPa, neglecting the small pressure drop across the combustor. Typically, gas turbines operating above 950 °C (1223.15 K) are actively cooled, while gas temperatures above 1200 °C (1473.15 K) largely make use of film cooling and thermal barrier coatings to keep the material temperature below the allowable limits [2]. Higher turbine inlet temperatures for direct fired power cycles are within the realm of possibility, but there is a need to study turbine cooling for sCO2 power cycles.

Film cooling and impingement cooling are critical to first stage turbine vanes and blades of gas turbines especially where the turbine inlet temperatures can exceed 1475 K. At lower temperatures, the cooling effectiveness, reduced heat load, and aerodynamic penalties and other factors can be considered before employing external cooling techniques. Though the current sCO2 turbines are expected to operate at lower turbine hot gas temperatures, the freestream Reynolds numbers are expected to be orders of magnitude higher than that of the gas turbines. At 30 MPa and 1473.15 K, the Reynolds number for a given chord length can be anywhere between 10-15 times that of conventional gas turbines. The thermal load on the airfoil depends on several factors such as pressure and temperature gradients, turbulence level, surface curvature, surface roughness, laminar-to-turbulent transition, and flow unsteadiness. As suggested by Cunha [3], if properly accounted using a coefficient, the external surface Nusselt number can be expressed as a function of the freestream Reynolds number by employing a modified version of the flat plate correlation as shown below.

\[ Nu_x = C_{airfoil} R e^4 P r^{1/3} \]  

Under such high thermal loads, extreme operating pressures, supercritical nature of the working fluid, external cooling techniques like film cooling might become necessary. Turbine cooling has been studied extensively for gas turbines in the past five decades [4]. However, little attention has been given to cooling engine components using supercritical CO2. To the authors’ knowledge, this is one of the first studies to explore film cooling using supercritical carbon dioxide.

In the recent years, internal cooling using sCO2 has received some interest. Searle et al. [5] conducted experimental and numerical investigations of three variants of internal cooling configurations – dimples only, ribs only and ribs with dimples have been explored in the HEET facility [6] with sCO2 as the coolant. Numerical analysis of these three internal cooling designs was carried out over a range of Reynolds number from 80,000 to 250,000. Based on the experimental results and numerical predictions, the authors found that the Nusselt number augmentation increases as higher Reynolds numbers were approached, whereas prior work on internal cooling of air-breathing gas turbines predicted a decay in the heat transfer enhancement as Reynolds number increases.

Khadse et al. [7] investigated the heat transfer effects on the first stage vane of a sCO2 turbine. A single vane with 6 internal circular channels along with the freestream was modeled numerically as a conjugate problem. Film cooling holes were not modeled by Khadse et al. The freestream temperature and pressure were modeled at 1350 K and 28 MPa respectively but the authors’ do acknowledge a higher turbine inlet temperature for future oxy-combustion sCO2 cycles. For the coolant mass flow rates used in the study, the wall temperature was not sufficiently lowered, and the authors have discussed film cooling as a possible way to reduce the airfoil surface temperature.

2.2 FILM COOLING

Gas turbine cooling has been studied since the 1970s. It plays a vital role as it allows engines to operate at higher turbine entry temperatures, and subsequently at higher efficiency, by keeping the vane/blade within the maximum allowable working temperature. Film cooling is critical for gas turbines as it protects the vane and blade surface from the hot gas at the location of injection as well as the downstream region by forming a thin layer of coolant film. Performance of a film cooling geometry is usually expressed in terms of a normalized temperature ratio or film effectiveness. As discussed in [8], the adiabatic wall temperature is often used as the reference temperature for
estimating the heat transfer on a film cooled component and subsequently the cooling effectiveness as well (Eq. 2, for incompressible flows).

\[
\eta = \frac{T_{g}-T_{ad}}{T_{g}-T_{c}}
\]  

Earlier studies [9,10] were focused on measuring the film effectiveness as adiabatic wall temperature was found to vary more than the heat transfer coefficient which was probably reasonable at lower blowing ratios. Liess [11] investigated the performance of a 35° inclined film cooling holes by measuring both the adiabatic effectiveness and the heat transfer coefficient on a flat plate with copper strips at various conditions. The upstream boundary layer displacement thickness (\(\delta_1\)), however, was found to have a strong influence on effectiveness but no measurable effect on heat transfer coefficient. For e.g., roughly doubling the \(\delta_1/d\) from 0.25 to 0.5, reduced the adiabatic effectiveness at \(x/d = 10\) from 0.2 to 0.1. Also, the increment in the heat transfer coefficient due to coolant injection was found to be as high as 50% in the near hole region (\(x/d < 5\)) at highest blowing ratio studied while staying flat near 20-25% past \(x/d > 15\). Hay et al. [12] explored a novel technique based on a heat-mass transfer analogy to measure local heat transfer coefficients over film cooled surface with a similar hole inclination angle. At higher blowing ratios, a 35% increase in the laterally averaged heat transfer coefficient ratio was observed.

Most of the film cooling studies, except for a few, utilized CO2 to simulate film cooling under engine conditions where the coolant to freestream density ratio is expected to be closer to 1.8. The applicability of using a secondary fluid with higher density but at a temperature closer to mainstream was studied by Teekaram et al. [13]. Eckert et al. [14] reported his findings on the similarity analysis of film cooling experiments typically conducted at near ambient condition and its scaling at high temperatures in gas turbines. Sinha et al.[15] conducted experiments to measure film cooling effectiveness on a flat plate made of polystyrene foam (Styroform, \(k = 0.027 \text{ W/mK}\)) using a single row of holes. Their test section had cylindrical film cooling holes inclined at 35 deg to the surface with an L/D of 1.75. Similar to [10], the study focused on understanding the relative importance of density ratio, blowing ratio, velocity ratio, and momentum flux ratio on film effectiveness. The authors found that the centerline effectiveness scales with mass flux ratio and momentum flux ratio at lower and higher blowing rates respectively. A single parameter was not sufficient to scale either the centerline or the laterally averaged effectiveness when the density ratio was varied. Schmidt et al. [16] investigated a 60° compound angle on film cooling effectiveness of cylindrical and forward expanded hole using the same test setup. Ekkad et al. [17] conducted transient film cooling experiments using a liquid crystal technique to study the effect of compound angle on film effectiveness. Unlike before, a single transient experiment was conducted on a test section made of plexiglass (\(k \sim 0.18 \text{ W/mK}\)), to estimate both film effectiveness as well as heat transfer coefficient due to film cooling. Baldauf et al. [18] studied the scalability of flow parameters for film cooling for engine conditions using CFD simulations. Following a similar argument, the authors simplified the original expression for laterally averaged effectiveness obtained from the Buckingham PI theorem. The influence of pressure could be neglected and the temperature ratio (\(T_c/T_g\) or \(DR\)) was found to be sufficient to account for ratios of transport properties and Prandtl number within an error margin of 2-3%. With the help of CFD simulations, the effects of Reynolds number (\(Re_d > 5500\)) and Eckert number (\(Ec_g\)) on adiabatic effectiveness was shown to be small, resulting in a much-simplified expression shown below.

\[
\bar{\eta} = f \left( BR, DR \text{ or } I, Tu, \frac{\delta_1}{d}, \frac{x}{d}, \frac{s}{d}, \frac{L}{d}, \alpha \right)
\]  

Burd & Simon [19] found the coolant supply geometry (co-flow, counter-flow, short hole with unrestricted plenum) to have a noticeable effect on both the centerline and laterally averaged effectiveness. Steady state heat transfer experiments were conducted on a test plate made out of silicon phenolic laminate plate (\(k = 0.25 \text{ W/mK}\)). Interestingly, the adiabatic effectiveness was measured from extrapolated wall temperature obtained by thermocouple traverse measurement under film cooling. As a result, measurements were made at discrete locations (6 along streamwise and 23 along lateral direction about a single hole). Effect of hole length and free stream turbulence has been studied in detail by [20,21]. Gritsch et al. [22] measured adiabatic effectiveness of shaped holes on a test plate made of a relatively higher temperature plastic material (TECAPEK, \(k = 0.2 \text{ W/mK}\)) using an IR camera system. Drost & Bolcs [23] investigated film cooling performance of cylindrical holes on a nozzle guide vane under different freestream conditions. Heat transfer experiments were conducted over a test surface made of plexiglass using a transient liquid crystal technique.
Baldauf et al. [24] conducted flat plate experiments to obtain a spatial resolution of adiabatic (film) effectiveness due to cylindrical film cooling holes. Experiments were conducted on a semi-crystalline thermoplastic material (Tecapek) at moderately high hot gas temperatures \((T_g = 550 K)\) and at mass flow rate of 1.3 kg/s. Greiner et al. [25] explored the scaling parameters to compare the film cooling performance from ambient and near engine conditions using CFD. They found that thermophysical properties or property ratios do affect scaling of adiabatic effectiveness. In addition to matching density & blowing ratio, Prandtl number & Reynolds numbers needed to be matched to scale film cooling effectiveness accurately.

Several film cooling hole geometries have been proposed ([26–34]) that have been proven to provide an effectiveness higher than the conventional cylindrical holes at laboratory conditions. Typically, these studies have employed test articles that were made of low thermal conductivity materials such as low and moderate temperature thermoplastics to estimate the film effectiveness.

### 2.3 CROSSFLOW COOLANT INJECTION

In most experimental and numerical film cooling studies, the coolant is typically fed through a quiescent plenum unlike gas turbines were the coolant is fed from an internal channel perpendicular to the hot gas flow path. Only a few known studies have investigated the effects of crossflow channel on film cooling effectiveness. Gritsch et al. [35] conducted a comprehensive study investigating the effects of internal channel Mach number on adiabatic effectiveness distribution over a flat surface. Three types of film cooling hole shapes (cylindrical, fan-shaped, and laid-back fan-shaped hole) were studied for a range of blowing ratios (0.5 to 2) and coolant crossflow Mach numbers (0, 0.3 & 0.6). The density ratio was maintained close to 1.85 and assumed to be representative of typical engine conditions. Experiments were conducted on a thermoplastic material with a thermal conductivity of 0.2 W/mK. In all cases studied, film cooling effectiveness distribution was found to be affected in the near hole region \((x/d < 8)\) when the coolant channel Mach number was increased from 0 to 0.3 and 0.6. In case of a cylindrical hole, the coolant lateral coverage was found to increase with the peak effectiveness shifting towards the coolant channel upstream side \((z/d < 0)\) thereby resulting in a skewed distribution. The coolant crossflow increased the laterally averaged effectiveness at all blowing ratios expect at 0.5 indicating that the jet detachment typically expected at higher momentum flux ratios was absent. In case of the fan-shaped hole, at higher crossflow Mach numbers, the effectiveness distribution was skewed with the peak values occurring near \(\frac{z}{d} \geq 1\) when compared to a plenum fed hole where peak values are usually present towards the hole center \((-1 < \frac{z}{d} < 1)\).

A three part study was carried out by Saumweber et al. [36–38] where the effects of free stream turbulence, internal coolant passage was studied for a range of geometric variations. Over a range of blowing ratios studied for a 6° fan-shaped hole, higher coolant crossflow Mach numbers (0.3 & 0.6) were found to have a tendency to skew the coolant distribution over the surface and the local effectiveness in the \(-z\) direction \((z/d < 0)\) except for at the highest \(M_a\) and lowest blowing ratio where the effectiveness distribution was skewed in the \(+z\) direction \((z/d > 0)\). Consistent with Gritsch et al. the extent of skewness in the local adiabatic cooling effectiveness and the subsequent laterally averaged value, though not monotonic, was a function of the crossflow Mach number, blowing ratio and the diffusor angle.

More recent study from McClintic et al. [39] looked at a range of crossflow velocity ratio (coolant channel to hotgas) and coolant jet to hot-gas velocity ratio for the 7-7-7 shaped hole to understand the impact of crossflow velocity ratio at various coolant injection rates and understand scalability. A significant finding was the relationship between coolant jet characteristics and the coolant channel to coolant jet velocity ratio (VRi). The takeaway from these findings is that at lower injection rates (VR < 1), the adiabatic effectiveness was insensitive to VRc. VRi, by virtue of its relationship with VRc & VR, is expected to have little impact on adiabatic effectiveness as well. At higher VR (> 1.1), key film cooling such as centerline effectiveness, Cd, etc. scaled better with VRi.

### 2.4 FILM COOLING CFD

Several research groups have compared the performances of various turbulence models in predicting the adiabatic cooling effectiveness and heat transfer enhancement due to film cooling. Conjugate effects of film cooling has been studied by [40–44]. A few studies pertinent to the current work are explained in detail. Other relevant studies are referenced in the appropriate sections.

Harrison et al. [45]’s study included RKE, SKW and RSM models where the RKE and RSM models were coupled with enhanced wall treatment approach to model the near wall region. Of the three, the Standard \(k\omega\) (SKW) model...
was found to be best predict the laterally averaged effectiveness with the RKE being the worst and vice versa for the centerline effectiveness. Consistent with other studies, the RKE and SKW did not predict the lateral spread of the coolant. Though the RSM model accounts for anisotropy, it did not capture the lateral spreading of coolant as accurately as expected. The performance was similar in predicting the heat transfer coefficients. The SKW model was recommended for predicting laterally averaged effectiveness which is one of the more commonly studied and reported outcomes in the film cooling literature. Arguably, cooled gas turbine models often rely on laterally averaged or area averaged effectiveness in estimating the performances of a particular cooling configuration.

Na et al. [46] looked at the effect of three different RANS based eddy diffusivity models (realizable $k\epsilon$, shear-stress transport (SST-KW), and Spalart-Allmaras (SA)) to predict film cooling performance over a flat plate and a semi-cylindrical leading edge. A second order upwind differencing scheme was used in the commercial solver Fluent. The eddy diffusivity models were primarily investigated since high fidelity models such as DES/LES/DNS are computationally too expensive for design purposes even though they offer better accuracy. Off the three turbulence models, the SST-KW and SA models were found to predict laterally averaged adiabatic effectiveness on a flat plate reasonably well. The realizable $k\epsilon$ (RKE) severely underpredicted the laterally averaged value though was relatively closer to the experimentally measured centerline effectiveness.

Stratton et al. [47] studied the effects of crossflow in a setup similar to the one used in the current work. Film cooling effectiveness of 8 compound angle holes ($\beta = 45^\circ$) were numerically predicted using two commonly used turbulence models (SST-KW and RKE) at three different blowing ratios (BR=0.5, 1.0, and 1.5). The mainstream flow velocity and the boundary layer thickness upstream of the holes were 13.8 m/s and $\delta = 2.8d$ where $d$ is film cooling hole diameter. Simulations were conducted using a hot air at 303 K and using nitrogen coolant at 202 K, resulting in a density ratio of 1.5. The authors found out that at lower blowing ratios, the RKE model predicted the laterally averaged effectiveness better than the SST-KW when compared to the experimental results. At higher blowing ratios, an opposite trend was observed.

Kampe et al. [48] performed experiments and CFD simulations to analyze the flow field and estimate the surface temperatures downstream of cylindrical and shaped holes. The kw-SST turbulence model was employed for the numerical predictions. In terms of the cylindrical hole flow field, such as velocity distribution, magnitudes, jet expansion, and secondary flow, the authors found very good match between the experiments and CFD simulations. From cooling effectiveness perspective, the predictions were well even though the shape of the laterally averaged effectiveness was not the same.

Quite a few studies ([30,40,43,49–52]) have relied on the $k\omega$ based turbulence models to predict the behavior of a jet in a crossflow problem. The $k\epsilon$ based equations have also been used to varying degrees of success in the film cooling literature ([53–57]). In the current study, both turbulence models are briefly studied to understand their influence in predicting film cooling effectiveness. Given the lack of information on film cooling for sCO$_2$ turbines, the numerical predictions presented in the current work are expected to provide a preliminary understanding what can be expected.
3 METHODS

3.1 NUMERICAL DOMAIN

It is quite common in film cooling studies to investigate the cooling performance over a flat surface as it provides an opportunity to isolate the effects of surface curvature and secondary flows. With the objective of understanding the difference in performance due to a nature of the working fluid and nature of coolant injection, the current study employs three different variations of a flat plate test geometry. The first design uses a familiar film cooling flat plate setup where film cooling holes are fed through a plenum. A second design has the same hole diameter \( d = 0.2 \text{ in (5.08 mm)} \) but the plenum was replaced with a crossflow channel beneath the film cooling holes (see Figure 1). The final design has a crossflow channel beneath a smaller film cooling hole, something that might be representative of engine conditions \( d = 0.025 \text{ in (0.635 mm)} \). Harrison et al. [45] found that a channel height of 6d was sufficient and provided results similar to a channel of height 20d. To be on the more conservative side, a channel height of 10d was used in the current study in addition to modelling the top wall using a symmetric boundary condition. The current study models one-hole pitch in the lateral direction by using symmetric boundary conditions, thereby effectively reducing the computation cost. The remaining walls were treated as adiabatic surfaces.

![Figure 1: A generic numerical model used to study film cooling under crossflow injection](image)

3.2 MODEL

The numerical domain for the first two designs is meshed with roughly 1.14 million patch conforming tetrahedral elements (see Figure 2 for reference). The flat surface upstream and downstream of the film cooling hole exit, film cooling hole and the surface flush with the hole inlet are meshed with an element size 1/6 of the hole diameter while the rest of the domain is meshed coarsely. Neighboring cells in the vicinity of these surfaces, a region within the range of 2.5d, are also influenced by the element size. To reduce the mesh skewness and improve the orthogonal quality of the mesh inside the hole, esp. near its inlet and exit, small fillets \( r_f = d/10 \) were introduced in the all designs. Owing to the computational cost and number of design points, Reynolds-Averaged Navier-Stokes equations were solved using a steady state solver instead of resorting to DNS or Large eddy simulations (LES). The shear stress transport variant of the \( k\omega \) and the realizable \( k\varepsilon \) (RKE) model provided the turbulence closure. An enhanced wall treatment method was employed for near wall modelling. Both the SST-KW and the RKE are integrated till the wall. The prism layers of varying first layer thicknesses were added depending on the mainstream Reynolds number. The wall \( y^+ \) kept close to 1 for all the cases. A green gauss node based gradient approach was selected as it was node-based gradient is known to be more accurate than the cell-based gradient particularly on an irregular unstructured meshes. A second order upwind spatial discretization scheme was used for all the variables. A coupled solver with a pseudo transient formulation was selected instead of a pressure based segregated solver (and pressure-velocity coupling schemes) as it was found to accelerate convergence. Convergence criteria was set to 1E-6 for both momentum and energy equations and 1E-5 for the turbulent kinetic energy and specific dissipation rate equations. The area averaged temperature on the surface downstream of film cooling hole exit was also monitored in addition to the
residuals. For most cases, the area averaged temperature was observed to reach a steady state value within 150-200 iterations and the residuals were found to converge within 400 iterations. Additionally, 600 iterations were carried out to be on the conservative side.

Figure 2: Sample tetrahedral mesh near the film cooling hole. Test case #5.

3.3 TEST CASES

The test plan and cases are shown in Table 1 and Table 2 respectively. The coolant temperature was estimated for a given blowing ratio (BR), density ratio (DR), and hot gas conditions. The range of conditions studied here can be described as typical for air breathing gas turbines. However, as observed by Uysal et al. [58], sCO₂ turbines can be operated at higher DRs than what is conventional. Future studies can explore this behavior to fully reap the benefits of a higher density coolant.

A mesh size of d/6 implies that roughly 6 elements are present within a space equal to one film cooling hole diameter. The test conditions were obtained using the fluid properties as a function of temperature and pressure for both air and sCO₂ (see Figure 3) were estimated using the CoolProp [59] library and a Python routine. Upon completion of the test cases shown in Table 2, a few additional simulations were carried out to inspect the accuracy of the predicted adiabatic effectiveness. A hybrid RANS-LES turbulence model, named Scale Adaptive Simulation, was the first one to be explored at this stage. This model is described as an improved version of the Unsteady RANS turbulence model as it is capable of resolving the turbulence spectrum (smaller eddies) in flows that have inherent unsteadiness. A really fine time step size of $10^{-5}$ s was chosen. Simulations were run for 700 timesteps which is roughly more than twice the time taken for the flow to travel from inlet to the outlet of the numerical domain. Numerical data was sampled to estimate a time averaged solution for 700 additional time steps. For a freestream velocity of 60 m/s, the Strouhal number based on film cooling hole diameter was roughly 8.5. After that, the grid independent study was extended with a finest mesh (size = d/12). The adiabatic effectiveness is presented in the next section.

<table>
<thead>
<tr>
<th>Case</th>
<th>Case No.</th>
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<th>Crossflow</th>
<th>Turbulence model</th>
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<td>Validation with literature, grid independent study</td>
<td>3-5</td>
<td>0.2</td>
<td>10d</td>
<td>No</td>
<td>SST k-w</td>
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<td>Crossflow coolant injection</td>
<td>1-2</td>
<td>0.2</td>
<td>30d</td>
<td>Yes</td>
<td>SST k-w &amp; RKE EWT</td>
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<td>0.025</td>
<td>10d</td>
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### Table 2: Test Cases

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<th>DR</th>
<th>Tg (K)</th>
<th>Tc (K)</th>
<th>Vg (m/s)</th>
<th>VRc</th>
<th>T1g (%)</th>
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<td>d/10</td>
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<td>982.10</td>
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<td>736.58</td>
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<td>2</td>
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<td>736.58</td>
<td>213.84</td>
<td>0.2</td>
<td>10.0%</td>
<td>5.0%</td>
<td>d/8</td>
</tr>
</tbody>
</table>
Figure 3: Fluid properties as a function of temperature: a) Air, b) sCO\(_2\)
4 OBSERVATIONS

To enable an efficient way of comparing results from several test cases, the unstructured data set on a particular surface from the simulation had to be converted to a structured array format. The wall temperature and nodal coordinates on the surface downstream of the film cooling hole exit was exported as a text file. As mentioned above, the numerical domain is meshed with roughly 1.14 million tetrahedral elements. Surface data exported from the simulation typically contain the nodal values on the base triangular face of these tetrahedrons. A temperature value is associated with every \( \{x,y\} \) corresponding to the location of the mesh node. Using the triangulation API within the Matplotlib library, the triangular mesh data was interpolated to a structure grid of size \( d/10 \) where \( d \) is the film cooling hole diameter (for e.g., \( d = 0.2 \) in for the validation case).

Figure 4 shows a scatter plot comparison between the original (unstructured data from the simulation, Figure 4a) and the interpolated (structured dataset, Figure 4b) surface temperature. For reference, the trailing edge of the film cooling hole exit is located at \( x/d = z/d = 0 \) with the hole lying between \(-0.5 \leq z/d < 0.5\). The interpolated data was found to carry some high frequency noise like components. A mild gaussian filter (low pass filter) was used to smooth out the interpolated data. Satisfactory reproduction of original prediction is key to the comparative study. Further comparison on the statistical difference between the two dataset is shown in Table 3. It can be observed that the difference in the minimum, maximum and mean values are 0.16 %, -0.01%, & -0.46% respectively. Finally, to corroborate the accuracy of the interpolation, the laterally averaged data at several downstream locations estimated directly from CFD Post is compared with the interpolated data in Figure 5 for case #5 from Table 3. A different test condition was selected for comparison with CFD post as it provides an opportunity to evaluate the accuracy of interpolation for more than one case. The adiabatic effectiveness is estimated using Eq. 2.

![Figure 4: Surface temperature Comparison: a) CFD (unstructured) and b) interpolated (structured, grid size = d/10). Test case #1](image)

Table 3: Statistical comparison of surface temperature (K)

<table>
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<th>Function</th>
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<tr>
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4.1 GRID INDEPENDENT STUDY

Three different elements sizes were chosen initially to estimate the impact of mesh on film cooling performance prediction. The marginal difference in laterally averaged effectiveness (see Figure 6) between the SST – kω and SAS model, indicates that the initial grid (d/6) chosen was sufficiently fine. To be on the conservative side, a grid size = d/8 or d/10 was chosen for all the cases.

4.2 COMPARISON WITH LITERATURE

After obtaining a grid independent solution, numerical predictions were compared against the experimental data published in the gas turbine film cooling literature. Figure 7 shows the laterally averaged effectiveness (\( \bar{\eta} \)) predicted using the SST – kω RANS model to lie close to the experimentally reported values. A noticeable variation in \( \bar{\eta} \) reported by various studies can be observed as well. The experimental uncertainty, not shown here, will further widen...
the spread of the data. This highlights the importance of studying factors other than blowing ratio and density ratio that could influence the adiabatic effectiveness.

The $SST-k\omega$ model predictions were also found to lie closer to the effectiveness predicted using the Standard $k\omega$ model by Harrison et al [45]. The momentum flux ratio (see Equation below) is roughly close to 0.67 for the cases presented in Figure 7.

$$I = \frac{\rho_c V_c^2}{\rho_g V_g^2}$$

(4)

4.3 EFFECT OF CROSSFLOW INJECTION & TURBULENCE MODEL

As mentioned earlier, it is critical to understand the impact of coolant injection mechanism on adiabatic effectiveness. Equally important is the ability of the turbulence model in correctly predicting the trend, i.e., whether crossflow injection will increase or decrease the laterally averaged effectiveness ($\bar{\eta}$). Since the focus of the current study is to investigate film cooling for sCO$_2$ turbines, only one crossflow velocity ratio $VR_c = \frac{V_c h}{V_g} = 0.2$ was chosen. Future efforts shall focus on higher values for $VR_c$ (0.4 – 0.8). The blowing ratio and density ratio for these simulations can be identified by referring Table 1 and Table 2. Figure 8a and Figure 8b show the effect of crossflow coolant injection and choice of turbulence model on laterally averaged effectiveness and local adiabatic effectiveness respectively.

Gritsch et al. [35] found that when the coolant was supplied through a crossflow channel, the laterally averaged effectiveness increased as seen in Figure 8a for the case of $SST-k\omega$ model. Under crossflow conditions, the peak adiabatic effectiveness was found to shift away from the centerline ($z/d=0$), towards the coolant flow direction beneath the film cooling hole (+ve z direction). Also, the SST kw clearly predicts the jet separation that has been typically reported at higher blowing ratios. The RKE model, on the other hand, was found to underpredict the jet detachment. These findings were observed by Stratton et al. [60] as well. The RKE also predicted a lower $\bar{\eta}$ under crossflow conditions. As a result, the SSTKW model was used for the remaining test cases. Having said that and based on the literature review, it appears that the RKE model might have predicted adiabatic effectiveness accurately at lower blowing ratios when compared to the SSTKW model. The SSTKW model has been observed to overpredict jet separation in certain cases. The impact of turbulence model on film cooling behavior fed through a crossflow channel needs to be studied comprehensively in conjunction with other relevant factors shown in Eq. 3.
4.4 EFFECT OF REYNOLDS NUMBER

Reynolds number used in experimental film cooling studies are typically in the range of 3,000-20,000 as it is expected to represent the engine conditions. The characteristic length scale used in the definition of Reynolds number for this application is the film cooling hole diameter.

Schroeder et al. [34] studied performance of a baseline shaped hole for a Reynolds number range of 2,800 – 20,600. McClintic et al. [61] studied the effect of crossflow on axial shaped holes at a mainstream approach Reynolds number of 6,000. Anderson et al. investigated the effects of freestream Mach number, Reynolds number (5,500 – 13,000), and boundary layer thickness on film cooling effectiveness of shaped holes. The range of Reynolds number in Baldauf et al.’s [62] study was in the range 6,800 – 14,000. The effect of Reynolds number on film cooling effectiveness for air is shown in Figure 9b. Figure 9a shows that, in the near hole region (\(x/d < 10\)), the laterally averaged effectiveness (\(\bar{\eta}\)) is largely independent of the Reynolds number \(Re_d \geq 5,872\). Past 15 hole diameters, the \(\bar{\eta}\) at \(Re_d = 5,872\) is higher than at \(\bar{\eta}\) higher Reynolds number by roughly 10%.
4.5 EFFECT OF BLOWING RATIO & DENSITY RATIO

Of all the factors [63] influencing film cooling effectiveness, blowing ratio and density ratio are agreed to be the most influential scaling parameters. Scaling these two quantities also scales the other two important ratios, namely the velocity and momentum flux ratio. The velocity ratio has been found useful in understanding the coolant-mainstream interaction, especially the shear layers that influence the shape of the coolant jet once it exits the hole. The momentum flux ratio, on the other hand, is useful in identifying conditions that could potentially result in the coolant film getting separated from the surface [64].

The effect of blowing ratio and density ratio on $\bar{\eta}$ using coolant air fed through a plenum has been well studied. The peak effectiveness was found to occur at a blowing ratio $\sim 0.6 - 0.85$. Increasing the density ratio from 1.0 to 2.0, with values closer to 1.8 being closer to realistic engine conditions, increases $\bar{\eta}$ as denser coolant has the tendency to stay close to the surface (lowers momentum flux ratio).
Figure 10 shows the $\bar{\eta}$ for air and sCO$_2$ turbines under a mild crossflow ($VR_c = 0.2$). At higher blowing ratio ($BR = 1.0$), increasing the density ratio from $DR = 1.5$ to $2.0$, increases the laterally averaged effectiveness consistent with the expectations. As mentioned earlier, a mild separation was observed near the hole exit followed by a peak in effectiveness shortly after, between $5 < x/d < 10$, due to the nature of crossflow coolant injection. However, at lower blowing ratio ($BR = 0.5$), for the most part, $\bar{\eta}$ decreased with increasing density ratio. Since the momentum flux ratio ($I = 0.17$) was sufficiently low enough to begin with, reducing it further (to $I = 0.125$) did not change the coolant film characteristics as it did at higher blowing ratio ($BR = 1.0, DR = 1.5$ to $2.0, I = 0.67$ to $0.5$). The reduced momentum flux ratio simply appears to have caused a lower $\bar{\eta}$.

Film cooling with supercritical CO$_2$ showed similar features and trends when compared to gas turbines. The laterally averaged film effectiveness at $BR = 0.5$ is greater than at $BR = 1.0$ and the effect of density ratio seems to be quite similar when compared to Figure 10a. At higher blowing ratio ($BR = 1.0$), the effect of coolant jet separation in the near hole region, $0 < x/d < 10$, is clearly visible even at higher density ratio ($DR = 2.0$). This behavior is quite pronounced for sCO$_2$ when compared to using air at engine conditions as peak $\bar{\eta}$ can be found only in the far
downstream region, $x/d \geq 25$. The role of momentum flux ratio for film cooling with sCO2 needs more attention. As crossflow injection was typically found to increase $\bar{\eta}$, future studies can explore its role as well.

### 4.6 FILM COOLING EFFECTIVENESS: AIR VS. SCO2

The laterally averaged effectiveness presented in Figure 10 is presented in Figure 11 but with an emphasis on comparing film cooling performance for air and sCO2.

![Figure 11: Laterally averaged effectiveness at a) BR=0.5 & c) BR=1.0; Difference in laterally averaged values between air and sCO2 at b) BR=0.5 & d) BR=1.0](image)

The primary objective of this study was to investigate whether film cooling can be pursued as a potential thermal solution for sCO2 turbines. Irrespective of the blowing ratio, the $\bar{\eta}$ with sCO2 was found to be lower than with air at any given density ratio. However, at the lowest blowing ratio (BR=0.5), the difference in $\bar{\eta}$ appears to be really low (Figure 11b), especially after $x/d = 15$. The maximum difference was found to occur between $4 \leq x/d \leq 6$. Interestingly, the laterally averaged effectiveness is still higher than 0.2 up until $x/d \sim 20$ at both density ratios, indicating that film cooling can be viable at these conditions for the sCO2 turbines.

At higher blowing ratio ($BR = 1.0$) a, significant reduction in the $\bar{\eta}$, especially in the near hole region ($2 \leq x/d \leq 7$), makes it difficult to recommend a cylindrical hole for film cooling sCO2 turbines. At higher density ratio ($DR = 2.0$), the $\bar{\eta}$ consistently stays higher than 0.2 over $x/d \geq 8$. Future studies can look at the effect of hole exit shaping (shaped holes) on higher blowing ratios to improve effectiveness near the hole exit.

Figure 12 shows the local distribution of adiabatic effectiveness at different blowing ratios and density ratios. The coolant flow direction beneath the flat plate is towards the $+ve$ $z$ direction. At $BR = 0.5$ & $DR = 1.5$, the location of
the peak effectiveness for air and sCO$_2$ appear shifted from the centerline ($z/d = 0$) but are on the either side. However, at higher density ratio, they both seem to be shifted towards the $+ve$ $z$ value. At higher blowing ratio, the peak effectiveness trend appears to be consistent with density ratio. The effect of crossflow Mach number on local film effectiveness (contours) has been presented by Gritsch et al. [35]. Future studies can focus on analyzing the velocity distribution inside the film cooling hole for both fluids under the effect of crossflow.

Figure 12: Adiabatic or film effectiveness distribution downstream of cylindrical hole exit. a) BR = 0.5 & DR = 1.5, b) BR = 0.5 & DR = 2.0, c) BR = 1.0 & DR = 1.5, and d) BR = 1.0 & DR = 2.0
5 CONCLUSIONS

The current study explores the possibility of cooling the vanes and blades of a direct-fired sCO\textsubscript{2} turbine using film cooling geometries. The operating conditions of a direct-fired sCO\textsubscript{2} cycle and thermophysical properties of the fluid at those conditions can alter the flow field characteristics of the coolant jet and its mixing with the mainstream. Very little information is present in the literature regarding the performance of film cooling geometries employing supercritical CO\textsubscript{2}. A computational fluid dynamic model is used to study the coolant jet exiting a cylindrical hole located on a flat plate, with the coolant being fed by an internal channel. Steady-state RANS equations were solved along with the SST kω model to provide the turbulence closure. Numerical predictions revealed that the crossflow effects and jet lift-off were more pronounced in the case of sCO\textsubscript{2} when compared to a gas turbine. Spatial distribution of cooling effectiveness and laterally averaged adiabatic effectiveness were presented at different operating conditions. At low blowing ratios (BR=0.5), film cooling appears to be a viable thermal solution for sCO\textsubscript{2} turbines based on the conditions studied. At higher blowing ratios, shaped cylindrical holes can be considered as significant jet separation was observed in the near hole region. Recommendations for future studies were also given wherever applicable.
6 NOMENCLATURE

BR  Blowing ratio
CFD  Computational fluid dynamics
D  Film cooling hole diameter
DES  Detached eddy simulation
DNS  Direct numerical simulations
DR  Density ratio
\( l \)  Momentum flux ratio
LES  Large eddy simulations
RANS  Reynolds averaged Navier-Stokes
RKE  Realizable k-epsilon (\( k\varepsilon \))
SAS  Scale adaptive simulation
sCO\(_2\)  Supercritical carbon dioxide
SKW  Standard k-omega (\( k\omega \))
SST-KW  Shear stress transport k-omega (\( k\omega \))
\( V \)  Velocity
VR  Velocity ratio
\( \rho \)  Density
\( \eta \)  Film effectiveness

Subscripts
\( c \)  Coolant
\( \text{ch} \)  Channel
\( g \)  Hot gas
7 REFERENCES


