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USING CONJUGATE HEAT TRANSFER TO ASSESS THE COOLING

PERFORMANCE ON A TURBINE ENDWALL

A Dissertation in

Mechanical Engineering

by

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ABSTRACT

Advancements in cooling for applications such as gas turbines components require improved understanding of the complex heat transfer mechanisms and the interactions between those mechanisms. Turbine designers often rely on multiple thermal protection techniques, including internal cooling, external film cooling, and thermal barrier coatings to efficiently cool components and limit the use of coolant. Traditionally, the effectiveness of such cooling technologies is quantified by considering the convection and conduction heat transfer mechanisms separately. The current research considers the combined effects of both internal and external cooling in a single experiment or simulation using a conjugate heat transfer approach.

The geometry used for this study is a turbine blade endwall, which is influenced by three-dimensional vortices generated by the passage flow. The experiments and computational simulations include impingement and film cooling as well as conduction through the endwall. Appropriate geometric and flow parameters are properly scaled to ensure engine relevant dimensionless temperatures are obtained. The overall effectiveness, which is a scaled wall temperature, is compared for multiple cases, including different cooling configurations and the implementation of endwall contouring, thermal barrier coatings, and contaminant deposition. The measurements showed that impingement cooling was a larger contributor to the combined overall effectiveness compared to film cooling. The area-averaged contoured endwall overall effectiveness was similar to the flat endwall despite local differences in film cooling effectiveness and impingement effectiveness. The thermal barrier coating significantly increased overall effectiveness by reducing the external heat transfer. Contaminant deposition increased surface roughness, which increased external heat transfer to the endwall.

Computational simulations of conjugate heat transfer and time-resolved flowfield measurements helped to understand the complex mechanisms acting together to generate the overall cooling effectiveness. The predicted endwall temperature fields show the three-dimensional temperature gradients present in conjugate heat transfer. The flat and contoured endwall flowfield measurements show that film cooling jets and the passage vortex interact to generate the secondary flowfield impacting endwall heat transfer. Several recommendations for mitigating increased heat transfer and optimizing the cooling performance for gas turbine endwalls are given.
TABLE OF CONTENTS

List of Figures ......................................................................................................................... vii
List of Tables ........................................................................................................................... xiv
Nomenclature ............................................................................................................................ xv
Acknowledgments .................................................................................................................. xviii

Chapter 1. Introduction ............................................................................................................. 1
1.1 Background and Motivation ............................................................................................... 1
1.2 Conjugate Model Development ....................................................................................... 6
1.3 Objectives ........................................................................................................................ 7
1.4 Outline of Dissertation .................................................................................................... 8

Chapter 2. Review of Relevant Literature .............................................................................. 10
2.1 Conjugate Heat Transfer for Gas Turbine Applications .................................................. 10
2.1.1 Flat Plate Studies ........................................................................................................ 11
2.1.2 Leading Edge Studies ................................................................................................. 12
2.1.3 Airfoil Studies ............................................................................................................. 12
2.2 Impingement Heat Transfer ............................................................................................. 14
2.3 Flow and Heat Transfer Effects of Endwalls and Endwall Contouring ......................... 15
2.4 Thermal Effects of Thermal Barrier Coatings ................................................................ 18
2.5 Simulations of Multiphase Particle Deposition .............................................................. 19
2.6 Summary and Uniqueness. ............................................................................................... 21

Chapter 3. Overall Effectiveness of a Blade Endwall With Jet Impingement and Film Cooling ......................................................................................................................... 24
3.1 Introduction ...................................................................................................................... 24
3.2 Relevant Literature .......................................................................................................... 25
3.3 Conjugate Endwall Surface ............................................................................................. 27
3.4 Experimental Methods ................................................................................................... 29
3.5 Measurement Methods and Uncertainty .......................................................................... 33
3.6 Results and Discussion ................................................................................................... 35
3.6.1 Film Cooling Only Results ........................................................................................... 37
3.6.2 Impingement Cooling Only Results ............................................................................. 38
3.6.3 Combined Film and Impingement Cooling ................................................................. 39
3.6.4 Comparison of Individual and Combined Cooling Effects ......................................... 39
3.6.5 One-Dimensional Calculation of Combined Effectiveness Based on Individual Cooling Features ............................................................................................................ 42
3.7 Conclusions .................................................................................................................... 46

Chapter 4. Conjugate Heat Transfer Analysis of the Effects of Impingement Channel Height for a Turbine Blade Endwall ......................................................................................... 47
Abstract .................................................................................................................................. 47
Chapter 5. Overall Effectiveness and Flowfield Measurements for an Endwall With Non-Axisymmetric Contouring ........................................... 71
Abstract .................................................................................. 71
5.1 Introduction ......................................................................... 71
5.2 Relevant Literature ............................................................... 72
5.3 Conjugate Endwall Model .................................................... 75
5.4 Experimental and Computational Methods ......................... 77
5.4.1 Experimental Methods ..................................................... 77
5.4.2 Computational Methods .................................................. 81
5.4.3 Flowfield Measurement Methods ..................................... 83
5.5 Results and Discussion .......................................................... 84
5.5.1 Effects of Contouring on Overall Effectiveness ................. 84
5.5.2 Effects of Film Cooling on Contoured Flowfield ............... 88
5.6 Conclusions ....................................................................... 92

Chapter 6. Effects of Non-Axisymmetric Endwall Contouring and Film Cooling on the Passage Flowfield in a Linear Turbine Cascade ................. 94
Abstract .................................................................................. 94
6.1 Introduction ......................................................................... 94
6.2 Experimental Methods ....................................................... 95
6.3 Results and Discussion ........................................................ 102
6.3.1 Measurements at the Passage Exit Plane ....................... 103
6.3.2 Measurements along the Streamwise Direction ............... 111
6.3.3 Unsteadiness of the Passage Vortex ............................... 118
6.4 Conclusions ...................................................................... 119

Chapter 7. Conjugate Heat Transfer Measurements and Predictions of a Blade Endwall With a Thermal Barrier Coating ......................................... 121
Abstract .................................................................................. 121
7.1 Introduction ......................................................................... 121
7.2 Relevant Literature .............................................................. 122
7.3 Conjugate Endwall Model .................................................. 124
7.4 Experimental Methods ........................................................ 127
7.5 Computational Methodology ............................................ 131
7.6 Results and Discussion ........................................................ 134
7.6.1 Measured and Predicted Temperatures Without TBC ....... 134
7.6.2 Measured and Predicted Temperatures With TBC ............ 139
7.7 Conclusions ...................................................................... 144
Chapter 8. Simulations of Multiphase Particle Deposition on a Gas Turbine Endwall With Impingement and Film Cooling ................................................................. 145
  Abstract ........................................................................................................... 145
  8.1 Introduction ................................................................................................ 145
  8.2 Relevant Literature ................................................................................. 146
  8.3 Conjugate Endwall and Deposition Model .............................................. 148
  8.4 Experimental Methods .......................................................................... 151
  8.5 Results and Discussion ........................................................................... 156
    8.5.1 Effects of Deposition on External Temperatures ............................... 156
    8.5.2 Effect of Deposition on Internal Temperatures ................................. 161
  8.6 Conclusions .............................................................................................. 162

Chapter 9. Conclusions ...................................................................................... 163
  9.1 Recommendations for the Turbine Designer .......................................... 166
  9.2 Recommendations for Future Work ....................................................... 168

References ........................................................................................................ 170

Appendix A. Experimental Design and Construction ....................................... 179
  A.1 Thermal Barrier Coating Assembly ....................................................... 179
  A.2 Particle Image Velocimetry Detailed Experimental Methods .................. 180

Appendix B. One- and Two-Dimensional Heat Transfer Calculations ............ 182
  B.1 Derivation of Overall Effectiveness Equations ....................................... 182
  B.2 One Dimensional Calculation of Internal Heat Transfer Coefficients ....... 184
  B.3 Measurements of Internal Coolant Warming Factor ............................... 186

Appendix C. Contoured Endwall Effectiveness Measurements and Predictions .. 188
  C.1 Contoured Endwall Overall Effectiveness Spatially Resolved Measurements .. 188
    C.1.1 Film Cooling Only Measurements .................................................... 188
    C.1.2 Impingement Cooling Only Measurements ........................................ 188
  C.2 Contoured Endwall Overall Effectiveness With TBC ............................ 190
    C.2.1 Film Cooling Only Measurements with TBC ...................................... 190
    C.2.2 Impingement Only Measurements with TBC .................................... 191
    C.2.3 Combined Film and Impingement Results with TBC ....................... 193
  C.3 Contoured Endwall TBC Effectiveness .................................................. 195
  C.4 Conclusions .............................................................................................. 196

Appendix D. Uncertainty Analysis .................................................................... 198
  D.1 Uncertainty in Overall Effectiveness ..................................................... 198
  D.2 Uncertainty in Coolant Flowrate ............................................................ 199
  D.3 Uncertainty in Velocity Measurements with PIV .................................... 200
LIST OF FIGURES

Figure 1.1 Schematic of an F100-PW-220 aircraft gas turbine showing major engine components, adapted from (“http://www.f-16.net,” 2015). ................................................................. 2

Figure 1.2 Cross section of a typical aviation engine combustor and first stage vane and blade (Bunker, 2009). ........................................................................................................ 3

Figure 1.3 Sources of contaminant deposition in turbines: (a) IGCC schematic showing coal gasification, and (b) volcanic ash deposition on the vanes of an aircraft engine (Hamed et al., 2006). ......................................................................................................................... 4

Figure 1.4 Evolution of allowable gas turbine inlet temperature due to advancements in turbine cooling and protective materials (Kehlhofer, 2009). ........................................................................................................ 5

Figure 1.5. Configuration of a conjugate wall with (a) impingement and film cooling, (b) film cooling only, and (c) impingement cooling only. ........................................................................................................ 6

Figure 2.1. Three dimensional secondary flow model from Langston (1980) ......................... 15

Figure 2.2 Heat transfer measurements for the Pack-B cascade at Re_{exit} of 2x10^5 (Lynch et al., 2011b), (a) Nusselt number contours for the flat endwall, (b) heat transfer augmentation contours due to endwall contouring................................................................................. 17

Figure 3.1 Depiction of the (a) large-scale low-speed wind tunnel, with a corner test section housing the Pack-B cascade, and (b) the coolant loop with auxiliary cooling capability and the inlet flow development section........................................................................................................ 30

Figure 3.2 Schematic of the Pack-B linear blade cascade with blade and passage numbering and top view of the conjugate endwall. ........................................................................................................ 31

Figure 3.3 Pack-B cascade static pressure distribution at the blade midspan compared to a CFD prediction (Lynch et al., 2011a). ........................................................................................................ 32

Figure 3.4 Schematic of the internal and external cooling scheme from the side view (a) and the top view (b). .......................................................................................................................... 33

Figure 3.5 Discharge coefficient measured as a function of pressure ratio compared to Burd and Simon (1999) and Barringer et al. (2002). ........................................................................................................ 35

Figure 3.6 Coolant and wall temperatures of the conjugate with film and impingement cooling. 36

Figure 3.7 Contours of $\phi_f$ for blowing ratios: (a) $M_{avg} = 0.6$, (b) $M_{avg} = 1.0$, (c) $M_{avg} = 2.0$, with 30° inclined holes and plenum boundaries overlaid, and (d) pitchwise laterally averaged $\phi_f$ plotted as a function of axial distance................................................................. 37
Figure 3.8 Contours of $\phi_o$ for blowing ratios: (a) $M_{avg} = 0.6$, (b) $M_{avg} = 1.0$, (c) $M_{avg} = 2.0$, with 90° impingement holes and plenum boundaries overlaid, and (d) pitchwise laterally averaged $\phi_o$ plotted as a function of axial distance. ................................................. 38

Figure 3.9 Contours of $\phi$ for: (a) $M_{avg} = 0.6$, measured (b) $M_{avg} = 1.0$, measured (c) $M_{avg} = 1.0$, predicted, (d) $M_{avg} = 2.0$, measured, and (e) $M_{avg} = 2.0$, predicted, with 30° inclined film holes, 90° impingement holes, and plenum boundaries overlaid. ................................................. 40

Figure 3.10 Area averaged $\phi$ (using area outlined in Figure 3.4b) plotted as a function of blowing ratio for all three cooling configurations. .................................................................................................. 40

Figure 3.11 Pitchwise laterally averaged $\phi$ plotted as a function of axial distance for the three cooling configurations at $M_{avg} = 1.0$. .................................................................................................. 41

Figure 3.12 Overall effectiveness of all cooling configurations plotted as a function of $y/p$ at $x/C_{ax} = 0.22$ for (a) $M_{avg} = 0.6$, (b) $M_{avg} = 1.0$, and (c) $M_{avg} = 2.0$. ......................................................... 42

Figure 3.13 Coolant and wall temperatures of the conjugate wall with (a) film cooling only, and (b) impingement cooling only. .......................................................................................................... 44

Figure 3.14 Comparison of laterally averaged $\phi_{calc}$ and $\phi_{meas}$ plotted as function of axial distance for all three blowing ratios. .................................................................................................. 45

Figure 4.1 Important parameters and temperatures of the conjugate wall with (a) film and impingement cooling and (b) impingement cooling only. .................................................................................................. 53

Figure 4.2 Depiction of the (a) large-scale low-speed wind tunnel, with a corner test section housing the Pack-B cascade, and (b) the coolant loop with auxiliary cooling capability and the inlet flow development section. .................................................................................................. 56

Figure 4.3 Schematic of the Pack-B linear blade cascade with blade and passage numbering and top view of the conjugate endwall. .................................................................................................. 57

Figure 4.4 Schematic of internal and external cooling scheme from the side view (a) and the top view (b), also showing area average outline and locations of internal thermocouples. ......... 59

Figure 4.5 Depiction for the case of $H/D = 2.9$ of (a) the computational domain and boundary conditions, (b) the surface grid for the endwall, (c) the prism layer volume grid in the mainstream, channel, and plenum, and (d) the volume grid in the holes and impingement channel. .................................................................................................. 61

Figure 4.6 Contours of measured $\phi_o$ for (a) $M_{avg} = 1.0$, $H/D = 2.9$ (b) $M_{avg} = 1.0$, $H/D = 0.6$ (c) $M_{avg} = 2.0$, $H/D = 2.9$ (d) $M_{avg} = 2.0$, $H/D = 0.6$, with 90° impingement holes and plenum boundaries overlaid. 63

Figure 4.7 Contours of measured $\phi$ for (a) $M_{avg} = 1.0$, $H/D = 2.9$ (b) $M_{avg} = 1.0$, $H/D = 0.6$, (c) $M_{avg} = 2.0$, $H/D = 2.9$, (d) $M_{avg} = 2.0$, $H/D = 0.6$, with 30° inclined film holes, 90° impingement holes, and plenum boundaries overlaid. .................................................................................................. 64
Figure 4.8 Pitchwise laterally averaged effectiveness plotted as a function of axial distance for different values of impingement channel spacing, H/D, (a) measured for impingement only, and (b) measured and predicted for film and impingement. .......................................................... 65

Figure 4.9 Contours of predicted $\phi$ for (a) $M_{avg} = 1.0$, H/D = 2.9, (b) $M_{avg} = 1.0$, H/D = 5.8, (c) $M_{avg} = 1.0$, H/D = 2.9 (d) $M_{avg} = 1.0$, H/D = 0.6, (e) $M_{avg} = 2.0$, H/D = 10.2, (f) $M_{avg} = 2.0$, H/D = 5.8 (g) $M_{avg} = 2.0$, H/D = 2.9, (h) $M_{avg} = 2.0$, H/D = 0.6, with 30° inclined film holes, 90° impingement holes, and plenum boundaries overlaid. .......................................................... 66

Figure 4.10 Predicted in-plane streamlines and contours of non-dimensional temperature in the impingement channel for the first impingement hole in the last row for $M_{avg} = 2.0$ and (a) H/D = 10.2 (b) H/D = 2.9, (c) H/D = 0.6. ...................................................................................................................... 67

Figure 4.11 Measured and predicted area-averaged $\phi$ and $Nu_{D,i}$ plotted as a function of impingement channel height for $M_{avg} = 1.0$ and 2.0. ...................................................................................................................... 69

Figure 5.1 Heat transfer measurements for the Pack-B cascade at $Re_{exit}$ of 2x10^5 (Lynch et al., 2011b), (a) Nusselt number contours for the flat endwall, (b) heat transfer augmentation contours due to endwall contouring with box for area-averaging. ...................................................................................................................... 74

Figure 5.2 One-dimensional conjugate model of a conducting endwall with impingement and film cooling. ...................................................................................................................... 75

Figure 5.3 Depiction of (a) the large-scale low-speed wind tunnel, (b) the test section containing the Pack-B linear blade cascade and conjugate endwall, and (c) the side view of the plenum and impingement channel for the flat endwall. ...................................................................................................................... 77

Figure 5.4 Pack-B cascade static pressure distribution at the blade midspan compared to CFD predictions. ...................................................................................................................... 78

Figure 5.5 Comparison of oil flow visualization of endwall streaklines (Lynch et al., 2011b) with film cooling hole inlet and outlet locations for the (a) flat and (b) contoured endwalls, and (c) qualitative representation of the contoured endwall height. ...................................................................................................................... 81

Figure 5.6 Depiction of (a) the computational domain and boundary conditions and (b) the prism layer volume grid in the holes and impingement channel. ...................................................................................................................... 82

Figure 5.7 Planes measured with PIV (a) shown from above and (b) shown from the view of Plane C overlaid with flat endwall CFD tke contours for $M_{avg} = 2.0$. ...................................................................................................................... 83

Figure 5.8 Contoured endwall overall effectiveness for (a) $M_{avg} = 1.0$ measured, (b) $M_{avg} = 1.0$ predicted, (c) $M_{avg} = 2.0$ measured, and (d) $M_{avg} = 2.0$ predicted. ...................................................................................................................... 85

Figure 5.9 Measured endwall overall effectiveness for the flat endwall (Mensch & Thole, 2014) at (a) $M_{avg} = 0.6$, (b) 1.0, and (c) 2.0, and for the contoured endwall at (d) $M_{avg} = 0.6$, (e) 1.0, and (f) 2.0... ...................................................................................................................... 86
Figure 5.10 Laterally averaged overall effectiveness for (a) film and impingement, (b) film cooling only, and (c) impingement only, measured for the flat and contoured endwalls........... 87

Figure 5.11 In-plane time-averaged streamlines measured with PIV, colored by velocity magnitude for the contoured endwall for (a-c) no film cooling, (d-f) $M_{avg} = 1.0$, and (g-i) $M_{avg} = 2.0$.......................................................... 89

Figure 5.12 Turbulent kinetic energy measured with PIV for the contoured endwall for (a-c) no film cooling, (d-f) $M_{avg} = 1.0$, and (g-i) $M_{avg} = 2.0$.......................................................... 91

Figure 6.1 Depiction of (a) the large-scale low-speed wind tunnel, (b) the test section containing the Pack-B linear blade cascade and conjugate endwall, and (c) the side view of the plenum and impingement channel for the flat endwall. .......................................................................................................................... 96

Figure 6.2 Pack-B cascade static pressure distribution at the blade midspan compared to CFD predictions.................................................................................................................. 97

Figure 6.3 Comparison of oil flow visualization of endwall streaklines (Lynch et al., 2011b) with film cooling hole inlet and outlet locations for the (a) flat and (b) contoured endwalls, and (c) qualitative representation of the contoured endwall height.......................................................................................................................... 99

Figure 6.4 Planes measured with PIV shown (a) for the flat endwall from above, (b) for the contoured endwall from above, and (c) looking upstream from of Plane C overlaid with flat endwall CFD tke contours for $M_{avg} = 2$ (Mensch & Thole, 2015a)........................................................................ 101

Figure 6.5 Flow streamtraces of near endwall flow swept into the passage vortex, from CFD predictions for $M_{avg} = 2$ (Mensch & Thole, 2015a) for the (a) flat endwall top view, (b) flat endwall side view, (c) contoured endwall top view, (b) contoured endwall side view. ............. 103

Figure 6.6 Contours of time-averaged vorticity overlaid with the secondary velocity vectors measured in Plane C for the flat endwall with (a) no film cooling, (b) $M_{avg} = 1.0$, and (c) $M_{avg} = 2.0$.......................................................................................................................... 105

Figure 6.7 Contours of time-averaged vorticity overlaid with the secondary velocity vectors measured in Plane C for the contoured endwall with (a) no film cooling, (b) $M_{avg} = 1.0$, and (c) $M_{avg} = 2.0$. ...................................................................................................................... 108

Figure 6.8 Contours of turbulent kinetic energy overlaid with the secondary velocity vectors measured in Plane C for the flat endwall with (a) no film cooling, (b) $M_{avg} = 1.0$, and (c) $M_{avg} = 2.0$ .......................................................................................................................... 109

Figure 6.9 Contours of turbulent kinetic energy overlaid with the secondary velocity vectors measured in Plane C for the contoured endwall with (a) no film cooling, (b) $M_{avg} = 1.0$, and (c) $M_{avg} = 2.0$ .......................................................................................................................... 110

Figure 6.10 Contours of time-averaged spanwise (z) velocity overlaid with time-averaged velocity vectors in Planes A and B for the flat endwall for (a-b) no film cooling, (c-d) $M_{avg} = 1.0$, and (e-f) $M_{avg} = 2.0$...................................................................................................................... 112
Figure 6.11 Contours of time-averaged spanwise (z) velocity overlaid with time-averaged velocity vectors in Planes A and B for the contoured endwall for (a-b) no film cooling, (c-d) $M_{\text{avg}} = 1.0$, and (e-f) $M_{\text{avg}} = 2.0$...

Figure 6.12 Contours of turbulent kinetic energy overlaid with time-averaged velocity vectors in Planes A and B for the flat endwall for (a-b) no film cooling, (c-d) $M_{\text{avg}} = 1.0$, and (e-f) $M_{\text{avg}} = 2.0$...

Figure 6.13 Contours of turbulent kinetic energy overlaid with time-averaged velocity vectors in Planes A and B for the contoured endwall for (a-b) no film cooling, (c-d) $M_{\text{avg}} = 1.0$, and (e-f) $M_{\text{avg}} = 2.0$...

Figure 6.14 Histogram contours of velocity magnitude in Plane B at the trailing edge with the time-averaged velocity profile in black for (a) the flat endwall and (b) the contoured endwall.

Figure 7.1 Configuration of a conjugate endwall with impingement and film cooling and TBC.

Figure 7.2 Depiction of (a) the large-scale low-speed wind tunnel, and (b) the test section containing the Pack-B linear blade cascade and conjugate endwall.

Figure 7.3 Pack-B cascade static pressure distribution at the blade midspan compared to CFD predictions.

Figure 7.4 Schematic of internal and external cooling scheme from (a) the side view and (b) the top view showing the outline of the TBC and discrete thermocouple measurements taken on the endwall in the experiments.

Figure 7.5 Depiction of (a) the computational domain and boundary conditions, (b) the surface grid for the endwall and TBC, (c) the prism layer volume grid in the holes and impingement channel, and (d) the volume grid in the mainstream, channel, and plenum.

Figure 7.6 Overall effectiveness contours for $M_{\text{avg}} = 1.0$ (a) measured without TBC, (b) predicted without TBC, and (c) predicted under the TBC.

Figure 7.7 Overall effectiveness contours for $M_{\text{avg}} = 2.0$ (a) measured without TBC, (b) predicted without TBC, and (c) predicted under the TBC.

Figure 7.8 Comparison of overall effectiveness with and without TBC, showing measured and predicted values, along inviscid streamlines, PS for a$\text{–}c$ and SS for d$\text{–}f$.

Figure 7.9 Conjugate CFD prediction of non-dimensional temperature in the fluid and the solid at different two slices (a) at the first row of impingement holes and (b) at the second row of impingement holes.

Figure 7.10 Measured and predicted improvement with TBC, $\Delta\phi_{TBC}$, and the predicted $\Delta q_r$ for the external endwall surface plotted as a function of $M_{\text{avg}}$. 
Figure 7.11 Contours of TBC effectiveness for (a) $M_{\text{avg}} = 0.6$ measured, (b) $M_{\text{avg}} = 1.0$ measured, (c) $M_{\text{avg}} = 2.0$ measured, (d) $M_{\text{avg}} = 1.0$ predicted, and (e) $M_{\text{avg}} = 2.0$ predicted. .......................... 142

Figure 7.12 Comparison of TBC effectiveness with film and impingement cooling, showing measured and predicted values, along inviscid streamlines, for (a) $M_{\text{avg}} = 0.6$, (b) $M_{\text{avg}} = 1.0$, and (c) $M_{\text{avg}} = 2.0$. ............................................................................................................................. 143

Figure 8.1 Configuration of a conjugate endwall with impingement and film cooling and simulated deposition. .................................................................................................................................. 149

Figure 8.2 Depiction of (a) the large-scale low-speed wind tunnel, and (b) the test section containing the Pack-B linear blade cascade and conjugate endwall. ............................................................. 152

Figure 8.3 Schematic of internal and external cooling scheme from the top view, also showing the area average outline and locations of internal thermocouples. ......................................................... 154

Figure 8.4 Schematic of two-nozzle wax injection system located in the turbulence grid........ 155

Figure 8.5 Film cooling only contours of (a) $\phi_f$ without deposition for $M_{\text{avg}} = 0.6$, (b) wax effectiveness, $\omega_f$, and deposition photographs for $M_{\text{avg}} = 0.6$, (c) $\phi_f$ for $M_{\text{avg}} = 1.0$, and (d) $\omega_f$, and deposition photographs for $M_{\text{avg}} = 1.0$. ......................................................................................................................... 158

Figure 8.6 Film and impingement cooling contours (a) $\phi$ without deposition for $M_{\text{avg}} = 0.6$, (b) wax effectiveness, $\omega$, and deposition photographs for $M_{\text{avg}} = 0.6$, (c) $\phi$ for $M_{\text{avg}} = 1.0$, and (d) $\omega$, and deposition photographs for $M_{\text{avg}} = 1.0$. ......................................................................................................................... 158

Figure 8.7 Laterally averaged overall effectiveness without deposition, $\phi$, and wax effectiveness, $\omega$, across the passage for $M_{\text{avg}} = 0.6$ and 1.0 for (a) film cooling only, and (b) film and impingement. .................................................................................................................................. 159

Figure 8.8 Area average overall effectiveness without deposition, $\phi$, area average wax effectiveness $\omega$, and average internal effectiveness with and without deposition, $\phi_i$, $\phi_{i,\text{dep}}$, for film cooling only and combined film and impingement at different blowing ratios.......................... 160

Figure A.1 Photographs of the assembly of endwall with TBC: (a) ribbon thermocouples adhered to the endwall with epoxy and Kapton® tape, and (b) cork layer adhered to the flat endwall with holes cut for the film cooling holes. ........................................................................................................... 180

Figure A.2 Diagram of laser and high speed camera setup for the particle image velocimetry measurements. .............................................................................................................................................. 181

Figure B.1 Schematic of two-dimensional conduction in the endwall. ........................................ 183

Figure B.2 Average Nusselt numbers based on internal wall temperature measurements for the different geometries and flow conditions. ........................................................................................................ 185
Figure B.3. Average measured coolant warming factors, $\chi_f$, for the cases of film cooling only and film and impingement cooling, compared to predictions from an energy balance on the case of impingement only. .................................................................187

Figure C.1 Contoured endwall overall effectiveness measured for film cooling only (a) $M_{avg} = 0.6$, (b) $M_{avg} = 1.0$ and (c) $M_{avg} = 2.0$. ..................................................................................................................189

Figure C.2 Contoured endwall overall effectiveness measured for impingement cooling only (a) $M_{avg} = 0.6$, (b) $M_{avg} = 1.0$ and (c) $M_{avg} = 2.0$. ..................................................................................................................189

Figure C.4 Comparison of impingement only overall effectiveness with and without TBC, for both flat and contoured endwalls, along inviscid streamlines, PS (a)–(c) and SS (d)–(f). ........191

Figure C.3 Comparison of film cooling only overall effectiveness with and without TBC, for both flat and contoured endwalls, along inviscid streamlines, PS (a)–(c) and SS (d)–(f) (streamlines shown in Figure 7.4b). .................................................................192

Figure C.5 Predicted contoured endwall overall effectiveness with TBC for (a) $M_{avg} = 1.0$ and (b) $M_{avg} = 2.0$. ..................................................................................................................193

Figure C.6 Comparison of overall effectiveness with and without TBC, for both flat and contoured endwalls, along inviscid streamlines, PS a–c and SS d–f. .........................195

Figure C.7 Contoured endwall TBC effectiveness, $\tau$, with internal impingement plus film cooling, for: (a) $M_{avg} = 0.6$ measured, (b) $M_{avg} = 1.0$ measured, (c) $M_{avg} = 2.0$ measured, (d) $M_{avg} = 1.0$ predicted, and (e) $M_{avg} = 2.0$ predicted. .................................................................196
LIST OF TABLES

Table 3.1 Conjugate Endwall Parameters ................................................................. 29
Table 3.2 Flow Conditions and Blade Geometry ....................................................... 31
Table 3.3 Experimental Test Matrix ......................................................................... 36
Table 4.1 Conjugate Endwall Parameters ................................................................. 54
Table 4.2 Flow Conditions and Blade Geometry ....................................................... 58
Table 5.1 Conjugate Contoured Endwall Parameters ............................................... 76
Table 5.2 Flow Conditions and Blade Geometry ....................................................... 79
Table 5.3 Distance from the Endwall of the Center of the Low Velocity Region $Z_p/S$ ........................................................................................................ 90
Table 6.1 Flow Conditions and Blade Geometry ....................................................... 98
Table 7.1 Conjugate Endwall and TBC Parameters .................................................. 127
Table 7.2 Flow Conditions and Blade Geometry ....................................................... 129
Table 7.3 Measured Improvement in Overall Effectiveness Due to an Increase in Blowing Ratio and Due to the Addition of TBC ............................................... 140
Table 8.1 Conjugate Endwall and Deposition Parameters ........................................ 151
Table 8.2 Flow Conditions and Blade Geometry ....................................................... 154
Table A.1 PIV Setup and Calculation Parameters ................................................... 181
Table D.1 IR Overall Effectiveness Uncertainty Calculations .................................... 199
Table D.2 PIV Uncertainty Calculations .................................................................... 201
# NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Area</td>
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<tr>
<td>Bi</td>
<td>Biot number ((h_{c,t_w}/k_w))</td>
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<tr>
<td>C(_{ax})</td>
<td>Axial chord length</td>
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<td>C(_d)</td>
<td>Discharge coefficient ((\dot{m}<em>c/(A</em>{holes}\sqrt{2\rho \Delta P})))</td>
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<tr>
<td>C(_p)</td>
<td>Pressure coefficient ((P_s-P_{s,\infty})/(1/2\rho_{\infty}U_{\infty,\infty}^2))</td>
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<td>c(_p)</td>
<td>Specific heat</td>
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<tr>
<td>D</td>
<td>Hole diameter</td>
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<tr>
<td>DR</td>
<td>Density ratio ((\rho_c/\rho_{\infty}))</td>
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<td>H</td>
<td>Impingement gap height</td>
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<td>h</td>
<td>Convective heat transfer coefficient</td>
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<td>I</td>
<td>Momentum flux ratio ((\rho_cU_c^2/\rho_{\infty}U_{\infty,\infty}^2))</td>
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<tr>
<td>k</td>
<td>Thermal conductivity</td>
</tr>
<tr>
<td>L</td>
<td>Length</td>
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<tr>
<td>M</td>
<td>Blowing ratio ((\rho_cU_c/\rho_{\infty}U_{\infty,\infty}))</td>
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<tr>
<td>Ma</td>
<td>Mach number</td>
</tr>
<tr>
<td>(\dot{m})</td>
<td>Mass flowrate</td>
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<tr>
<td>N</td>
<td>Number of measurements</td>
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<tr>
<td>Nu</td>
<td>Nusselt number ((hD/k_{\text{fluid}}))</td>
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<tr>
<td>n</td>
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<td>P</td>
<td>Pressure</td>
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<tr>
<td>PIV</td>
<td>Particle Image Velocimetry</td>
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<tr>
<td>PS</td>
<td>Pressure side streamline</td>
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<td>Pitch length</td>
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<tr>
<td>q</td>
<td>Heat flux</td>
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<td>Thermal resistance ((t/k))</td>
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<tr>
<td>Re</td>
<td>Reynolds number ((\rho_{\infty}U_{\infty,\infty}C_{ax}/\mu_{\infty}))</td>
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<td>(\text{Re}_D)</td>
<td>Impingement Reynolds number ((\rho_{c,\infty}U_cD/\mu_{c,\infty}))</td>
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<td>S</td>
<td>Blade span</td>
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<tr>
<td>SS</td>
<td>Suction side streamline</td>
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</table>
Stk  Stokes number \( (\rho_p d_p^2 U_{\infty,\text{in}}/18 \mu_\infty L) \)

s  Distance along a streamline, streamwise coordinate

T  Temperature

TSP  Thermal Scaling Parameter

t  Thickness

tke  Turbulent kinetic energy \( (3/4[(u')^2+(v')^2]) \)

U  Velocity

\( U_{\text{inviscid}} \)  Average inviscid velocity vector

\( U_{\text{sec}} \)  Secondary velocity vector \( (U_{\text{meas}} - U_{\text{inviscid}}) \)

u'  Fluctuating velocity

x, y, z  Global coordinates, where x is blade axial direction

Greek

\( \chi_\eta \)  Internal coolant warming factor for \( \eta \) \( (T_\infty-T_{\text{c,exit}})/(T_\infty-T_{\text{c,in}}) \)

\( \chi_f \)  Internal coolant warming factor for \( \phi_f \) \( (T_\infty-T_{\text{c,inlet}})/(T_\infty-T_{\text{c,in}}) \)

\( \delta \)  Boundary layer thickness (99\%), uncertainty at 95\% confidence

\( \delta^* \)  Displacement thickness

\( \mu \)  Dynamic viscosity

\( \eta \)  Adiabatic effectiveness \( (T_\infty-T_{\text{aw}})/(T_\infty-T_{\text{c,exit}}) \)

\( \rho \)  Density

\( \phi \)  Overall effectiveness \( (T_\infty-T_w)/(T_\infty-T_{\text{c,in}}) \)

\( \phi_f \)  Overall effectiveness with film cooling \( (T_\infty-T_{w,f})/(T_\infty-T_{\text{c,in}}) \)

\( \phi_o \)  Overall effectiveness with impingement \( (T_\infty-T_{w,o})/(T_\infty-T_{\text{c,in}}) \)

\( \phi_{TBC} \)  Overall effectiveness under the TBC \( (T_\infty-T_w)/(T_\infty-T_{\text{c,in}}) \)

\( \theta \)  Momentum thickness

\( \tau \)  TBC effectiveness \( (T_\infty-T_{\text{TBC}})/(T_\infty-T_{\text{c,in}}) \)

\( \omega \)  Wax effectiveness \( (T_\infty-T_{\text{wax}})/(T_\infty-T_{\text{c,in}}) \)

Subscripts, Accents

\( \overline{\phantom{\text{L}}(\text{L})} \)  Laterally averaged
Area averaged
avg Average
aw Adiabatic wall
c,exit Coolant at film cooling hole exit
c,in Coolant upstream of impingement plate
c,inlet Coolant at film cooling hole inlet
calc Calculated
D Based on the hole diameter
dep Deposition
i Internal
∞ External
∞,in Mainstream conditions at the cascade inlet
f Film cooling only
film External driving temperature location
loc Local
meas Measured
o Impingement cooling only
p Particle
s Static
tot Total
w Wall or external wall surface
ACKNOWLEDGMENTS

I would like to first acknowledge my advisor, Dr. Karen Thole, for the tremendous opportunities you’ve given me, taking me as your student four years ago and supporting me throughout the process of achieving this degree. You believed in me even during times when I doubted myself, and I am incredibly grateful. I also am grateful for the friendships and support of my lab mates who all helped me in some way: Jeff Gibson, Robert Schroeder, Katie Kirsch, Chris Whitfield, Jason Ostanek, Molly Eberly, Alan Thrift, Mike Barringer, Ken Clark, Curtis Stimpson, Jake Snyder, Andrew Coward, Steve Lynch, Shane Haydt, Mike Leonetti and Corey Anderson. I will miss the camaraderie of learning from you and working with you. Thanks also to the undergrad and high school students who were very valuable to me: Jake Snyder, Kyra Thole, Aaron Bardell, Audrey Thole and Jed Gallo. I also received much technical and moral support for this research from several individuals outside of Penn State, including Dr. David Bogard at the University of Texas, Dr. Brent Craven of the FDA, Mark Zelesky of Pratt & Whitney, and Robin Ames and Dr. Seth Lawson of DOE-NETL. My work was funded with support from the DOE-NETL through the University Turbine Systems Research (UTSR) program. This program allowed me the opportunity to network with turbine researchers at annual UTSR conferences around the country. I’m thankful to have developed such a wonderful network during this degree.

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CHAPTER 1. INTRODUCTION

A continuing challenge in advanced cooling applications is understanding the interaction between multiple heat transfer processes, known as conjugate heat transfer. Conjugate heat transfer designates an approach to studying the overall heat transfer in the system, including convective heating and cooling at external surfaces, conduction within solid components, and radiation heat transfer in certain cases. The most effective cooling configurations employ multiple cooling methods, making them well suited to a conjugate approach. The heat transfer in many applications is often three-dimensional and influenced by complicated flow fields and thermal fields. An example of such an application is conjugate heat transfer in gas turbine applications, which is the topic of the following dissertation.

1.1 Background and Motivation

A schematic of a gas turbine is shown in the cutaway image in Figure 1.1. This particular image is an example of a gas turbine in an aircraft engine, but the major components are included in some form in all gas turbine engines, whether land-based or aircraft. Understanding the operation of gas turbines begins with the identification of the three basic components of the Brayton thermodynamic cycle, the compressor, the combustor and the turbine. The first process in the cycle is the pressure rise accomplished when air enters from the left in Figure 1.1 and passes through the compressor. In the combustor, fuel is injected, and the air is rapidly heated by combustion of the fuel in the air. Work is extracted from the high temperature and high pressure combustion products in the turbine section, as the flow rotates the rows of blade airfoils, powering the compressor rotation. The remaining energy is either extracted to generate electricity or ejected for propulsion. The stages of the turbine section incrementally expand the gases using a row of stationary vanes followed by a row of rotating blades.
The analysis of an ideal Brayton cycle provides qualitative trends for the performance of actual gas turbines. The Brayton cycle analysis shows that the overall thermal efficiency increases as the pressure ratio of the compressor increases. In addition to thermal efficiency, other important considerations for designers are the engine power to weight ratio and also allowable material temperatures. Increasing the pressure ratio while maintaining mass flow and fuel input results in higher turbine inlet temperatures. Generally gas turbines operate with turbine inlet temperatures beyond allowable limits for the metal surfaces in the turbine. The durability of the airfoils is maintained through highly sophisticated cooling mechanisms. The blue arrow in Figure 1.1 shows the source of coolant for the turbine components. High pressure air is extracted from the high pressure part of the compressor, bypasses the combustor, and routed to internal passages to the turbine section as shown in Figure 1.2. The amount of coolant extracted from the compressor should be minimized as it represents a loss to the overall efficiency.

The hot sections of gas turbines require multiple cooling mechanisms to provide highly effective cooling performance in these extreme conditions. The airfoil and endwall surfaces in the turbine simultaneously experience convective heating from the external hot combustion gases and internal cooling from relatively cooler air originating from the compressor. The coolant is routed through internal passages removing heat from the internal walls of the parts through serpentine channels often with heat transfer enhancement features such as ribs or pin fins. Some parts of the airfoil are internally cooled by impingement jet mechanisms, which are generated by perforated plates within the internal passages. After the internal cooling passages, coolant can
also pass through angled holes in the walls and generate a protective film of coolant on the external walls on the hot gas side to maximize the use of the coolant. Extensive use of film cooling can be seen in the holes in the surface of a vane and a blade in Figure 1.2. In addition, insulating, high temperature ceramic coatings, known as thermal barrier coatings (TBCs), are often applied on the external surfaces to further protect the parts that are exposed to the highest thermal loads.

Figure 1.2 Cross section of a typical aviation engine combustor and first stage vane and blade (Bunker, 2009).

An additional heat transfer challenge occurs when foreign material, such as coal ash or volcanic ash, deposits on turbine surfaces. An alternative power generation method known as integrated gasification combined cycle (IGCC) uses coal derived fuels in a gas turbine power system as shown in the schematic in Figure 1.3a. In the gasifier, coal is converted to hydrogen and carbon monoxide synthesis gas (syngas), which is used as the fuel instead of natural gas in a conventional industrial gas turbine. A complication of using this alternative fuel technology is residual solid particles that can remain through the gasification process despite efforts to remove them. Aircraft engines can ingest similar ash particles if the plane encounters a volcanic ash cloud. In either case, the ash particles can become molten upstream of the turbine due to the extreme temperatures experienced in the combustor section. Downstream of the combustor, the
particles encounter relatively cooler surfaces along the cooled airfoil components and tend to deposit on those surfaces. The deposition can interfere with the critical turbine cooling technologies and increase heat transfer to the components. Figure 1.3b provides evidence of ash deposition in the turbine section of an aircraft engine and the interference with cooling mechanisms (Hamed et al., 2006).

![Diagram of IGCC schematic showing coal gasification and volcanic ash deposition on the vanes of an aircraft engine](image)

**Figure 1.3 Sources of contaminant deposition in turbines:** (a) IGCC schematic showing coal gasification, and (b) volcanic ash deposition on the vanes of an aircraft engine (Hamed et al., 2006).

The combination of convective and conductive heat transfer processes, through the resulting local metal surface temperature, governs the service life of turbine components. As demands to increase engine performance and overall efficiency have continued, gas turbines have generated higher turbine inlet temperatures, requiring better heat transfer performance. Better heat transfer performance can be achieved by more effective and efficient cooling methods and by improvements to material properties and coatings as shown in Figure 1.4. The ability to evaluate cooling technologies with accurate predictions of metal temperature is essential for continuing advances in cooling effectiveness and efficiency.
Current practice to predict turbine metal temperature involves calculating the solid conduction using analytical or numerical tools while applying boundary conditions based on measurements of the separate convective effects. Most heat transfer literature report internal and external heat transfer coefficients ($h_i$ and $h_\infty$) measured using a constant heat flux or constant temperature boundary condition. These data may not include heat transfer augmentation effects due to film cooling or deposition roughness. In gas turbine systems, which commonly use film cooling, adiabatic film cooling effectiveness ($\eta$), measured with an adiabatic boundary condition, is used as the driving temperature for external convection in the presence of film cooling. The individual $\eta$, $h_i$ and $h_\infty$ are applied to analytical or numerical tools to determine a prediction for the actual metal temperature. An alternative to this isolated heat transfer analysis is direct determination of the non-dimensional surface temperature, referred to as the overall effectiveness ($\phi$), through a conjugate experiment or simulation. A properly scaled conjugate model couples the convective heat transfer and solid conduction, and provides relevant non-dimensional temperature and heat transfer data.
1.2 Conjugate Model Development

To achieve relevant experimental data for overall effectiveness, the appropriate non-dimensional parameters must be matched between the engine and the experimental simulation. A simplified model of the endwall with impingement and film cooling is depicted in Figure 1.5. Figure 1.5 identifies the critical parameters and temperatures for the conjugate model development, the experiments and the analysis. Figure 1.5 includes a sketch of the system temperature profile, through the external flow, film cooling jet, endwall and coolant flow. All non-dimensional temperatures are scaled by taking the difference between the hot mainstream temperature, $T_\infty$, and the temperature of interest and then dividing by the overall temperature difference between $T_\infty$ and the internal coolant temperature at the plenum supplying the coolant, $T_{c,\text{in}}$. To ensure $h_\infty$ is independent of coolant temperature, the external driving temperature is some fluid temperature above the endwall, known as $T_{\text{film}}$, rather than $T_\infty$ (Goldstein, 1971). The use of $T_{\text{film}}$ as the driving temperature for external convection causes the driving temperature to vary at each location depending on the coverage of the film cooling. The adiabatic wall temperature with film cooling, $T_{aw}$, is commonly measured and used as an estimate for $T_{\text{film}}$.

![Figure 1.5. Configuration of a conjugate wall with (a) impingement and film cooling, (b) film cooling only, and (c) impingement cooling only.](image)

From the one-dimensional heat transfer through the endwall in Figure 1.5a, Equation (1.1) can be derived for the dimensionless wall temperature, $\phi$ (Williams et al., 2014). To derive Equation (1.1), the wall overall heat transfer is set equal to the external convection heat transfer. The overall heat transfer is defined to be the heat transfer between the driving temperature for external convection, $T_{\text{film}}$, to the internal coolant temperature, $T_{c,\text{in}}$. Temperature dependencies...
of the heat transfer coefficients are neglected. Equation (1.1) is written in non-dimensional form, revealing the non-dimensional parameters that affect the one-dimensional heat transfer, $\text{Bi}$, $h_\infty/h_i$, and $\chi_\eta\eta$.

$$\phi = \frac{T_\infty - T_w}{T_\infty - T_{c,\text{in}}} = \frac{1 - \chi_\eta\eta}{1 + \text{Bi} + h_\infty/h_i} + \chi_\eta\eta$$  \hspace{1cm} (1.1)

The adiabatic effectiveness, $\eta$, is the non-dimensional adiabatic wall temperature given in Equation (1.2).

$$\eta = \frac{T_\infty - T_{aw}}{T_\infty - T_{c,\text{exit}}}$$  \hspace{1cm} (1.2)

Since $\eta$ is measured in the literature for an adiabatic wall, the normalizing temperature difference is different than that of $\phi$. Therefore, a correction factor, $\chi_\eta$, is included to account for the temperature difference between the plenum, at $T_{c,\text{in}}$, and the film cooling hole exit, at $T_{c,\text{exit}}$ (Williams et al., 2014).

$$\chi_\eta = \frac{T_\infty - T_{c,\text{exit}}}{T_\infty - T_{c,\text{in}}}$$  \hspace{1cm} (1.3)

The $\chi_\eta$ parameter represents the internal heating of the coolant through impingement and in-hole convective heat transfer. The product of these two non-dimensional temperatures, $\chi_\eta\eta$, represents the appropriately scaled non-dimensional film temperature, $T_{\text{film}}$.

Considering the case of two-dimensional conduction with the same convective boundary conditions, the analytical solution also depends on the ratio of the endwall width to endwall thickness. By extension, all geometric dimension ratios and heat transfer coefficient ratios are relevant in the actual three-dimensional geometry. When the geometry and non-dimensional parameters in Equation (1.1) are matched to the typical values in the engine, experiments can provide engine relevant dimensionless temperatures.

### 1.3 Objectives

There are four primary objectives to this dissertation related to conjugate heat transfer for gas turbine endwalls. The first objective is to demonstrate the use of a conjugate heat transfer model that is properly scaled to evaluate the thermal performance of a typical endwall geometry. The study implements proper scaling of the conduction and convection parameters to ensure that
meaningful endwall temperature and heat transfer results are found. A second objective is to understand the effect of various factors on the heat transfer in the endwall. The factors considered represent several relevant considerations already mentioned including: the cooling techniques included (i.e. film cooling and impingement cooling), the application of endwall contouring, the presence of a TBC layer, and the presence of contaminant deposition. The conjugate studies are performed using both experimental methods and computational simulations. A third objective is to assess the accuracy of the numerical predictions of conjugate heat transfer completed using computational fluid dynamics (CFD). Once validated, computational predictions can provide access to temperature and heat transfer data that cannot be easily measured. The final objective is to characterize the relationships between the highly skewed secondary flows over an endwall surface, film cooling, and conjugate heat transfer. The conjugate heat transfer through a wall in gas turbine applications has been approximated as one-dimensional through the wall thickness, according to the model introduced by Albert et al. (2004). This investigation seeks to characterize the secondary flow conditions that contribute to three-dimensional heat transfer effects in the endwall.

1.4 Outline of Dissertation

The following dissertation is presented in manuscript format. Chapter 2 provides an overview of the literature relevant to conjugate heat transfer for an endwall and identifies the uniqueness of the research. Chapters 3 through 8 present the dissertation methods, results and discussion through several peer-reviewed journal papers. The first paper, Chapter 3, provides the baseline conjugate heat transfer measurements for a flat endwall with internal impingement cooling and external film cooling. This chapter also evaluates the relative influence of impingement cooling and film cooling to the overall cooling effectiveness. Chapter 4 investigates the effects of varying the spacing between the impingement plate and the endwall through additional thermal measurements and computational simulations. The effects of non-axisymmetric endwall contouring are evaluated in Chapters 5 and 6. The paper in Chapter 5 focuses on thermal results, presenting measurements and computational predictions of overall cooling effectiveness comparing flat and contoured endwalls. Preliminary flowfield data measured for the contoured endwall with and without film cooling are also included in Chapter 5. The flowfield measurements are further analyzed and expanded to include the flat endwall in
Chapter 6. In Chapter 7, the thermal performance of an engine relevant TBC is quantified using measured and predicted thermal data. The final paper, Chapter 8, presents the results of experimental simulations of multi-phase particle deposition, a potential threat for IGCC turbines and for aircraft engines. The major findings of the individual studies are summarized in Chapter 9 along with the principal conclusions of the dissertation research as a whole. The appendices provide additional details about experimental methods, experimental uncertainty analyses, and effectiveness results for the contoured endwall with TBC.
CHAPTER 2. REVIEW OF RELEVANT LITERATURE

Literature relevant to the current study of conjugate heat transfer in an endwall includes the foundational literature for conjugate heat transfer in gas turbine applications, presented in Section 2.1. Relevant literature on jet impingement heat transfer is discussed in Section 2.2. Section 2.3 describes the unique aspects of endwall flow and heat transfer and the effects of endwall contouring found in the literature. The thermal effect of thermal barrier coatings, studied through conjugate heat transfer, is reviewed in Section 2.4. Section 2.5 reviews the literature regarding simulations of multiphase particle deposition in turbines. A summary of the relevant literature is given in Section 2.6 to highlight the uniqueness of the current work.

2.1 Conjugate Heat Transfer for Gas Turbine Applications

A conjugate heat transfer approach can be applied in experiments and computational simulations to determine the non-dimensional external metal temperature, also known as overall effectiveness, defined in Chapter 1. Overall effectiveness, $\phi$, is defined as the difference between the mainstream temperature and the external metal temperature, divided by the overall mainstream to coolant temperature difference. The $\phi$ definition along with its relationship to the important non-dimensional parameters were developed in Chapter 1. Albert et al. (2004) was the first to derive an equation for $\phi$ by considering one-dimensional heat transfer for gas turbine applications. The equation demonstrated it is necessary to match the Biot number, $Bi$, as well as the ratio of internal-to-external heat transfer coefficients, $h_\infty/h_i$, to acquire relevant scaled data in a conjugate simulation or experiment. The practice of reporting the values of these non-dimensional parameters only exists for more recent studies, although conjugate effects are still evident in studies without matched $Bi$ and $h_\infty/h_i$. Conjugate computational studies, usually performed with RANS, generally agree with corresponding temperature measurements when available. A common exception is for geometries where film cooling is important, since RANS is usually not able to accurately predict film cooling flow. The conjugate heat transfer literature is organized by the type of geometry to which it is applied, including flat plates, leading edge models, and airfoils, including vanes and blades.
2.1.1 Flat Plate Studies

In the gas turbine industry, the conduction within the metal components is often assumed to dominate the heat transfer, smear temperature gradients, and produce nearly constant wall temperature. However, the uniformity of temperature depends on the Bi. Several flat plate studies demonstrated that the wall temperature and overall effectiveness is not necessarily uniform but can vary along the plate due to film cooling. Wang and Zhao (2011) included the conjugate effects of conduction and film cooling for a flat plate with a two-dimensional slot geometry. They compared the results obtained for different wall boundary conditions including adiabatic and conjugate walls. Even for the case of a very low Bi ~ 0.03, Wang and Zhao (2011) showed that the overall effectiveness varied locally and was not completely uniform across the surface. Bohn et al. (2003) also included conduction and film cooling in a conjugate computational study for a three-dimensional flat plate. Bohn et al. compared adiabatic and overall effectiveness for a flat plate with one row of 30° film cooling holes. While adiabatic effectiveness, \( \eta \), varied greatly in the lateral direction, overall effectiveness was more uniform but not constant. In a similar study, Na et al. (2007) reported that laterally averaged overall effectiveness was significantly higher than laterally averaged adiabatic effectiveness.

Flat plate studies with film cooling and internal impingement jet cooling show that the overall performance is influenced more by the impingement than the film cooling. The first experiments to use an engine matched Bi experimental model were completed by Sweeney and Rhodes (2000) for a three-dimensional flat plate with internal impingement and angled film cooling holes. Between the impingement plate and the wall were heat transfer enhancement features in a Lamilloy® snowflake design. Their results showed that the distribution of external wall temperatures was dominated by impingement cooling over film cooling. The temperature distribution varied with different arrangements of impingement jets. When the jets were tightly spaced, the temperature distribution was smooth. When the jet spacing was increased, non-uniformities were observed in the external wall temperatures. Panda and Prasad (2012) also simulated a film cooled plate with and without internal impingement. The authors compared their computational results to experimental measurements along the centerline of the plate and found excellent agreement for predictions using the SST k-\( \omega \) model. Plates with different thermal conductivities (plastic and stainless steel) were considered, but the Bi and \( h_e/h_i \) of these
configurations were not reported. The results showed that the increase in overall effectiveness from adding internal impingement became more substantial as blowing ratio increased.

2.1.2 Leading Edge Studies

To study the stagnation region of a turbine airfoil, leading edge models are created from a half cylinder with extended walls. A series of experimental and computational studies incorporating impingement cooling and film cooling were performed on a conducting leading edge model (Albert et al., 2004; Dobrowolski et al., 2009; Maikell et al., 2011; Mouzon et al., 2005; Ravelli et al., 2010; Terrell et al., 2005). Maikell et al. (2011) measured overall effectiveness in the stagnation region with impingement and round film cooling holes. They observed continuous streaks of high overall effectiveness around the rows of film cooling holes as blowing ratio increased. Dobrowolski et al. (2009) performed conjugate simulations corresponding to the Maikell et al. experiments using the realizable k-ε turbulence model. The effect of impingement was applied by setting the internal wall temperature distribution to the temperature measured in the experiments. The simulations under-predicted the separation of the film cooling jets compared to the experiments. The same leading edge model but with shaped film cooling holes and impingement was considered by Mouzon et al. (2005) and Ravelli et al. (2010) with experiments and simulations respectively. Their results showed that the presence of impingement (H/D~5) did not have much effect on $\phi$ because in-hole convection dominated heat transfer in the stagnation region. The simulations by Ravelli et al. (2010) did show that the arrangement of impingement holes affected the internal flow recirculation and distribution of coolant between the film cooling holes. Terrell et al. (2005) examined contribution of in-hole convection to overall cooling in the stagnation region. Their experimental and computational study measured the total heat transfer to the coolant as it passed through shaped holes in the stagnation region. The heating of the coolant was found to be significant. Therefore, in-hole convection heat transfer and resulting temperature change of the coolant need to be accounted for in analyses of these systems.

2.1.3 Airfoil Studies

Conducting vane and blade geometries using film cooling and internal impingement cooling can provide insight into the thermal behavior and conjugate effects associated with film
and impingement cooling technologies. Experimental work of a conducting vane was pioneered by Hylton et al. (1983, 1988) and Turner et al. (1985) using a C3X vane. Although engine relevance was limited because matched Bi and $h_c/h_i$ were not confirmed, these studies improved understanding of the thermal fields of a conducting vane, and provided experimental data for benchmarking computational work.

Multiple authors (Albert & Bogard, 2013b; Dees et al., 2013; Dyson et al., 2012; Ledezma et al., 2011; Nathan et al., 2013; Williams et al., 2014) examined different areas of a C3X vane constructed to match engine Bi. Nathan et al. (2013) measured overall effectiveness on the stagnation region of the vane, which was cooled internally with impingement and externally with a showerhead arrangement of cooling holes. On the vane pressure side, Albert and Bogard (2013b) found that overall effectiveness decreased with increasing blowing ratio due to film cooling jet detachment. Dees et al. (2013) measured overall effectiveness on the suction side of a vane with an engine matched Bi of 0.4-1.6. Internal cooling was provided by a u-bend passage inside the vane that fed film cooling holes. The conjugate heat transfer was numerically simulated by Ledezma et al. (2011) using the standard k-ω turbulence model, and the results were compared to the experiments by Dees et al. (2013). Differences from the experiments were attributed to poor prediction of film cooling jet separation, the assumption of isotropic turbulence, and unsteady effects.

Internal impingement cooling was added to the suction side of the vane in Williams et al. (2014). In addition to measuring overall effectiveness, the authors tried to analytically predict $\phi$ using a one-dimensional model. They measured adiabatic effectiveness with a separate adiabatic vane and overall effectiveness with impingement only by blocking some film cooling holes in the suction side row. The un-blocked film cooling holes provided a path for the impingement jets to exhaust, while overall effectiveness was measured above the blocked holes. Using this configuration allowed Williams et al. (2014) to assume the internal heat transfer coefficient did not change between the two cases with and without film cooling. The prediction performed reasonably well for a range of momentum flux ratios. Dyson et al. (2012) used the SST k-ω turbulence model to perform numerical simulations of the experiments from Williams et al. (2014). Film cooling jet diffusion was under-predicted in the simulations. The insufficient diffusion led to over-predicted cooling effectiveness for attached jets and under-predicted effectiveness for detached jets.
Ni et al. (2011, 2013) simulated a fully film cooled vane and endwall geometry under flow conditions consistent with a dual spool engine with a pressure ratio of 40. The conjugate simulations were performed using the standard k-ω turbulence model. The results showed significant temperature variations along the vane chord at 5% and 62% span. When the boundary conditions were properly modeled, the predicted surface heat flux and temperature data along the vane chord at the two heights agreed with the point measurements obtained from experiments.

### 2.2 Impingement Heat Transfer

Internal heat transfer coefficients for engine relevant geometries of internal impingement cooling can be found in the papers by Florschuetz et al. (1981) and Hollworth and Dagan (1980). These two studies provide correlations for the Nu as functions of jet Re and geometric parameters with a constant temperature boundary condition. Florschuetz et al. (1981) considered staggered impingement jet geometries where the coolant was extracted laterally from one side. The authors found that the crossflow that developed in the channel generally degraded the heat transfer coefficient from the first row of jets to the exit row. Hollworth and Dagan (1983; 1980) measured the Nu for staggered impingement geometries where the coolant is extracted through angled holes in the target plate, which simulates a configuration with combined impingement and film cooling. Hollworth and Dagan (1980) provided a correlation for the area-averaged Nu for configurations with impingement and film cooling extraction. Although some geometric parameters are included in the impingement correlations, the ratio of impingement holes to extraction holes is not included, and this ratio may differ for realistic endwall geometries such as the one presented in our study. The impingement heat transfer effects of certain parameters, such as the distance between the impingement plate and the target, H, were reviewed by Viskanta (1993). The Nu for the impingement jets usually varied with the impingement channel height to hole diameter ratio, H/D, with a maximum occurring between H/D of 1.5 – 4 depending on the specific jet arrangement and method of Nu measurement (Florschuetz et al., 1981; Hollworth & Dagan, 1980; Viskanta, 1993). For impingement with film cooling extraction, Hollworth and Dagan (1980) found that for the smallest spacing between impingement jets, 5D, there was not much change in Nu for a wide range of H/D = 0.5 – 6.0.
2.3 Flow and Heat Transfer Effects of Endwalls and Endwall Contouring

Highly three-dimensional flow structures complicate the flow across the endwall of a vane or blade cascade compared to flow over a flat plate or flow across an airfoil. One aspect of the secondary flow is the formation of a leading edge vortex, similar to the horseshoe vortex that surrounds the leading edge of a cylinder in crossflow. As the legs of this vortex continue into the passage, they become subject to the strong pressure gradient formed from the turning of the mainstream flow. This pressure gradient pulls the pressure side leg of the vortex across the passage to the suction side of the adjacent blade, shown in Figure 2.1 as the passage vortex. On the other side of the leading edge, the suction side leg of the vortex remains attached to the suction side of the blade. This vortex becomes weaker as it goes downstream, meeting with the passage vortex in the downstream half of the blade. The suction side leg vortex is rotating counter to the larger passage vortex, and is labeled contour vortex in Figure 2.1.

Figure 2.1. Three dimensional secondary flow model from Langston (1980)

The pressure gradient between the pressure side and suction side of the passage induces crossflow along the endwall, which means the pressure gradient skews the direction of the slower moving fluid close to the wall. The primary flow direction of the endwall flow is across the passage, as shown by the short arrows in Figure 2.1, in contrast to the mainstream streamwise direction, which approximately follows the blade surfaces. The endwall crossflow contributes to the growth of the passage vortex as the vortex moves downstream. Measurements of these passage secondary flow structures and other induced vortices can be found in Langston et al. (1977), Wang et al. (1997), and Kang and Thole (2000) among others. The time-resolved
flowfield in a high pressure blade cascade was measured by Pu et al. (2014) to investigate the effects of inlet turbulence levels (2.7% and 10%) and upstream film cooling blowing ratio on the passage vortex. The higher turbulence level increased the size of the passage vortex at the trailing edge and moved the core away from the wall. Six converging slot holes, located in the endwall upstream of the blade, injected film cooling at different blowing ratios. Film cooling did not produce a consistent trend for the size or height of the passage vortex. The result depended on blowing ratio and turbulence level. At the highest blowing ratio the passage vortex was consistently smaller, weaker and closer to the endwall for both turbulence levels.

The secondary flows can locally increase the external heat transfer coefficient, in particular at the leading edge, the pressure side of the passage vortex, and in the wake. Kang et al. (1999) and Radomsky and Thole (2000) measured the distribution of endwall heat transfer coefficients for a high pressure vane cascade at various Reynolds numbers and freestream turbulence levels. The measurements were made using a constant wall heat flux boundary condition. The vane endwall results showed higher heat transfer coefficients at locations where the vortices turned downward and reduced the thickness of the boundary layer, such as at the leading edge and the pressure side of the passage. In addition, regions where the vortices turned away from the endwall corresponded to regions of lower heat transfer coefficients (Kang & Thole, 2000). Measurements by Lynch et al. (2011b) for the endwall of the Pack-B blade cascade showed similar regions of increased heat transfer at the leading edge, pressure side and the wake. The Nusselt number contours are presented in Figure 2.2a for the case of a flat endwall and Figure 2.2b for a contoured endwall discussed later in this section. Lynch et al. (2011b) also noted an increase in heat transfer on the suction side toward the trailing edge, which was attributed to the passage vortex impacting the trailing edge.
Secondary flows in the cascade also generate aerodynamic losses from the dissipation of useful kinetic energy in the rotation of the vortices. A technique for reducing the aerodynamic losses is to contour the shape of the endwall, so that there are local variations in the endwall height. Endwall contouring can be axisymmetric, meaning the height is uniform at a certain axial location. The alternative is non-axisymmetric, which uses a completely three-dimensional surface. With non-axisymmetric contouring, both convex and concave curvature are employed such that the secondary flows are reduced. The design of non-axisymmetric contours can be optimized for different parameters such as minimizing the cross-passage pressure gradient (Harvey et al., 2000) or minimizing aerodynamic losses (Praisner et al., 2007). Knezevici et al. (2010) completed seven-hole probe measurements with and without the endwall contour designed by Praisner et al. (2007) for the Pack-B cascade. They found that with contouring, the strength of the passage vortex was reduced, and the overall secondary kinetic energy was reduced by 13%.

The effect of endwall contouring on the passage vortex at two planes in the passage was measured by Lynch (2011) for the contouring designed by Praisner et al. (2007). The threedimensional average flowfield data were acquired with laser Doppler velocimetry for axial
planes mid-passage at 0.2Cax and at the trailing edge at 1.03Cax. The size and strength of the passage vortex measured data were revealed through the presentation of the data in the form of secondary velocities, secondary kinetic energy and turbulent kinetic energy. At the mid-passage plane the contoured endwall slightly reduced the strength of the passage vortex, but also slightly increased turbulent kinetic energy levels compared to the flat endwall. From the trailing edge flow measurements, it was found the contouring delayed the progression of the passage vortex to the suction side of the passage. Like the mid-passage plane, endwall contouring weakened the passage vortex but increased secondary and turbulent kinetic energy close to the endwall.

The effects of non-axisymmetric endwall contouring on heat transfer were investigated by Saha and Acharya (2008). Their contoured endwall design could achieve an overall reduction in predicted Nusselt number of 8%. This reduction was achieved through the same mechanisms that were used to reduce aerodynamic losses: reducing the cross-passage pressure gradient and the strength of the passage vortex. For the same contoured endwall, Gustafson et al. (2007) measured a reduction in total pressure loss of nearly 50%. Lynch et al. (2011b) measured the relative change in heat transfer coefficients with endwall contouring for the geometry designed by Praisner et al. (2007) and used by Lynch (2011) in the flowfield measurements. The results, given in Figure 2.2b, show that the contouring reduced heat transfer in the regions of the passage where the secondary flows had increased heat transfer in Figure 2.2a. On the pressure side of the passage, where the passage vortex impacts the endwall, a valley in the endwall contributes to reduced heat transfer compared to the flat case. On the downstream suction side, the secondary flows for the flat endwall had produced a local increase in heat transfer, which was mitigated by the contouring. Figure 2.2b also contains regions of significantly increased heat transfer, at the leading edge and upstream portions of the passage. The positive surface gradients thin the boundary layer and contribute to increased heat transfer. The flow encounters positive surface gradients at the upstream half of the hill on the pressure side and the downstream half of the valley on the suction side. Overall, the contouring reduced the area-averaged heat transfer coefficient across the entire passage by 3%, and the overall heat load by 2%.

2.4 Thermal Effects of Thermal Barrier Coatings

Thermal barrier coatings (TBCs) have been applied in many designs due to the potential to provide dramatic reductions in metal temperature (Padture et al., 2002). It is apparent that a
decrease in the conductivity of the TBC will improve the cooling performance. However, the improvement in metal temperature or overall effectiveness can only be quantified through conjugate heat transfer experiments and simulations. The existing literature includes studies that have applied a TBC to leading edge, vane and flat plate geometries.

A computational study of TBC applied to a conjugate wall with film cooling was completed by Na et al. (2007). The flat plate geometry contained a single 30° angled film cooling hole, and accounted for partial blockage of the film cooling hole by the TBC. Without TBC, the wall temperature was relatively uniform across the surface, indicating a low Bi, although the Bi was not stated. With TBC, decreased wall temperatures were reported. The thermal effect of TBC on a conjugate leading edge model was measured in experiments by Maikell et al. (2011). The Bi of 1-2.1 was matched to that expected in an engine. The TBC case had cooler leading edge wall temperatures but hotter external TBC temperatures due to the insulating effect of the TBC. Davidson et al. (2014b) investigated the improvement in wall temperature when TBC was added to a matched Bi vane with Bi of 0.3-1.1. The cooling performance increased significantly with TBC to the point that increasing blowing ratio did not provide a reduction in the vane wall temperature.

2.5 Simulations of Multiphase Particle Deposition

While many studies in the literature have studied the effects of deposition geometry and roughness on gas turbine cooling, dynamic simulation of the deposition process in a laboratory environment has the advantage of producing a distribution of deposition specific to the geometry and testing conditions. Dynamic molten particle deposition methods capture a variety of effects, most notably particle Stokes number (Stk) and molten character of the particles, both which determine the amount and character of the deposition.

Molten coal ash particles were dynamically deposited by Ai et al. (2012), Webb et al. (2012) and Casaday et al. (2013) in studies of deposition and temperature effects. These researchers used accelerated blowdown facilities in which combustion gases deposited the particles on stationary hardware. Ai et al. (2012) optically measured the external temperature on a film cooled surface and found that the surface temperature increased as more particles deposited. Increasing the blowing ratio mitigated the accumulation of deposition and rise in temperature. Webb et al. (2012) tested the consequences of injecting different types of coal ash
at different mainstream temperature conditions and discovered that there was a threshold temperature for deposition that varied depending on the type of ash. As the gas temperature was increased beyond the threshold, the deposit size and thickness increased. Casaday et al. (2013) also found levels of increased deposition due to higher gas temperatures from simulated hot streaks. In both studies, the deposition was thickest on the leading edge and pressure side of the vanes. Film cooling provided some benefit in reducing deposition for the types of ash that had the mildest deposition amounts (Webb et al., 2012).

A method to dynamically simulate molten particle deposition with wax in low speed wind tunnel facilities has been employed by several researchers (Albert & Bogard, 2012, 2013a; Davidson et al., 2014a; Kistenmacher et al., 2014; Lawson et al., 2013; Lawson & Thole, 2011, 2012a, 2012b). Lawson et al. (2013; 2011, 2012a, 2012b) simulated deposition on adiabatic endwall geometries with film cooling. The deposition thickness initially increased over time, but then reached a state of equilibrium, when the rate of deposition was balanced by the rate of erosion. Lawson et al. (2013; 2012a, 2012b) varied the molten character of the particles by changing the mainstream temperature. A Thermal Scaling Parameter (TSP) was defined to quantify the molten character of the particles. For a lower value of TSP, corresponding to more solid than liquid particles, the deposition increased surface roughness and reduced adiabatic effectiveness compared to the higher value of TSP.

A similar wax deposition facility was employed by Albert and Bogard (2012, 2013a), Davidson et al. (2014a) and Kistenmacher et al. (2014) for conducting vane geometries with internal cooling and film cooling. The appropriate conjugate parameters of $\text{Bi}$ and $h_c/h_i$ were matched to engine conditions. Albert and Bogard (2012, 2013a) observed significant amounts of deposition on the leading edge and pressure sides of the vane. However, there was threshold for surface temperature above which a significant amount of deposition was observed. When the vane surface temperature was below the threshold, deposition was partially mitigated. In light of the importance of surface temperature to deposition, Davidson et al. (2014a) and Kistenmacher et al. (2014) included the thermal effect of a TBC on the vane in their deposition simulations. Both wax and TBC thermal conductivities were appropriately scaled to the vane conductivity. Measurements of the vane temperature, under the TBC, showed lower temperatures for the case with deposition relative to no deposition due to the insulating effect of the deposition layer.
Adiabatic studies of endwall deposition have focused on the deposition effects to film cooling effectiveness, but have shown deposition patterns associated with endwall flows. Lawson and Thole (2012a, 2012b) measured the adiabatic effectiveness for a leading edge vane geometry and concluded that increased deposition was found where the vortices caused particle impaction on the endwall. Cassadaying et al. (2012) also demonstrated this phenomenon with numerical simulations of ash deposition on adiabatic endwalls. The simulations showed locally increased deposition where the flow of the vortex impacts the endwall, as well as locally decreased deposition where the flow of the vortex pulls away from the endwall.

2.6 Summary and Uniqueness

Although the previous sections discuss conjugate heat transfer studies for flat plates and airfoil geometries, conjugate heat transfer for an endwall has not been studied in the literature. As discussed in Section 2.2, significant skew in the endwall boundary layer and the development of secondary flows in the passage generate unique heat transfer boundary conditions on the external side of the endwall. The current study provides a comprehensive review of conjugate heat transfer with regard to multiple effects that have not previously been investigated for the unique case of an endwall geometry.

Although film cooling and internal impingement cooling are widely used in turbine cooling applications, the effects impingement cooling on overall cooling effectiveness have not been extensively studied. The relative contribution of different cooling schemes to the overall cooling effectiveness, including impingement cooling and film cooling, are quantified for the endwall case using conjugate experiments. Additionally, there is limited information on the effect of geometric parameters on impingement overall effectiveness. Most previous studies of impingement heat transfer have not used conjugate boundary conditions, with the exception of Ravelli et al (2010), who performed numerical simulations for a conducting leading edge geometry. Their study found that the impingement overall effectiveness was not greatly affected by a limited changed in impingement parameters. However, no previous studies have investigated the geometric dependence of impingement heat transfer or overall effectiveness in an endwall geometry. The current study investigates the effects on the performance of a conducting endwall, for a wide range of impingement channel heights. Spatially resolved temperature measurements in the current work also provide a useful benchmark for improved
conjugate computational simulations. Previous comparisons of experimental measurements and computational predictions of overall cooling effectiveness have primarily focused on film cooling effects. It follows that the temperature measurements and corresponding predictions presented, which focus on impingement effects, provide valuable contributions to the turbine cooling literature.

Because endwall contouring is a promising technique to reduce the strength and development of three-dimensional secondary flows in a vane or blade passage, its effects on heat transfer are of critical importance. A limited number of studies have investigated the effects of contouring on heat transfer coefficient or adiabatic effectiveness, but the current study is the first to investigate the heat transfer performance of endwall contouring with a conducting wall boundary condition through measurements and predictions of overall effectiveness. Time resolved flowfield measurements are used to verify the effect of contouring on the passage vortex, and reveal interactions between the passage vortex and film cooling. Previously flowfield measurements of the passage vortex in a linear cascade have not included film cooling within the passage. The trailing edge flow measurements in this study demonstrate that passage film cooling affects the strength and development of the passage vortex downstream.

The use of a conjugate boundary condition with a thermally conducting endwall allows investigation of the thermal effects of coatings, including thermal barrier coatings and contaminant deposition. Although it is known that TBCs protect airfoil surfaces, quantifying the extent to which TBCs can reduce wall temperatures and reduce external heating requires consideration of conjugate heat transfer and the appropriate scaling. The thermal effect of TBCs was studied for a fully cooled conjugate vane geometry (Davidson et al., 2014b), but the trends with TBC are verified for the endwall geometry in the current study. The current work also compares different approaches to improving cooling performance: adding a TBC versus increasing coolant blowing ratio. In the case of molten particle deposition, this process has not previously been studied on an endwall surface with appropriately scaled wall temperature conditions. The current study includes key internal cooling effects from both impingement and in-hole convection from film cooling. Additionally, the effect of deposition on wall overall cooling performance without TBC was not previously studied.

In summary, the current study fills a current gap in the literature for endwall conjugate heat transfer to understand the relative influence of impingement and film cooling, the accuracy
of computational predictions, and the effects of impingement geometry, endwall contouring, a TBC, and contaminant deposition.
CHAPTER 3. OVERALL EFFECTIVENESS OF A BLADE ENDWALL WITH JET IMPINGEMENT AND FILM COOLING*

Abstract

Ever-increasing thermal loads on gas turbine components require improved cooling schemes to extend component life. Engine designers often rely on multiple thermal protection techniques, including internal cooling and external film cooling. A conjugate heat transfer model for the endwall of a seven-blade cascade was developed to examine the impact of both convective cooling and solid conduction through the endwall. Appropriate parameters were scaled to ensure engine-relevant temperatures were reported. External film cooling and internal jet impingement cooling were tested separately and together for their combined effects. Experiments with only film cooling showed high effectiveness around film-cooling holes due to convective cooling within the holes. Internal impingement cooling provided more uniform effectiveness than film cooling, and impingement effectiveness improved markedly with increasing blowing ratio. Combining internal impingement and external film cooling produced overall effectiveness values as high as 0.4. A simplified, one-dimensional heat transfer analysis was used to develop a prediction of the combined overall effectiveness using results from impingement only and film cooling only cases. The analysis resulted in relatively good predictions, which served to reinforce the consistency of the experimental data.

3.1 Introduction

Advances in gas turbine technology for both power generation and aircraft propulsion are focused on increasing efficiency as well as maintaining or extending part service life. Even with current designs, gas temperatures at the inlet to the turbine often exceed allowable material temperatures, and internal cooling and external film cooling are used extensively to cool turbine walls. Thermal degradation of turbine parts has the potential to cause major engine problems, giving rise to costly repairs and downtime. Improved gas turbine efficiency will lead to even

higher thermal loads on the turbine. The ability to evaluate cooling schemes accurately with regard to meaningful performance metrics is crucial.

Ultimately the parameter of interest is the predicted metal temperature of the airfoils and associated surfaces. The non-dimensional scaled metal temperature commonly used is the overall effectiveness, $\phi$. Many studies report a distinctly different quantity known as the adiabatic effectiveness, $\eta$, representing the scaled adiabatic wall temperature, which does not account for internal cooling or heat conduction. Typically, $\phi$ is determined through numerical simulation of the solid conduction using imposed boundary conditions of the parameters: $\eta$, and the internal and external heat transfer coefficients ($h_i$ and $h_\infty$). Alternatively $\phi$ can be measured directly provided the proper experimental scaling, and the metal temperature can be predicted directly.

As will be discussed in the following sections, more recent studies have reported conjugate heat transfer effects and provided the scaled metal temperature. None of these papers, however, have reported conjugate heat transfer results for an endwall surface. Turbine endwall heat transfer is known to be greatly influenced by secondary flow effects, thus measurements of conjugate heat transfer for a film-cooled endwall with internal impingement cooling are of significant value.

3.2 Relevant Literature

Conjugate heat transfer models to determine the overall effectiveness ($\phi$) have been applied to various geometries including flat plates, leading edge models, and vane models. The conjugate effects of conduction and film cooling for a flat plate were examined in a computational study by Bohn et al. (2003). Bohn et al. compared adiabatic and overall effectiveness for a flat plate with one row of $30^\circ$ film cooling holes. While $\eta$ varied laterally across the plate, $\phi$ was relatively uniform. In a similar study, Na et al. (2007) reported that laterally averaged $\phi$ was significantly higher than laterally averaged $\eta$.

Sweeney and Rhodes (2000) experimentally simulated a conjugate surface by matching the engine Biot number, $Bi$, for a configuration including internal impingement jets and film cooling. Between the impingement plate and the wall were heat transfer enhancement features in a Lamilloy® snowflake design. Their results showed that impingement cooling dominated over film cooling in the influence on overall effectiveness. The overall effectiveness distribution was
smooth when impingement jets were tightly spaced, but non-uniformities were observed when the jet spacing became large. A recent study by Panda and Prasad (2012) involved experimental and computational measurements of overall effectiveness for a film cooled plate with and without internal impingement. The authors considered conduction through the wall, but did not report what parameter of their model was matched to an engine. Their results showed that the increase in overall effectiveness from adding internal impingement became more substantial as blowing ratio increased.

Leading edge and vane studies involving both film cooling and internal impingement cooling also provide insight into conducting wall configurations. A series of experimental and computational studies incorporating impingement cooling and film cooling were performed by the same research group on a leading edge model (Albert et al., 2004; Dobrowolski et al., 2009; Maikell et al., 2011; Mouzon et al., 2005; Ravelli et al., 2010; Terrell et al., 2005). The leading edge study by Albert et al. (2004) showed that it was essential to match $\text{Bi}$ as well as the ratio of internal-to-external heat transfer coefficients to accurately measure overall effectiveness. Maikell et al. (2011) measured $\phi$ in the stagnation region with impingement and round film cooling holes. They observed continuous streaks of high $\phi$ around the rows of film cooling holes as blowing ratio increased. Dobrowolski et al. (2009) performed conjugate simulations corresponding to the Maikell et al. experiments. The effect of impingement was applied by setting the internal wall temperature distribution to the temperature measured in the experiments. The same leading edge model with shaped film cooling holes and impingement was considered by Mouzon et al. (2005) and Ravelli et al. (2010). Their results showed that the presence of impingement (H/D~5) did not have much effect on $\phi$ because in-hole convection dominated heat transfer in the stagnation region. Terrell et al. (2005) examined the in-hole convection in more detail. Their experimental and computational study measured the total heat transfer to the coolant as it passed through shaped holes in the stagnation region. The heating of the coolant was found to be significant and needed to be accounted for in their analyses.

Experimental work of a conducting vane was pioneered by Hylton et al. (1983, 1988) and Turner et al. (1985) using a C3X vane. Although engine relevance was limited because matched $\text{Bi}$ and $h_\text{i}/h_\text{t}$ were not confirmed, these studies improved understanding of the thermal fields of a conducting vane, and provided experimental data for benchmarking computational work. In a series of studies (Albert & Bogard, 2013b; Dees et al., 2013; Ledezma et al., 2011; Nathan et al.,
2013; Williams et al., 2014), multiple authors examined different areas of a C3X vane constructed to match engine Bi. Nathan et al. (2013) measured $\phi$ on the stagnation region of the vane, which was cooled internally with impingement and externally with a showerhead arrangement of cooling holes. On the vane pressure side, Albert and Bogard (2013b) found that $\phi$ decreased with increasing blowing ratio due to jet detachment. Dees et al. (2013) and Ledezma et al. (2011) examined $\phi$ on the vane suction side experimentally and computationally, respectively. Internal cooling was provided by a u-bend passage inside the vane that fed film cooling holes. Impingement cooling was added to the suction side of the vane in Williams et al. (2014). In addition to measuring $\phi$, the authors tried to analytically predict $\phi$. They measured $\eta$ on an adiabatic vane and the overall effectiveness with impingement only by blocking some film cooling holes in the suction side row. The un-blocked film cooling holes provided a path for the impingement jets to exhaust. The prediction performed reasonably well for a range of momentum flux ratios.

Although much attention has been paid to conjugate simulations for internally and externally cooled vanes, no studies have been done which consider the endwall surface. Part of the uniqueness of endwall heat transfer is the presence of secondary passage flows based on endwall flow models such as that of Langston et al. (1980). The passage vortices that develop along the endwall skew the direction of endwall flow and locally increase the external heat transfer coefficient. What distinguishes this work from previous studies is the presentation of experimental data for a conjugate endwall with properly scaled conditions as discussed in the next section.

### 3.3 Conjugate Endwall Surface

To achieve relevant experimental data for overall effectiveness, a conjugate surface with appropriate non-dimensionless parameters must be matched between the engine and the experimental model. Using such a model provides a scaled wall temperature measurement that best represents what would occur in the engine, considering all of the convection and solid conduction heat transfer. The experimental endwall model in this study incorporates external film cooling, wall conduction, and internal impingement jet cooling, thereby including the conjugate effects.
The non-dimensional parameters influencing the heat transfer are given in Equation (3.1), which is derived from a one-dimensional consideration of the heat transfer from the external driving temperature, expressed as $T_{aw}$, to the internal coolant temperature at the plenum supplying the impingement cooling, $T_{c,\text{in}}$ (Williams et al., 2014).

$$\phi = \frac{T_\infty - T_w}{T_\infty - T_{c,\text{in}}} = \frac{1 - \chi_\eta \eta}{1 + \text{Bi} + h_\infty/h_i} + \chi_\eta \eta \tag{3.1}$$

Equation (3.1) demonstrates the importance of matching Bi and the ratio of external to internal heat transfer coefficients. A correction factor for $\eta$, $\chi_\eta$, is defined in Equation (3.2) to ensure that the normalizing temperature difference matches that of $\phi$.

$$\chi_\eta = \frac{T_\infty - T_{c,\text{exit}}}{T_\infty - T_{c,\text{in}}} \tag{3.2}$$

The $\chi_\eta$ parameter represents the internal heating of the coolant from the supply plenum, at $T_{c,\text{in}}$, through impingement and the film cooling hole, exiting at $T_{c,\text{exit}}$.

The non-dimensional parameters matched for the endwall in this study relative to those of a typical engine are given in Table 3.1. The external heat transfer coefficient, $h_\infty$, is enhanced due to passage secondary flows, and has been previously reported for this cascade by Lynch et al. (2011b). Although Lynch et al. measured $h_\infty$ without film cooling, film cooling augmentation on the endwall is assumed to be minor (Dees et al., 2010; Eriksen & Goldstein, 1974). The internal heat transfer coefficient, $h_i$, is enhanced with impingement jet cooling. To estimate the average $h_i$, Nusselt number correlations in the literature for impingement cooling with and without crossflow (Florschuetz et al., 1981; Hollworth & Dagan, 1980) were applied. The range of $h_\infty/h_i$ values had some variation with blowing ratio, but stayed reasonably close to 1.0. A reasonable Bi range was achieved by scaling the endwall thickness and using Corian®, a DuPont material, for the endwall.
Table 3.1 Conjugate Endwall Parameters

<table>
<thead>
<tr>
<th></th>
<th>Model</th>
<th>Engine</th>
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</thead>
<tbody>
<tr>
<td>$M_{\text{avg}}$</td>
<td>0.6, 1.0, 2.0</td>
<td>1.0–2.0</td>
</tr>
<tr>
<td>$k_w$, W/m-K</td>
<td>0.99–1.06</td>
<td>22</td>
</tr>
<tr>
<td>$t_w$, cm</td>
<td>1.27</td>
<td>0.20</td>
</tr>
<tr>
<td>$\text{Bi} = h_\infty t_w/k_w$</td>
<td>0.3–0.7</td>
<td>(Lynch et al., 2011b)</td>
</tr>
<tr>
<td>$h_\infty/h_i$</td>
<td>0.4–2.5</td>
<td>(Florschuetz et al., 1981; Hollworth &amp; Dagan, 1980; Lynch et al., 2011b)</td>
</tr>
</tbody>
</table>

3.4 Experimental Methods

Measurements of endwall overall effectiveness were obtained for a linear Pack-B blade cascade using the large scale, low speed, closed loop wind tunnel depicted in Figure 3.1. In this facility, the flow was split into three flow paths. The top and bottom portions, with dark walls in Figure 3.1a, passed through chilled water heat exchangers, while the middle portion passed through a heater bank supplying the mainstream flow to the test section. More details regarding the wind tunnel and flow conditioning elements can be found in (Lynch et al., 2011b).

Mainstream temperatures were measured by a 5-thermocouple rake inserted 0.52C ax upstream of the blade leading edge at multiple locations across the cascade. The mainstream temperatures measured were averaged to find $T_\infty$. The maximum variation from the average $T_\infty$ for any location was ± 0.6°C.

The secondary flow for the endwall coolant supply was removed from the top white channel of the wind tunnel and further cooled by an auxiliary heat exchanger as shown in Figure 3.1b. The auxiliary heat exchanger circulated a sub 0°C glycol-water mixture from the auxiliary chiller. To prevent any ice formation on the heat exchanger fins, a desiccant drier was installed upstream of the heat exchanger. A laminar flow element, LFE, measured the total coolant flowrate, before the lines split the coolant feed for three separate plenums, which are described in detail later. The configuration shown in Figure 3.1 provided a mainstream to coolant temperature difference of about 40°C, resulting in a density ratio, DR, of 1.15. Coolant temperature was measured by two thermocouples ~8.7D below the impingement plate, and ~8.7D below the endwall when there was no impingement plate. The two thermocouples agreed to within ± 3°C or less.
Figure 3.1 Depiction of the (a) large-scale low-speed wind tunnel, with a corner test section housing the Pack-B cascade, and (b) the coolant loop with auxiliary cooling capability and the inlet flow development section.

The top view of the test section, shown in Figure 3.2, contained a seven blade, six passage linear cascade based on the low-pressure turbine Pack-B airfoil, a generic geometry which has been used extensively in the literature (Knezevici et al., 2010; Lake et al., 1999; Lawson et al., 2013; Lynch et al., 2011a, 2011b; Mahallati et al., 2007; Murawski & Vafai, 2000; Popovic et al., 2006; Praisner et al., 2007, 2008; Zoric et al., 2007). The operating conditions as listed in Table 3.2 include engine matched Re and geometric specifications. The inlet mainstream velocity, $U_\infty$, was measured by inserting a pitot probe $0.52C_{ax}$ upstream of each blade leading edge. There was very little variation in $U_\infty$ across the cascade, as the standard deviation over the mean was less than 1%. The test section inlet boundary layer parameters, also listed in Table 3.2, were measured previously by Lynch et al. (2011b). Based on the upstream boundary layer measurements, at the blade inlet plane $\delta/S$ was 0.071 and the freestream turbulence was 4% (Lynch et al., 2011b).
Figure 3.2 Schematic of the Pack-B linear blade cascade with blade and passage numbering and top view of the conjugate endwall.

Table 3.2 Flow Conditions and Blade Geometry

<table>
<thead>
<tr>
<th>Scale factor</th>
<th>8.6</th>
<th>Boundary Layer Parameters 2.85C_a</th>
</tr>
</thead>
<tbody>
<tr>
<td>C_a</td>
<td>0.218 m</td>
<td>Upstream of Blade 4 (Lynch et al., 2011b)</td>
</tr>
<tr>
<td>p/C_a</td>
<td>0.826</td>
<td>( \delta/S )</td>
</tr>
<tr>
<td>S/C_a</td>
<td>2.50</td>
<td>( \delta^*/S )</td>
</tr>
<tr>
<td>Inlet Re</td>
<td>( 1.22 \times 10^5 )</td>
<td>( \theta/S )</td>
</tr>
<tr>
<td>Exit Re</td>
<td>( 1.98 \times 10^5 )</td>
<td>Boundary layer shape factor 1.34</td>
</tr>
<tr>
<td>Inlet U_{\infty}</td>
<td>10.5 m/s</td>
<td>( u'/U )</td>
</tr>
<tr>
<td>Inlet, exit flow angles</td>
<td>35°, 60°</td>
<td>0.061</td>
</tr>
<tr>
<td>Inlet, exit Ma</td>
<td>0.029, 0.047</td>
<td>0.0062</td>
</tr>
</tbody>
</table>

To ensure uniformity and periodicity of the cascade, static pressure taps in the blade midspan were used to measure the pressure distribution before all experiments. A typical set of pressure coefficient, \( C_p \), data is plotted in Figure 3.3 as a function of normalized axial distance.
for all the blades. The measured $C_p$ agreed well with the inviscid CFD prediction (Lynch et al., 2011a), confirming flow uniformity for all passages.

![Figure 3.3 Pack-B cascade static pressure distribution at the blade midspan compared to a CFD prediction (Lynch et al., 2011a).](image)

The endwall in this study was constructed with a geometric configuration and parameters of $Bi$ and $h_e/h_i$ relevant to engine design. The schematic in Figure 3.4a shows the generic internal and external endwall cooling scheme used. Coolant flow is directed into a stagnant plenum passing around a splash plate. The plenum feeds an array of 28 staggered holes in an impingement plate, which feed ten angled film cooling holes in the endwall. The diameter, $D$, is the same for the film and impingement holes. Film cooling holes are inclined at an angle of $30^\circ$ relative to the surface, corresponding to a hole length to diameter ratio, $L/D$, of 5.8. Figure 3.4b shows the locations of the impingement jets and film holes. The film cooling hole inlets are staggered between the impingement jets. The film cooling hole exits are oriented to align with endwall streaklines obtained using oil flow visualization (Lynch et al., 2011b). The area outlined in black is used to calculate the area averaged $\phi$ presented in the results.
Figure 3.4 Schematic of the internal and external cooling scheme from the side view (a) and the top view (b).

To study the effects of the internal and external cooling features separately as well as together, the endwall was divided into three sections of two passages. Each section was fed independently by a separate plenum and incorporated a different cooling arrangement. The center passages, 3 and 4 (see Figure 3.2), contained both internal impingement cooling and external film cooling. Passages 1 and 2 had a static plenum and film cooling and no impingement plate. Passages 5 and 6 had impingement cooling only, which imposed crossflow. Instead of exhausting the coolant through film cooling holes, the coolant flowed out of the channel in the pitchwise (y) direction from a slot above passage 6. The dark area of the endwall in Figure 3.2 represents the endwall constructed of Corian® material, and the light area represents the rest of the endwall constructed of medium density fiberboard.

3.5 Measurement Methods and Uncertainty

The primary measurements in each experiment are the coolant flowrate used to calculate blowing ratio, the coolant and mainstream temperatures, and the endwall temperatures. As mentioned previously, the total coolant flowrate fed to each plenum is measured with an LFE. For experiments using film cooling, the flowrate is adjusted to achieve the desired film cooling
blowing ratio. Blowing ratios reported in this paper reflect the average blowing ratio of all ten film cooling holes, hence the use of $M_{\text{avg}}$. The local blowing ratio for each film cooling hole, $M_{\text{loc}}$, is calculated by considering the static exit pressure of each film cooling hole, measured using pressure taps installed in passage 6. For the cases with only impingement cooling, the mass flow rate of coolant is matched to the total mass flow rate corresponding to the three $M_{\text{avg}}$ values from film cooling. For a 95% confidence interval the uncertainty in coolant flowrate is estimated to be $\pm 3\%$, using the sequential perturbation method described in Moffat (1988).

For film cooled experiments the pressure drop can be used to calculate the average discharge coefficient, $C_d$, of the film cooling holes or the impingement plate plus film holes. Figure 3.5 shows the measured values of $C_d$, plotted in filled symbols, as a function of the average pressure ratio, where $P_{\text{tot,c}}$ is the plenum pressure upstream of the impingement plate, and $P_{\text{loc,\infty}}$ is the local static exit pressure of the hole. The results for film cooling only (filled squares) agree with previous data for 30° inclined holes at low pressure ratios (open squares) (Barringer et al., 2002; Burd & Simon, 1999). The effective $C_d$ for the impingement plate and the film holes (filled circles) is slightly lower than film cooling only, as expected for an additional flow restriction.

Steady state infrared (IR) thermography was used to measure surface temperatures on the endwall. To maximize the spatial measurement resolution and take advantage of the scaled up geometry, a FLIR P20 IR camera was used to measure endwall surface temperatures. The ceiling of the test section contained 16 removable viewing ports, distributed across five blade passages to allow direct optical access for the IR camera. At each viewing location the IR camera was placed perpendicular to the endwall surface at a distance of approximately 56 cm to acquire images. With a camera field of view angle of 25° and a camera resolution of 320 x 240 pixels, the resulting image resolution was 1.3 pixels/mm, which equates to 5.7 pixels/D. Thermocouples were placed in discrete locations on the endwall surface, arranged so that at least two thermocouples were captured in each image. When the thermocouples indicated that steady-state was achieved, five IR images were acquired at each port location. At each location the images were calibrated for emissivity and reflected temperature by minimizing the difference between the thermocouple readings and the image temperatures. The emissivity was typically 0.92 because all endwall surfaces were painted with flat black paint. After calibration, the five images were averaged, exported to an in-house MATLAB program, which assembled the
averaged images from each location. Once a complete endwall temperature map was obtained, the data were reduced to $\phi$.

![Discharge coefficient measured as a function of pressure ratio compared to Burd and Simon (1999) and Barringer et al. (2002).](image_url)

Figure 3.5 Discharge coefficient measured as a function of pressure ratio compared to Burd and Simon (1999) and Barringer et al. (2002).

The partial derivative method (Moffat, 1988) was used to determine the uncertainty in $\phi$. The largest source of uncertainty comes from the calibration of the IR images which provide $T_w$. The bias error of $T_w$ was estimated to be at most 0.8°C from the difference of the IR image temperatures from the thermocouples. The precision error of $T_w$ was estimated to be 0.3°C from the standard deviation of the five images. Using a confidence interval of 95%, the total uncertainty in measurement of $\phi$ was estimated to be ± 0.02.

3.6 Results and Discussion

Overall effectiveness was measured for the three cooling arrangements at three different blowing ratios: $M_{avg} = 0.6$, 1.0, and 2.0 as listed in Table 3.3. The momentum flux ratio, $I_{avg}$, and the associated $h_x/h_i$ are also shown. As mentioned previously, $h_x$ was measured and reported by Lynch et al. (2011b), and $h_i$ was estimated from correlations (Florschuetz et al., 1981; Hollworth
In the second column of Table 3.3, the non-dimensional temperature that was measured for each cooling arrangement is defined. The definitions in Table 3.3 are important to note for the analyses that will be described later in the paper. The differences in overall effectiveness occur in the notation of the coolant temperature: \( T_{c,\text{inlet}} \) for film cooling only, versus \( T_{c,\text{in}} \) for the cases with impingement cooling. The locations of these temperatures are shown in Figure 3.6. \( T_{c,\text{in}} \) is the coolant temperature upstream of the impingement plate. \( T_{c,\text{inlet}} \) is coolant temperature at the inlet of the film cooling hole. For the case with no impingement plate \( T_{c,\text{inlet}} \) is approximately equal to the stagnant coolant temperature.

<table>
<thead>
<tr>
<th>Cooling Arrangement</th>
<th>Measurement</th>
<th>( M_{\text{avg}} )</th>
<th>( I_{\text{avg}} )</th>
<th>( h_\infty/h_i )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Film cooling</td>
<td>( \phi_f = \frac{T_\infty - T_{w,f}}{T_\infty - T_{c,\text{inlet}}} )</td>
<td>2.0</td>
<td>3.5</td>
<td>N/A</td>
</tr>
<tr>
<td>Film cooling</td>
<td>( \phi_f = \frac{T_\infty - T_{w,f}}{T_\infty - T_{c,\text{inlet}}} )</td>
<td>2.0</td>
<td>3.5</td>
<td>0.4-1.0</td>
</tr>
<tr>
<td>Impingement</td>
<td>( \phi_f = \frac{T_\infty - T_{w,o}}{T_\infty - T_{c,\text{in}} \text{ in}} )</td>
<td>1.0</td>
<td>0.9</td>
<td>0.6-1.4</td>
</tr>
<tr>
<td>Impingement</td>
<td>( \phi_f = \frac{T_\infty - T_{w,o}}{T_\infty - T_{c,\text{in}} \text{ in}} )</td>
<td>1.0</td>
<td>0.9</td>
<td>0.6-1.4</td>
</tr>
<tr>
<td>Film &amp; Impingement</td>
<td>( \phi = \frac{T_\infty - T_{w}}{T_\infty - T_{c,\text{in}} \text{ in}} )</td>
<td>1.0</td>
<td>0.9</td>
<td>0.7-1.6</td>
</tr>
<tr>
<td>Film &amp; Impingement</td>
<td>( \phi = \frac{T_\infty - T_{w}}{T_\infty - T_{c,\text{in}} \text{ in}} )</td>
<td>0.6</td>
<td>0.3</td>
<td>1.1-2.5</td>
</tr>
</tbody>
</table>

Figure 3.6 Coolant and wall temperatures of the conjugate with film and impingement cooling.
3.6.1 Film Cooling Only Results

The overall effectiveness contours for film cooling only, $\phi_f$, are presented in Figure 3.7a-c. The film cooling holes and plenum boundaries are shown for reference. Although there is near stagnant air below the endwall, conduction caused $\phi_f$ to be above zero along the pressure side of the passage. In-hole conduction effects are apparent from the increased $\phi_f$ upstream of the film cooling hole exits. Also, there is evidence of cross passage flow sweeping film coolant to the suction side of the passage in all cases. From the contours, the film cooling jets appear lifted off the surface for $M_{avg} = 1.0$ and 2.0, while most of the jets remain attached to the endwall for $M_{avg} = 0.6$. Laterally averaged $\phi_f$ for film cooling only is plotted as a function of axial distance in Figure 3.7d. The $\bar{\phi}_f$ is almost the same for $M_{avg} = 0.6$ and 1.0. There is a peak around $x/C_{ax} = 0.15$, coinciding with the first row of film cooling holes, and a dropoff after $x/C_{ax} > 0.55$ near the last film cooling hole. Local increases are apparent for $M_{avg} = 2.0$ at the first row of holes, as well as in the downstream parts of the passage, after $x/C_{ax} > 0.3$.

$$\phi_f = \frac{T_{\infty} - T_{w,f}}{T_{\infty} - T_{c,inlet}}$$

Figure 3.7 Contours of $\phi_f$ for blowing ratios: (a) $M_{avg} = 0.6$, (b) $M_{avg} = 1.0$, (c) $M_{avg} = 2.0$, with $30^\circ$ inclined holes and plenum boundaries overlaid, and (d) pitchwise laterally averaged $\phi_f$ plotted as a function of axial distance.
3.6.2 Impingement Cooling Only Results

For impingement cooling only the $\phi_o$ contours with impingement hole locations are shown in Figure 3.8a–c. Figure 3.8a specifies direction of coolant exiting the impingement channel under the endwall for the case of impingement only. Compared to film cooling at the same flow rate, the peak values of $\phi_o$ are lower than $\phi_f$, but the distribution of $\phi_o$ is more uniform than $\phi_f$. The influence of the external passage flow can be observed in higher effectiveness near the blade suction side compared to the blade pressure side. As hot external flow travels across the middle of the passage, the cooler endwall reduces the temperature of the flow. The cooler passage flow carries less heat to the downstream suction side of the passage resulting in cooler wall temperatures. Laterally averaged $\phi_o$ is plotted as a function of axial distance in Figure 3.8d. The highest values of $\phi_o$ occur in the middle part of the passage and decrease after $x/C_{ax} > 0.5$, near the last row of impingement jets. In contrast to $\bar{\phi}_f$, $\bar{\phi}_o$ increases with blowing ratio due to the corresponding increase in impingement jet flow rate and internal heat transfer. Average impingement Nusselt numbers estimated from literature correlations (Florschuetz et al., 1981) are listed in the legend in Figure 3.8d. A similar trend for impingement cooling only was noted for the vane in Williams et al. (2014).

Figure 3.8 Contours of $\phi_o$ for blowing ratios: (a) $M_{avg} = 0.6$, (b) $M_{avg} = 1.0$, (c) $M_{avg} = 2.0$, with 90° impingement holes and plenum boundaries overlaid, and (d) pitchwise laterally averaged $\phi_o$ plotted as a function of axial distance.
3.6.3 Combined Film and Impingement Cooling

Overall effectiveness for internal impingement and film cooling, $\phi$, was measured in passages 3 and 4 in the same experiment. The contours are presented in Figure 3.9a-c with the impingement and film cooling hole locations. Boundaries just below the blades are shown, which prevented coolant from crossing from one passage to another in the channel above the impingement plate. Although there is high $\phi$ throughout much of the passage, $\phi$ varies significantly and is not uniform. The primary effect of film cooling, conduction within the film cooling holes, results in high $\phi$ around film holes. The most noticeable blowing ratio effect is at $M_{\text{avg}} = 2$ with increased cooling particularly from convection in the holes.

Figure 3.9d shows laterally averaged measured $\phi$ from passage 4 plotted as a function of axial distance. Similar to film cooling only, there is a peak at $x/C_{ax} = 0.15$ around the first row of film holes, and a decrease in $\phi$ after $x/C_{ax} > 0.55$. The influence of impingement can be observed in the increase of $\bar{\phi}$ with blowing ratio. Figure 3.9 also shows a repeat experiment performed for film and impingement at $M_{\text{avg}} = 1.0$. Good agreement between the two experiments for $M_{\text{avg}} = 1.0$ as well as both passages in Figure 3.9a-c supports the repeatability and reproducibility of the results.

3.6.4 Comparison of Individual and Combined Cooling Effects

The data presented in Figure 3.7, Figure 3.8, and Figure 3.9 are summarized in Figure 3.10 in which area averaged, using the area in Figure 3.4b, overall effectiveness is presented as a function of $M_{\text{avg}}$. Area averaged $\phi$ for film and impingement increases from about 0.3 to 0.4 with an increase in $M_{\text{avg}}$ from 0.6 to 2.0. Consistent with the laterally averaged data, an increase in blowing ratio improves area averaged $\bar{\phi}_o$ more than $\bar{\phi}_f$. Figure 3.10 also highlights the improved cooling that results from adding impingement. From this perspective, the increase of $\bar{\phi}$ from $\bar{\phi}_f$ is larger for higher blowing ratios. In other words $\bar{\phi}$ increases faster than $\bar{\phi}_f$. This trend was also observed for a flat plate in Panda and Prasad (2012). Alternatively, the effect of adding film cooling to an impingement cooled plate indicates a smaller benefit, especially for $M_{\text{avg}} = 2.0$. 

39
Figure 3.9 Contours of $\phi$ for: (a) $M_{avg} = 0.6$, measured (b) $M_{avg} = 1.0$, measured (c) $M_{avg} = 1.0$, predicted, (d) $M_{avg} = 2.0$, measured, and (e) $M_{avg} = 2.0$, predicted, with 30° inclined film holes, 90° impingement holes, and plenum boundaries overlaid.

Figure 3.10 Area averaged $\phi$ (using area outlined in Figure 3.4b) plotted as a function of blowing ratio for all three cooling configurations.
The laterally averaged effectiveness at $M_{avg} = 1.0$ for film cooling only, impingement only, and film and impingement are plotted together in Figure 3.11. Laterally averaged $\phi$ for the combined cooling scheme is higher than $\bar{\phi}_f$ or $\bar{\phi}_o$ from $x/C_{ax} = 0.05$, around the inlets of the film cooling holes, to $x/C_{ax} = 0.35$, just before the last row of impingement holes. However, $\bar{\phi}_o$ is not much lower than $\bar{\phi}$ indicating that the effectiveness is dominated by impingement cooling. Upstream and downstream, $x/C_{ax} < 0.05$ and $> 0.45$, $\bar{\phi}_o$ is higher than $\bar{\phi}$. This unexpected result for impingement only occurs because, after impingement, coolant flows away from the middle of the passage and convectively cools the internal wall in the region $x/C_{ax} < 0.05$ and $> 0.45$. With combined film and impingement, the coolant is instead directed into the film cooling holes.

Figure 3.11 Pitchwise laterally averaged $\phi$ plotted as a function of axial distance for the three cooling configurations at $M_{avg} = 1.0$.

Just downstream of the first row of film cooling holes, at axial distance $x/C_{ax} = 0.22$, $\phi$ is plotted across the passage as a function of normalized distance, $y/p$, in Figure 3.12a-c. Peaks and valleys due to the film cooling jets can be seen in the cases with film cooling. For impingement cooling $\phi_o$ has a relatively flat distribution from $y/p = 0.2$ to 0.55, corresponding to the impingement jet locations. For film cooling there is a peak in $\phi_f$ and $\phi$ around $y/p = 0.25$, corresponding to one of the film cooling holes in the diagonal row. The drop after this peak, especially for $M_{avg} = 2.0$, is attributed to detachment of the film cooling jets from the first row of
holes. Figure 3.12a-c also shows that $\phi_o$ is closest to $\phi_f$ at $M_{avg} = 0.6$, but $\phi_o$ becomes closest to $\phi$ at $M_{avg} = 2.0$.

![Figure 3.12 Overall effectiveness of all cooling configurations plotted as a function of $y/p$ at $x/C_{ax} = 0.22$ for (a) $M_{avg} = 0.6$, (b) $M_{avg} = 1.0$, and (c) $M_{avg} = 2.0$.](image)

3.6.5 One-Dimensional Calculation of Combined Effectiveness Based on Individual Cooling Features

Noting the apparent influence of film cooling and impingement on the combined overall effectiveness, an analysis was performed using the measured values of $\phi_f$ and $\phi_o$ to calculate the combined film cooling and impingement effectiveness, $\phi$. Comparing the results of such an analysis to measured $\phi$ confirms the consistency of the data acquired in the different sections of the cascade. Applying a one-dimensional analysis and reasonable assumptions to the heat transfer of the combined film and impingement cooled conducting wall, an equation can be written for $\phi$ in terms of $\phi_f$ and $\phi_o$. First, using the notation presented in Figure 3.6, the following expression for the heat flux, $q$, can be written.

$$ q = \frac{T_{film} - T_{w,i}}{\frac{1}{h_\infty} + \frac{t}{k_w}} - h_\infty (T_{film} - T_w) $$

where $T_{film}$ is used to represent the effective driving temperature for external convection, and $T_{w,i}$ is the internal wall surface temperature. Equation (3.3) can be used to solve for the wall temperature, $T_w$, and the remaining parameters are either known ($h_\infty$ and $k_w$) or can be estimated from measurements of $\phi_f$ and $\phi_o$ ($T_{film}$ and $T_{w,i}$). Estimating $T_{film}$ from the results of $\phi_f$ assumes that the external driving temperature is not strongly affected by the presence of impingement.
Likewise, estimating $T_{w,i}$ from the results of $\phi_o$ assumes that the internal wall temperature is not strongly affected by the presence of film cooling. This second assumption is supported by the observation that even the external wall temperature, expressed as $\phi$, appeared to be more influenced by impingement than by film cooling because $\phi_o$ was almost as high as $\phi$.

Gathering the non-dimensional $T_{\text{film}}$ from $\phi_f$ requires another one-dimensional equation for the heat transfer through a film cooled conducting wall. Figure 3.13a shows that with film cooling only, the internal cooling is approximated as strictly conduction within the coolant air from the location of the thermocouple at ~8.7D below the endwall. The plenum is nearly stagnant and natural convection is not expected due to the stable coolant temperature distribution. Since the conductivity of air is very low, the heat gained by the coolant due to the internal heat transfer is neglected, and the thermocouple measurement is estimated as $\sim T_{\text{c,inlet}}$.

Using Figure 3.13a, the heat flux through the film cooled endwall, $q_f$, is defined in Equation (3.4).

$$q_f = \frac{T_{\text{film}} - T_{\text{c,inlet}}}{h_\infty + \frac{t}{k_w} + \frac{L_{c,in}}{k_{c,in}}} = h_\infty \left( T_{\text{film}} - T_{w,f} \right)$$

From Equation (3.4), an expression for the non-dimensional $T_{\text{film}}$ can be found in terms of measured quantities, including $T_{w,f}$, expressed as $\phi_f$.

An expression for the non-dimensional $T_{w,i}$ is found in a similar manner considering the one-dimensional heat transfer through an impingement cooled wall, shown in Figure 3.13b. The heat flux through the endwall with impingement cooling can be written as

$$q_o = \frac{k_w}{t} \left( T_{w,o} - T_{w,i} \right) = h_\infty \left( T_{w,o} - T_{w,o} \right),$$

which can be rearranged into a non-dimensional $T_{w,i}$ in terms of measured quantities, including $T_{w,o}$, expressed as $\phi_o$. 

43
Figure 3.13 Coolant and wall temperatures of the conjugate wall with (a) film cooling only, and (b) impingement cooling only.

Using Equations (3.3), (3.4) and (3.5) and the definitions of $\phi_f$ and $\phi_o$, Equation (3.6) can be written for the calculated $\phi$ with film and impingement cooling.

$$\phi_{calc} = \phi_o + \chi_f \left( \frac{Bi}{1+Bi} \right) \left[ \phi_f - \left( \frac{(1-\phi_f)}{h} \left( \frac{L}{k_c} \right) \right) \right]$$

(3.6)

The quantity, $\chi_f$, is the internal coolant warming factor necessary to correct for the difference in normalizing coolant temperatures between $\phi$ and $\phi_f$ (see definitions in Table 3.3). The internal coolant warming factor, $\chi_f$, accounts for the coolant warming through internal impingement, similar to $\chi_h$ accounting for the coolant warming through in-hole convection in (Williams et al., 2014). The coolant temperature increase due to impingement, between the plenum at $T_{c, internal}$ and the inlet to the film cooling hole at $T_{c,inlet}$, can be estimated from a total energy balance on the case of impingement only. Assuming the heat gained by the coolant comes from the endwall only, $T_{c,inlet}$ can be estimated as the temperature of the coolant exiting the impingement channel in Figure 3.13b, leading to Equation (3.7).

$$\chi_f = \frac{T_{\infty} - T_{c,inlet}}{T_{\infty} - T_{c,inlet}} = 1 - \sum \left( \frac{h_{\infty} \phi_o A}{m_c c_p} \right)$$

(3.7)

The values of $\chi_f$ were 0.75 for $M_{avg} = 0.6$, 0.82 for $M_{avg} = 1.0$, and 0.88 for $M_{avg} = 2.0$.

When Equation (3.6) is applied, the laterally averaged $\phi_{calc}$ (across the area outlined in Figure 3.4b) are calculated for the combined film cooling and impingement cooling cases and plotted in Figure 3.14. The calculated values agree very well with those measured at $M_{avg} = 0.6$. Although agreement for all blowing ratios is relatively good in the upstream parts of the passage,
more deviation occurs downstream as blowing ratio increases. The overpredictions downstream are probably an effect of higher measured effectiveness for impingement only than for film and impingement after $x/C_{ax} > 0.45$, seen in Figure 3.11. This was attributed to differences in the impingement flow with and without film cooling present. With impingement only, the impingement jets are exhausted laterally as shown in Figure 3.8a and the resulting channel flow cools the downstream internal wall more than when the coolant is directed into the film cooling holes. Additional sources of deviation arise from the assumptions made in the analyses, including using the same $h_{\infty}$ for cases with and without film cooling, and assuming the coolant air below the film cooling only endwall was stagnant. The literature has shown that external heat transfer coefficient can increase with the addition of film cooling, and this effect increases with blowing ratio (Dees et al., 2010; Eriksen & Goldstein, 1974). An underestimated $h_{\infty}$ would result in a higher $\phi_{\text{calc}}$, consistent with the results of the calculation in Figure 3.14.

![Graph showing laterally averaged $\phi_{\text{calc}}$ and $\phi_{\text{meas}}$ plotted as function of axial distance for all three blowing ratios.](image)

**Figure 3.14** Comparison of laterally averaged $\phi_{\text{calc}}$ and $\phi_{\text{meas}}$ plotted as function of axial distance for all three blowing ratios.

Overall the favorable comparison between the individual cooling features and the overall impingement plus film cooling is well predicted. These results provide insights as to the contribution of each for a conjugate endwall.
3.7 Conclusions

Overall effectiveness data were presented for an endwall with external film cooling and internal impingement jet cooling. These two cooling mechanisms were examined separately and together at three different film cooling blowing ratios. The results indicated that with film cooling alone higher effectiveness values were measured near the film cooling holes due to convection within the holes. Increasing blowing ratio increased the cooling effectiveness near the hole exits, but did not significantly improve effectiveness elsewhere in the passage given the jets separated from the surface. Alternatively, impingement cooling showed clear improvement with increasing coolant flowrates. In addition the effectiveness distribution for impingement cooling was very uniform.

Combining internal and external cooling produced higher overall effectiveness compared to impingement cooling or film cooling alone. The passages cooled with impingement and film cooling showed high effectiveness near the holes, and increasing effectiveness with blowing ratio. Effectiveness results from film cooling alone and impingement cooling alone were used to calculate the effectiveness of the combined cooling scheme. The calculations compared reasonably well to the measured data using a simple one-dimensional heat transfer analysis. The agreement was best for \( M_{\text{avg}} = 0.6 \) and 1.0. As the blowing ratio increased, the increasing influence of internal impingement on overall effectiveness caused more deviations from the analyses due to impingement flow differences.

The current study demonstrates the conjugate methodology and overall trends for a generic impingement and film-cooled endwall, regardless of the specific airfoil geometry. These results demonstrate the importance of testing internal and external cooling schemes together to capture conjugate effects. There were differences in impingement cooling and film cooling behavior when the two were combined. Measurements of conjugate heat transfer on an endwall fill a gap in overall effectiveness data for gas turbine endwalls. This study also provides a valuable data set to benchmark future conjugate simulations.
CHAPTER 4. CONJUGATE HEAT TRANSFER ANALYSIS OF THE EFFECTS OF IMPINGEMENT CHANNEL HEIGHT FOR A TURBINE BLADE ENDWALL†

Abstract

Advancements in cooling for applications such as gas turbines components require improved understanding of the complex heat transfer mechanisms and the interactions between those mechanisms. Critical cooling applications often rely on multiple thermal protection techniques, including internal cooling and external film cooling in gas turbine airfoils, to efficiently cool components and limit the use of coolant. Most research to quantify the effectiveness of such cooling technologies for gas turbine applications has isolated internal and external cooling in separate experiments. The research presented in this paper uses a conjugate heat transfer approach to account for the combined effects of both internal and external cooling. The geometry used for this study is a turbine blade endwall that includes impingement and film cooling as well as the relevant conduction through the endwall. Appropriate geometric and flow parameters were scaled to ensure engine relevant dimensionless temperatures were obtained. Using the conjugate heat transfer approach, the effect of varying the height of the impingement channel was examined using spatially resolved external wall temperatures obtained from both experiments and simulations. A one-dimensional heat transfer analysis was used to derive the average internal heat transfer coefficients from the experimental results. Both experiments and simulations showed good agreement between area averaged cooling effectiveness and impingement heat transfer coefficients. The cooling effectiveness and heat transfer coefficients peaked for an impingement channel height of around three impingement hole diameters. However, the heat transfer coefficients were more sensitive than the overall effectiveness to the changes in height of the impingement channel.

4.1 Introduction

A continuing challenge in advanced cooling applications is understanding the interaction between multiple heat transfer mechanisms, which is referred to as conjugate heat transfer. Conjugate heat transfer is the combined result from convective heating and cooling, conduction within the walls, and radiation heat transfer. In many applications such as along gas turbine components, the most effective cooling configurations are often three-dimensional and are surrounded by complicated flow fields and thermal fields. In a gas turbine engine, the airfoil and endwall surfaces simultaneously experience convective heating from the hot combustion gases and convective cooling from air supplied by the compressor that has bypassed the combustor. The convective cooling occurs both internal to the airfoil, such as through jet impingement, and external to the airfoil, such as through small angled holes in the airfoil walls providing what is known as film cooling. In combined impingement and film cooling, the cooling air impinges on the internal walls, and then passes through the film cooling holes to generate a protective film of coolant on the outer wall. The combination of the convective and conductive heat transfer processes determines the resulting wall temperature, which governs the service life of the turbine components. Therefore, accurate predictions of component temperature are critical to evaluate cooling technologies.

Current practices to predict turbine component temperatures involve calculating the solid conduction using analytical or numerical tools while applying convective boundary conditions based on separate internal and external experiments or analyses. Most literature in gas turbine heat transfer reports either heat transfer coefficients measured with a constant heat flux boundary condition or adiabatic film cooling effectiveness measured with an adiabatic boundary condition. The latter is applied in analytical or numerical tools to represent the reference temperature for external convection in the presence of film cooling. An alternative to this isolated heat transfer analysis is direct determination of the non-dimensional wall temperature, referred to as the overall effectiveness ($\phi$), since that is the value of most interest to turbine designers. To determine the non-dimensional temperature, a conjugate experiment or simulation must use a properly scaled conjugate model that couples the convective heat and cooling and solid conduction.

As will be discussed in the following sections, recent experiments and simulations have begun to investigate conjugate heat transfer effects to provide scaled metal temperatures. This
study focus on the conjugate heat transfer results due to variations in the internal impingement cooling geometry, building upon the results for a blade endwall with impingement and film cooling (Mensch et al., 2014; Mensch & Thole, 2014). Conjugate experiments and computational simulations are used to examine the influence of internal impingement cooling geometry on wall temperatures and internal heat transfer coefficients. The convective cooling under the endwall of a turbine airfoil is of interest in this study, specifically the effects of the distance between the impingement plate and the endwall target.

4.2 Relevant Literature

Numerous experiments with constant temperature and constant heat flux boundary conditions are found in the literature for turbine airfoils, but these studies provide only a portion of the required boundary condition information to predict the actual endwall temperature. Internal heat transfer coefficients for engine relevant geometries of internal impingement cooling can be found in the papers by Florschuetz et al. (1981) and Hollworth and Dagan (1980). These two studies provide correlations for the Nu as functions of jet Re and geometric parameters with a constant temperature boundary condition. Florschuetz et al. (1981) considered staggered impingement jet geometries where the coolant was extracted laterally from one side. The authors found that the crossflow that developed in the channel generally degraded the heat transfer coefficient from the first row of jets to the exit row. Hollworth and Dagan (1983; 1980) measured the Nu for staggered impingement geometries where the coolant is extracted through angled holes in the target plate, which simulates a configuration with combined impingement and film cooling. Hollworth and Dagan (1980) provided a correlation for the area-averaged Nu for configurations with impingement and film cooling extraction. Although some geometric parameters are included in the impingement correlations, the ratio of impingement holes to extraction holes is not included, and this ratio may differ for realistic endwall geometries such as the one presented in our study. The impingement heat transfer effects of certain parameters, such as the distance between the impingement plate and the target, H, were reviewed by Viskanta (1993). The Nu for the impingement jets usually varied with the impingement channel height to hole diameter ratio, H/D, with a maximum occurring between H/D of 1.5 – 4 depending on the specific jet arrangement and method of Nu measurement (Florschuetz et al., 1981; Hollworth & Dagan, 1980; Viskanta, 1993). For impingement with film cooling extraction,
Hollworth and Dagan (1980) found that for the smallest spacing between impingement jets, 5D, there was not much change in Nu for a wide range of H/D = 0.5 – 6.0.

Conjugate heat transfer models to determine the overall effectiveness, φ, or non-dimensional wall temperature, have been applied to various geometries including flat plates, leading edge models, and full turbine airfoil models. The conjugate effects of conduction and film cooling for a flat plate were examined by Wang and Zhao (2011) with a two-dimensional slot geometry. They compared the results obtained for different wall boundary conditions including adiabatic and conjugate walls. While the adiabatic wall temperatures varied across the surfaces, the scaled conducting wall temperature, φ, was relatively uniform. In the gas turbine industry, the conduction in the metal components is often assumed to dominate the heat transfer, smear temperature gradients, and produce nearly constant φ. However, even for the case of a very low Biot number, Bi ~ 0.03, Wang and Zhao (2011) showed that φ varied locally and was not uniform across the surface. Conjugate heat transfer experiments for turbine applications were pioneered by Hylton et al. (1983, 1988) and Turner et al. (1985). Although the Bi was not identified, these studies improved the understanding of the thermal fields of a conducting vane, and provided experimental data for benchmarking computational work.

Studies including both experimental and computational conjugate heat transfer results provide important comparisons for validation of computational design tools. Papanicolaou et al. (2001) compared computational simulations of conjugate heat transfer to experimental measurements made on a flat plate with film cooling for a Bi ~ 3 and a Bi ~ 0.08. Their effectiveness results highlighted the large differences in temperature distribution that are observed for experiments with different Bi. Panda and Prasad (2012) also compared experimental and computational results for a flat plate with film cooling with and without additional cooling by internal impingement jets. The authors considered conduction through the wall, but did not report the Bi of their model. The simulations showed good agreement with the experimental results along the plate centerline.

Albert et al. (2004) developed a one-dimensional heat transfer equation, which demonstrated that it is essential to match Bi as well as the ratio of external-to-internal heat transfer coefficients, h_∞/h_i, in order to acquire relevant temperature data in a conjugate experiment. The first experiments to use an engine matched Bi experimental model were completed by Sweeney and Rhodes (2000) for a three-dimensional flat plate with internal
impingement and film cooling. Between the impingement plate and the wall were heat transfer enhancement features in a Lamilloy® snowflake design. Their results showed that impingement cooling dominated over film cooling in the distribution of external wall temperatures. The temperature distribution varied with different arrangements of impingement jets. When the jets were tightly spaced, the temperature distribution was quite uniform. When the jet spacing increased, non-uniformities were observed the external wall temperature.

Additional matched Bi experiments and simulations were performed by Maikell et al. (2011) and Dobrowolski et al. (2009) on a leading edge model with internal jet impingement and external film cooling. The effects of jet impingement were applied to the simulations by setting the internal wall temperature distribution equal to the measured temperature distribution from the experiments. The simulated heat flux results were compared to the predicted heat flux, calculated using the adiabatic wall temperature and the $h_{\infty}$ obtained from a constant wall heat flux simulation. Mouzon et al. (2005) and Ravelli et al (2010) tested the same leading edge model but with shaped film cooling holes with and without impingement cooling, and found there was little difference when impingement was added due to the high effectiveness of the film cooling used. Ravelli et al (2010) varied the size and location of the impingement holes relative to the film cooling holes in the numerical portion of the study, but the overall effectiveness was not significantly changed. However, the arrangement of holes affected the internal flow recirculation and distribution of coolant between the film cooling holes. Williams et al. (2014) and Dyson et al. (2012) performed experiments and computations, respectively for the suction side of a vane model having an engine matched Bi of 0.4-1.6 with film cooling and impingement. Williams et al. (2014) also measured the overall effectiveness with impingement cooling only by blocking some film cooling holes in the suction side row. The un-blocked film cooling holes provided a path for the impingement jets to exhaust, which minimized the alteration of the impingement flow path and isolated the film cooling external effects from the film cooling effects on the impingement flow.

The overall thermal performance of a turbine endwall geometry with impingement and film cooling was investigated by Mensch and Thole (2014) and Mensch et al. (2014) with matched Bi experiments and simulations. The external heat transfer for an endwall differs from that of a flat plate, leading edge or airfoil surface, due to the influence of passage secondary flows including the horseshoe and passage vortices. The passage vortices that develop along the
endwall skew the direction of endwall flow and locally increase the external heat transfer coefficient, as measured by Kang and Thole (2000). The distribution of the external endwall heat transfer coefficients for the airfoil geometry in this study was investigated by Lynch et al. (2011a) through both experiments and computational predictions with a constant wall heat flux boundary condition. For the conjugate endwall studies (Mensch et al., 2014; Mensch & Thole, 2014), the endwall was cooled with the same cooling features as the current study. Results indicated that the internal impingement cooling had a greater influence than film cooling on the scaled endwall temperatures.

Since internal impingement cooling is important in turbine endwall heat transfer, the current study seeks to examine the impact on overall effectiveness of varying geometric parameters relative to impingement cooling. Although data are available for the dependence on geometric parameters of heat transfer coefficient, previously available experimental databases do not include data derived from conjugate experiments. For a conducting leading edge, Ravelli et al. (2010) found that the overall effectiveness was not greatly affected by a limited changed in impingement parameters. The current study investigates the effects on the performance of a conducting endwall, for a wide range of impingement channel heights.

4.3 Simulation Methods

4.3.1 Conjugate Methodology

To achieve relevant, scaled wall temperature data, a conducting surface with appropriate non-dimensionless parameters must be matched between the experimental model and the actual turbine airfoil being simulated. This matching ensures that the scaled results best represent the behavior of the turbine surface, considering all of the heat transfer and cooling relevant to the wall. The experimental model in this study incorporates external film cooling, wall conduction, and internal impingement jet cooling, thereby including the conjugate effects. Note that for most turbine blade applications, radiation is not included since the blades do not see the combustor and nearly all the surfaces seen by a blade are at similarly high temperatures.

A simple schematic of the heat transfer model showing the pertinent temperatures and properties is shown in Figure 4.1a for a configuration with impingement and film cooling and Figure 4.1b for impingement cooling only. All non-dimensional temperatures are scaled by the
overall temperature difference between the hot mainstream temperature, \( T_\infty \), and the internal coolant temperature at the plenum supplying the coolant, \( T_{c,in} \). The reference temperature for the external convection coefficient with film cooling is assumed to be a temperature representing the overall film temperature, \( T_{\text{film}} \), (typically the adiabatic wall temperature) which accounts for the mixing of the coolant with the mainstream (Goldstein, 1971). The non-dimensional film temperature is assumed to be independent of the boundary condition at the wall, and only a function of geometry and the coolant and mainstream Reynolds numbers. The temperature of the coolant after impingement, called \( T_{c,inlet} \), is approximately the same in both cases, as verified by thermocouple measurements. The equivalence of these temperatures indicates that the total heat transferred from the internal endwall surface to the coolant is the same regardless of how the coolant exits the channel.

![Diagram of conjugate wall with film and impingement cooling](image)

**Figure 4.1** Important parameters and temperatures of the conjugate wall with (a) film and impingement cooling and (b) impingement cooling only.

To illustrate the non-dimensional parameters influencing the heat transfer, Equation (4.1) is presented for the overall effectiveness, \( \phi \). Equation (4.1) is derived by equating the external convection heat flux to the overall one-dimensional heat transfer from \( T_{\text{film}} \) to \( T_{c,in} \) for the general case of impingement and film cooling shown in Figure 4.1a. When there is no film cooling, such as in Figure 4.1b, \( T_{\text{film}} \) is equal to \( T_\infty \), and the non-dimensional temperature in Equation (4.1) becomes zero. Equation (4.1) is similar to the one presented recently for studies with internal and external cooling, such as Williams et al. (2014), except for the use of \( T_{\text{film}} \) as the driving temperature, rather than the adiabatic wall temperature.
Equation (4.1) demonstrates the importance of matching the geometry and the non-dimensional parameters of Bi, and $h_{\infty}/h_i$ for conjugate studies to be relevant to actual conditions. The ranges of the non-dimensional parameters matched to that of a gas turbine are given in Table 4.1 showing that the model meets the criteria of matching the Bi and $h_{\infty}/h_i$. The external heat transfer coefficient, $h_{\infty}$, is enhanced from that of a flat plate due to passage secondary flows (Lynch et al., 2011b). Although the $h_{\infty}$ measured does not account for film cooling, film cooling augmentation on the endwall is assumed to be minor (Dees et al., 2010; Eriksen & Goldstein, 1974). The internal heat transfer coefficient, $h_i$, is enhanced with impingement jet cooling. To estimate the average $h_i$ in the design of the experiment, Nusselt number, Nu, correlations in the literature for impingement cooling with and without crossflow (Florschuetz et al., 1981; Hollworth & Dagan, 1980) are applied. The range of $h_{\infty}/h_i$ values has some variation with blowing ratio, but stays reasonably close to 1. A reasonable Bi range is achieved by scaling the endwall thickness and thermal conductivity through the use of Corian®, a thermally conductive material, to manufacture the endwall. Flow conditions and geometric scaling of the model were designed to match the non-dimensional film temperature.

**Table 4.1 Conjugate Endwall Parameters**

<table>
<thead>
<tr>
<th>Model</th>
<th>Engine</th>
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<tbody>
<tr>
<td>$M_{avg}$</td>
<td>1.0, 2.0</td>
</tr>
<tr>
<td>$k_w$, W/m-K</td>
<td>0.99–1.06</td>
</tr>
<tr>
<td>$t_w$, cm</td>
<td>1.27</td>
</tr>
<tr>
<td>$Bi = h_{\infty}/k_w$</td>
<td>0.3–0.7</td>
</tr>
<tr>
<td>$h_{w}/h_i$</td>
<td>0.4–2.5</td>
</tr>
</tbody>
</table>

As a check, the internal heat transfer coefficient can be calculated using measured temperatures from a simple one-dimensional analysis with knowledge of three temperatures. Considering the general case shown in Figure 4.1 (a), the relevant temperatures are $T_{film}$, $T_w$, $T_{w,i}$,
and $T_{c,in}$. The mainstream temperature, $T_\infty$, cannot be used directly since it is not the external driving temperature with film cooling as previously described. Since $T_{film}$ is difficult to quantify, the latter three temperatures are used to determine $h_i$. Just as the external wall temperature, $T_w$, is non-dimensionalized as $\phi$, the data for $T_{w,i}$ can be non-dimensionalized by the mainstream and coolant temperatures to define an internal overall effectiveness, as shown in Equation (4.2).

$$\phi_i = \frac{T_\infty - T_{w,i}}{T_\infty - T_{c,in}}$$

(4.2)

To derive an equation for $h_i$ using the coolant and wall temperatures, the internal convection heat flux is set equal to the one-dimensional conduction heat flux through the wall. Equating two adjacent modes of heat transfer limits the impact of three-dimensional effects. This approach generates the following Equation (4.3) for $h_i$ obtained from the experiments. The $h_i$ obtained from the computational results is directly calculated from the local heat flux and temperatures, shown in Equation (4.4).

$$h_{i,\text{mean}} = \frac{k_w (\phi_i - \phi)}{t_w (1 - \phi_i)}$$

(4.3)

$$h_{i,\text{pred}} = \frac{q_{w,i}}{(T_{w,i} - T_{c,in})} = \frac{q_{w,i}}{(T_\infty - T_{c,in}) (1 - \phi_i)}$$

(4.4)

4.3.2 Experimental Methods and Uncertainty

Steady state experiments were performed for the endwall of a linear blade cascade inserted into a large scale, low speed, closed loop wind tunnel depicted in Figure 4.2. The wind tunnel split into mainstream and coolant flow paths upstream of the test section, as shown in Figure 4.2a where the tunnel widens. The coolant flow was diverted into the top blue section. The mainstream flow continued through the center portion of the wind tunnel, which contained a heater bank, turbulence grid, and other flow conditioning elements. More details regarding the wind tunnel can be found in (Lynch et al., 2011b). Mainstream temperatures were measured by five thermocouples on a rake inserted 0.5 blade axial chords upstream of the blade leading edge at multiple locations across the cascade. The measured mainstream temperatures were averaged to determine $T_\infty$. The maximum variation from the average $T_\infty$ for any location was ± 0.6°C. A Pitot probe, also inserted 0.5Cax upstream, was used to measure the inlet mainstream velocity, $U_\infty$. The standard deviation over the mean $U_\infty$ was less than 1%.
Figure 4.2 Depiction of the (a) large-scale low-speed wind tunnel, with a corner test section housing the Pack-B cascade, and (b) the coolant loop with auxiliary cooling capability and the inlet flow development section.

From the top blue channel of the wind tunnel, the coolant was extracted and passed through a drier and a chilled glycol-water heat exchanger as shown in Figure 4.2b. A laminar flow element measured the total coolant flowrate, before the flow entered one of three separate plenums below the endwall, which are described in detail later. The configuration shown in Figure 4.2 provided a mainstream to coolant temperature difference of about 40°C, resulting in a coolant to mainstream density ratio, $DR$, of about 1.15. The coolant temperature, $T_{c,in}$, was measured by two thermocouples below the impingement plate whereby the two thermocouples agreed to within ± 3°C for a typical average $T_{c,in} = 10°C$. When film cooling was included in the endwall, the coolant flowrate was adjusted such that desired film cooling blowing ratio was achieved. Blowing ratios reported in this paper reflected the average blowing ratio over all film cooling holes, $M_{avg}$. The local blowing ratio for each film cooling hole, $M_{loc}$, was calculated by scaling the total coolant flowrate by the contribution of each film cooling hole, determined by the local static exit pressure of each hole. For the cases without film cooling, the mass flow rate of coolant was matched to the total mass flow rate corresponding to the $M_{avg}$ values with film cooling. For a 95% confidence interval, the uncertainty in coolant flowrate was estimated to be ± 3%, using the sequential perturbation method described in Moffat (1988).

The top view of the test section is shown in Figure 4.3, which shows the seven blade, six passage linear cascade. The blade geometry used was a generic airfoil common in the literature (Knezevici et al., 2010; Lake et al., 1999; Lawson et al., 2013; Lynch et al., 2011a, 2011b;
Mahallati et al., 2007; Mensch et al., 2014; Mensch & Thole, 2014; Murawski & Vafai, 2000; Popovic et al., 2006; Praisner et al., 2007, 2008; Zoric et al., 2007). The conducting portion of the endwall is shown in green. Passages 1 and 2 were not used in this study. The center passages, 3 and 4, had film and impingement cooling, so the plenum below these passages contained an impingement plate. Passages 5 and 6 had impingement cooling only, and the coolant was exhausted laterally through a vertical slot below blade 7. The blade geometric parameters are listed in Table 4.2 along with the inlet mainstream flow conditions. The test section inlet boundary layer parameters were measured at 2.85C_{ax} upstream of the center blade by Lynch et al. (2011b). To ensure uniformity and periodicity of the cascade, static pressure taps in the blade midspan were used to measure the pressure distribution before all experiments. The measured static pressure around each airfoil agreed well with the inviscid CFD prediction, confirming flow uniformity for all passages (Mensch et al., 2014).

Figure 4.3 Schematic of the Pack-B linear blade cascade with blade and passage numbering and top view of the conjugate endwall.
Table 4.2 Flow Conditions and Blade Geometry

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<tbody>
<tr>
<td>Scale factor</td>
<td>8.6</td>
<td>Inlet flow angle</td>
<td>35°</td>
</tr>
<tr>
<td>$C_{ax}$</td>
<td>0.218 m</td>
<td>Exit flow angle</td>
<td>60°</td>
</tr>
<tr>
<td>$p/C_{ax}$</td>
<td>0.826</td>
<td>Inlet Ma</td>
<td>0.029</td>
</tr>
<tr>
<td>$S/C_{ax}$</td>
<td>2.50</td>
<td>Exit Ma</td>
<td>0.047</td>
</tr>
<tr>
<td>Inlet Re</td>
<td>$1.22 \times 10^5$</td>
<td>Inlet $\delta/S$</td>
<td>0.061</td>
</tr>
<tr>
<td>Exit Re</td>
<td>$1.98 \times 10^5$</td>
<td>Inlet $u'/U_\infty$</td>
<td>0.06</td>
</tr>
</tbody>
</table>

Generic internal and external endwall cooling schemes were used to achieve the matched parameters of $Bi$ and $h_x/h_i$ and scaled geometry, shown in Figure 4.4. Coolant flow is directed into the plenum by means of a splash plate. The coolant flows through an array of 28 holes in the impingement plate. When film cooling is included, the coolant leaves the impingement channel through ten cylindrical film cooling holes angled at 30° to the endwall surface, corresponding to a hole length-to-diameter ratio, $L/D$, of 5.8. The diameter, $D$, is the same for the film and impingement holes, 4.4 mm. The cross sectional area of the plenum is over 100 times that of the combined cross sectional area of the impingement holes, and over 300 times that of the combined cross sectional area of the film cooling holes. Figure 4.4b shows the passage location of the impingement jets and film holes. The first row of impingement jets is aligned with the leading edge plane, $x = 0$, and staggered thereafter with a spacing of $4.65D$ in both $x$- and $y$-directions. The film cooling hole inlets are located between impingement rows and are separated by the same spacing as the impingement jets. The film cooling holes are oriented in the $x$-$y$ plane to align with endwall streaklines obtained using oil flow visualization (Lynch et al., 2011b). Figure 4.4b also identifies the four locations of surface thermocouples on the internal side of the endwall, with red dots. The blue circles show the locations of the $T_{c,inlet}$ thermocouples, placed in the flow at the entrance to the middle film cooling hole in rows 1 and 2. Finally, the area outlined in black is used to calculate the area averaged $\phi$ presented in the results.
Figure 4.4 Schematic of internal and external cooling scheme from the side view (a) and the top view (b), also showing area average outline and locations of internal thermocouples.

External surface temperatures on the endwall were measured with steady state infrared (IR) thermography to maximize the spatial measurement resolution and take advantage of the scaled up geometry. The IR camera viewed the endwall from a distance of 56 cm through removable ports in the ceiling of the test section. At steady-state, five IR images were acquired at each port location. The resolution of each image was 1.3 pixels/mm or 5.7 pixels/D. The images were calibrated for emissivity and reflected temperature by minimizing the difference between the image and thermocouples embedded at the endwall outer surface. The emissivity was typically 0.92 because all endwall surfaces were painted with flat black paint. After calibration, the images were assembled into a single endwall temperature map. The uncertainty in $\phi$ was determined to be $\pm 0.02$ using a confidence interval of 95% with the partial derivative method (Moffat, 1988). Ribbon-type thermocouples were placed at four locations on the internal endwall surface, to be used in calculating the non-dimensional internal wall temperature, $\phi_i$. The internal wall thermocouple locations are presented in Figure 4.4b with red dots. Using a 95% confidence interval, the uncertainty in $\phi_i$ is estimated to be $\pm 0.01$.

4.3.3 Computational Methodology

Conjugate computational simulations were performed using a commercial computational fluid dynamics (CFD) software (FLUENT 13.0.0, 2010). The steady-state RANS and energy
equations were solved with the segregated pressure-based SIMPLE algorithm and SST k-ω turbulence model (Menter, 1994) for closure. The SST k-ω model has shown reasonable agreement with experimental results in turbomachinery applications (Dyson et al., 2012; Lynch et al., 2011a; Panda & Prasad, 2012; Schwänen & Duggleby, 2009; Snedden et al., 2009). The computational domain is shown in Figure 4.5a. A velocity inlet was applied 3.5 blade axial chords upstream of the blade leading edge. The inlet had a uniform temperature distribution to match the experiments and a mainstream velocity of 10.5 m/s. The inlet boundary layer profile was generated using the boundary layer code TEXSTAN (Crawford, 2009) to match the momentum thickness Reynolds number measured by Lynch et al. (2011b) 2.85 blade axial chords upstream of the blade leading edge in the inlet flow direction. At the exit of the domain, an outflow boundary condition was applied 1.5 blade axial chords downstream of the blade trailing edge in the x-direction. Symmetry was imposed at the top of the domain, which was located at the midspan of the blade in the experiments. Periodic boundaries were imposed on the sides of the domain, which cut through the middle of the passage and internal cooling passages below the endwall. The plenum extended 65D below the impingement plate to reflect the dimensions of the experimental plenum. A mass flow inlet boundary condition was applied at the entrance to the plenum, where the prescribed mass flow rate and temperature matched the experimental conditions for each blowing ratio. The air properties for the flow were determined by incompressible-ideal gas for density and Sutherland’s law for viscosity. Polynomial fits were used to incorporate the temperature dependence of the air thermal conductivity and air specific heat. A thermally-coupled wall interface was used at all conjugate solid/fluid boundaries. All other wall surfaces in the domain were modeled as adiabatic.

Separate unstructured grids were generated for the conducting endwall solid and the flow domain. Commercial grid generation software (Pointwise 17.1r3, 2013) was used for the endwall. The unstructured endwall grid, shown in Figure 4.5b, contained 1.5 million cells. The open source grid generation code, Advancing-Front/Local-Reconnection (ALFR3) (Marcum & Gaither, 1999), was used to generate the flow grids. The flow grids were comprised of mainly tetrahedral cells and layers of wall-normal prism cells on key surfaces (the blade, the entire external endwall surface, the internal endwall surface, the film cooling holes, and the impingement holes). To resolve the boundary layer on these surfaces the first grid point was located at a y⁺ less than one. The unstructured flow grid for the case of H/D = 2.9 is depicted in
Figure 4.5d, which shows a slice in the y-plane through a film cooling hole inlet and an impingement hole, and in Figure 4.5c, which shows a slice in the x-plane through the mainstream, film cooling holes, impingement channel, impingement holes, and plenum.

Figure 4.5 Depiction for the case of H/D = 2.9 of (a) the computational domain and boundary conditions, (b) the surface grid for the endwall, (c) the prism layer volume grid in the mainstream, channel, and plenum, and (d) the volume grid in the holes and impingement channel.

The solution was determined to be converged when the normalized residuals reached $1 \times 10^{-4}$ and the area-averaged endwall $\phi$ changed by less than 0.0015 over 500 iterations. A grid independence test was done for the H/D = 2.9 case. The initial grid of 9.8 million flow cells and 1.5 million endwall cells was uniformly refined to a grid containing a 16.1 million flow cells and 2.5 million endwall cells. For $M_{avg} = 1$, the area-averaged $\phi$ over the endwall changed by $10^{-4}$ from the initial grid to the refined grid. In addition, the total heat flux at the internal endwall surface varied by less than 0.25% from the initial grid to the refined grid. Therefore, it was concluded that the initial grid of 9.8 million flow cells and 1.5 million endwall cells was of
sufficient resolution for the conjugate heat transfer predictions, and the CFD solutions were grid insensitive.

4.4 Results and Discussion

The effect of varying the height of the impingement channel was investigated for both cases of impingement cooling only and combined impingement and film cooling. Experimental overall effectiveness, $\phi$, results were obtained for $H/D = 2.9$ and 0.6 at coolant flowrates corresponding to blowing ratios of $M_{avg} = 1$ and 2. The $Re_D$ of the impingement jets were approximately 1000 for $M_{avg} = 1$ and 1900 for $M_{avg} = 2$. Computational $\phi$ predictions were obtained for the combined film and impingement case. Additional impingement channel heights were examined with the computations, including $H/D = 5.8$ and 10.2, at blowing ratios of $M_{avg} = 1$ and 2. The trends of $\phi$ and internal Nusselt number, $Nu_{D,i}$, with $H/D$ were evaluated, and flow streamlines within the channel were examined to understand the trends.

Measurements of impingement only effectiveness, $\phi_o$, are shown along with the impingement hole locations in Figure 4.6 for both impingement channel heights and two impingement $Re_D$. The flowrates for the cases in Figure 4.6a and b are equivalent to a blowing ratio of 1 when film cooling is included. The flowrates of Figure 4.6c and d correspond to a blowing ratio of 2. Figure 4.6a and c are for $H/D = 2.9$ and Figure 4.6b and d are for $H/D = 0.6$. Figure 4.6a specifies direction of coolant exiting the impingement channel under the endwall. The effectiveness in the center of the impingement cooled area is less for $H/D = 0.6$ than for $H/D = 2.9$ as shown by the higher $\phi_o$ values for Figure 4.6a and c. This trend with $H/D$ is consistent with the literature for impingement cooling. Heat transfer coefficients slightly increase with $H/D$ up to a peak between 1.5 and 4 (Florschuetz et al., 1981; Viskanta, 1993). With $H/D = 0.6$ there is also increased $\phi_o$ (cooler wall) on the upstream suction side of the passage compared to $H/D = 2.9$. The increase on the upstream suction side is the result of internal cooling by the narrow channel flow that develops as the impingement jets are exhausted in that direction. Along the downstream half of the suction side of the blade, there is elevated effectiveness for all four cases. As the external flow near the endwall goes through the passage, it is swept from toward the suction side due to the large pressure gradient. This external flow develops a thermal boundary layer as it passes over the cooled endwall above the impingement array. The cooler boundary
layer fluid provides some cooling downstream of the array, along the downstream half of the suction side of the blade.

Figure 4.6 Contours of measured $\phi_\infty$, for (a) $M_{\text{avg}} = 1.0$, $H/D = 2.9$ (b) $M_{\text{avg}} = 1.0$, $H/D = 0.6$ (c) $M_{\text{avg}} = 2.0$, $H/D = 2.9$ (d) $M_{\text{avg}} = 2.0$, $H/D = 0.6$, with $90^\circ$ impingement holes and plenum boundaries overlaid.

Measurements of overall effectiveness for combined impingement and film cooling, $\phi$, are also reported for two blowing ratios and two impingement channel heights. Figure 4.7a and c show the results for $H/D = 2.9$, and Figure 4.7b and d show the results for $H/D = 0.6$. The contours are presented for two adjacent passages measured in the same experiment. Good agreement between the two passages reinforces confidence in the periodicity of the cascade. The impingement and film cooling hole locations are shown as well as the channel boundaries which prevented coolant from crossing from one passage to another in the channel above the impingement plate. Although there were significant differences in impingement only $\phi_\infty$ for different $H/D$, $\phi$ is almost indistinguishable between the two values of $H/D$ for the combined film cooling and impingement cases. Measurements of the average Nu for impingement jets staggered with angled extraction holes are available from Hollworth and Dagan (1980) for impingement jet spacing of 5D, close to the spacing used in the current study. The Nu results for staggered film cooling extraction are unchanged between $H/D = 1$ and 2.5 (Hollworth & Dagan, 1980), which is consistent with the trends of $\phi$ found in Figure 4.7.
Figure 4.7 Contours of measured $\phi$ for (a) $M_{avg} = 1.0, \ H/D = 2.9$ (b) $M_{avg} = 1.0, \ H/D = 0.6$, (c) $M_{avg} = 2.0, \ H/D = 2.9$, (d) $M_{avg} = 2.0, \ H/D = 0.6$, with 30° inclined film holes, 90° impingement holes, and plenum boundaries overlaid.

The contour data are laterally averaged across the pitch (y) direction and plotted as functions of axial distance in Figure 4.8a for impingement only, $\phi_0$, and in Figure 4.8b for combined impingement and film, $\phi$. The plot of impingement only in Figure 4.8a shows that the cases with H/D = 0.6 have higher laterally averaged effectiveness than H/D = 2.9 upstream of $x/C_{ax} = 0.05$. However, for most of the passage (0.1 < $x/C_{ax} < 0.6$), the case with H/D = 0.6 is less effective. For combined impingement and film cooling in Figure 4.8b, there is no significant change in laterally averaged effectiveness when H/D is varied between 2.9 and 0.6.

Computational simulations were completed to investigate the behavior of $\phi$ as H/D was increased beyond 2.9, to H/D = 5.8 and 10.2. The $\phi$ contours are shown for all impingement channel heights in Figure 4.9, for $M_{avg} = 1$ and 2. The impingement and film cooling holes and the channel boundaries are overlaid. In comparing the same conditions between Figure 4.7 and 4.9, the measured $\phi$ and computational predictions were found to match reasonably well, especially for $M_{avg} = 2$, where the computations correctly predict the detachment of the film cooling jets (Mensch et al., 2014). The agreement for $M_{avg} = 2$ can also be observed in the laterally averaged data in Figure 4.8a.
Figure 4.8 Pitchwise laterally averaged effectiveness plotted as a function of axial distance for different values of impingement channel spacing, H/D, (a) measured for impingement only, and (b) measured and predicted for film and impingement.

No significant differences exist between H/D = 0.6 and 2.9 in the predicted contours shown in Figure 4.9c, d, g and h, which is a similar trend to the experimental results in Figure 4.7. However, for M_{avg} = 1 and H/D = 0.6 in Figure 4.9d, the peak \( \phi \) in the center of the impingement area is highest at the smallest H/D. As H/D increases, the peak effectiveness decreases, and the area in the center of the impingement array becomes warmer. Although \( \phi \) decreases in the center of the impingement area, \( \phi \) increases on the pressure side of the passage near the blade with increasing H/D. At larger H/D, the effect of the impingement cooling spreads to cool a greater portion of the endwall area outside of the impingement array. At the smallest impingement channel height, the impingement cooling produces high effectiveness in the center, but lower effectiveness in the surrounding area. At the largest impingement channel height, the impingement cooling appears to have spread and generate a more uniform effectiveness distribution throughout the impingement and surrounding area. Both the effects of high peak impingement effectiveness with low H/D, as well as more uniform effectiveness with high H/D are consequences of the coolant flow behavior between the impingement plate and the bottom of the endwall. These competing effects combine to produce the area averaged trends found in the literature and found in the subsequent discussion of the area averaged overall effectiveness results.
Figure 4.9 Contours of predicted $\phi$ for (a) $M_{avg} = 1.0$, $H/D = 10.2$, (b) $M_{avg} = 1.0$, $H/D = 5.8$, (c) $M_{avg} = 1.0$, $H/D = 2.9$ (d) $M_{avg} = 1.0$, $H/D = 0.6$, (e) $M_{avg} = 2.0$, $H/D = 10.2$, (f) $M_{avg} = 2.0$, $H/D = 5.8$ (g) $M_{avg} = 2.0$, $H/D = 2.9$, (h) $M_{avg} = 2.0$, $H/D = 0.6$, with 30° inclined film holes, 90° impingement holes, and plenum boundaries overlaid.

Flow streamlines within the impingement channel are examined to better understand how the H/D affects the impingement cooling. In Figure 4.10, the non-dimensional temperatures and in-plane streamlines are given for a slice of the impingement channel at the $x/C_{ax} = 0.47$ plane, which passes through the center of the downstream row of impingement holes. The portion of the data shown focuses on the impingement hole in this row closest to the pressure side, as shown in Figure 4.10d. The predictions are provided at $M_{avg} = 2$ for $H/D = 10.2$ in Figure 4.10a, $H/D = 2.9$ in Figure 4.10b, and $H/D = 0.6$ in Figure 4.10c.

\[ \phi = \frac{T_{w} - T_{w,\infty}}{T_{x} - T_{w,\infty}} \]
Figure 4.10 Predicted in-plane streamlines and contours of non-dimensional temperature in the impingement channel for the first impingement hole in the last row for $M_{avg} = 2.0$ and (a) $H/D = 10.2$ (b) $H/D = 2.9$, (c) $H/D = 0.6$.

The streamlines for the two smaller impingement channel heights, $H/D = 0.6$ and $2.9$, exhibit the typical impingement jet behavior. The jet impacts the wall, and then moves away from the center of the jet along the wall, providing effective cooling for the area just above the jet. At the largest channel height, the flow is highly three-dimensional and does not exhibit impingement jet behavior. Also in the case of $H/D = 10.2$, the temperature of the flow reaching the endwall is much warmer than the jets in the smaller impingement channels. The warmer temperature and slower velocity of the coolant account for the lower $\phi$ in the area just above the impingement jets as $H/D$ increases. The reason for increased $\phi$ on the pressure side of the
passage as H/D increases can also be seen. At the highest channel spacing, Figure 4.10a, most of the coolant coming out of the jet flows to left, which is away from the other impingement jets and toward the pressure side of the channel. For the smaller H/D in Figure 4.10b and c, there is an inflow of warmer flow from the pressure side which blocks the spreading of the coolant. In the case of H/D = 0.6, the warmer air from the pressure side circulates counter to the impingement jet flow in a vortex approximately the size of the channel. In the case of H/D = 2.9, the vortex from the impingement jet flow is about half the height of the channel, so the flow coming from the pressure side can pass under the impingement vortex and mix with the coolant coming from the jet. Figure 4.10b shows that the temperature of the flow coming from the pressure side of the channel is cooled as it comes toward the jet.

The overall trends with H/D can be assessed by quantifying the area average of the overall effectiveness, which indicates the expected average wall temperature. The area used for the average, outlined in Figure 4.4b, surrounds the impingement holes and the portions of the endwall cooled by film cooling. Both the measurements and predictions of $\bar{\phi}$ are plotted in Figure 4.11 as a function of H/D. The measured $\bar{\phi}$ is slightly higher for H/D = 2.9 compared to H/D = 0.6. The predictions show a larger different in $\bar{\phi}$ between the smallest two channel heights than the measurements, but the predicted trend is the same direction as the measurements. As H/D increases beyond 2.9, the predicted $\bar{\phi}$ is reduced slightly, but overall there is not a significant change in $\bar{\phi}$ for this range of H/D.

The average internal heat transfer coefficient, $\bar{h_i}$, was calculated from the measurements of $\phi$ and $\phi_i$ at the four locations in Figure 4.4b using Equation (4.3) for film and impingement cooling at H/D = 2.9. The $\bar{h_i}$ was non-dimensionalized to Nusselt number based on $D$ and the thermal conductivity of the coolant. The coolant thermal conductivity was determined from a polynomial fit of the air conductivity varying with temperature using the measured internal coolant temperature, $T_{c,in}$. Figure 4.11 shows the measured $\text{Nu}_{D,i}$ on the right axis at blowing ratios of 1 and 2 for H/D = 2.9 in solid red symbols. The $\text{Nu}_{D,i}$ for other impingement channel heights were obtained from the computational predictions using Equation (4.4). The predicted $\text{Nu}_{D,i}$ results are area averaged across the area outlined in Figure 4.4b and are given in Figure
4.11 in red with open symbols and dotted lines. The measured and predicted $\text{Nu}_{D,i}$ are in good agreement, especially for $M_{\text{avg}} = 1$. Unlike $\phi$, there is a clearly defined peak in the area-averaged $\text{Nu}_{D,i}$ at $H/D = 2.9$. The existence of an optimal $H/D$ is consistent with the reported maximum impingement Nu in the literature, which occurs between $H/D$ of $1.5 – 4$ (Florschuetz et al., 1981; Hollworth & Dagan, 1980; Viskanta, 1993).

![Figure 4.11](image-url)

**Figure 4.11** Measured and predicted area-averaged $\phi$ and $\text{Nu}_{D,i}$ plotted as a function of impingement channel height for $M_{\text{avg}} = 1.0$ and $2.0$.

### 4.5 Conclusions

The current study demonstrated the application of a conjugate methodology to understand the effect of multiple cooling technologies on the overall heat transfer for a gas turbine endwall. This methodology can also be applied to understand other systems that have complex heat transfer interactions. Conjugate experiments and simulations were used to examine the overall cooling effect of the impingement channel height-to-diameter ratio in particular. Two channel heights were evaluated in the experiments. For the cases of impingement only, there were local differences in overall effectiveness between the two channel heights, but the overall
effectiveness for the cases of film and impingement were indistinguishable. The impingement channel height mattered less to the overall effectiveness of cases with film cooling, compared to the cases with impingement only.

Because the conjugate simulations had good agreement with the measurements for average effectiveness and heat transfer coefficients, the simulations provided predictions of these quantities for a wider range of channel heights. It was found that both the overall effectiveness and the internal heat transfer coefficients peaked at the channel height to diameter ratio of 2.9, similar to previous impingement literature data. The streamlines and temperature contours in the impingement channel showed that the smaller channel heights restrict spreading of the coolant outside of the area above the impingement array. With a larger channel spacing, the impingement flow was more three-dimensional. This type of flow led to reduced effectiveness right above the jet compared to the smaller heights, but increased effectiveness for the surrounding area.

The results from this study confirmed that internal heat transfer coefficients of impingement geometries were sensitive to geometric parameters, such as the impingement channel height. However, the average external surface temperatures of the endwall with combined film and impingement cooling was not particularly sensitive to the impingement channel height. The result of surface temperature insensitivity to impingement channel height provides a useful design consideration for turbine designers because the key metric for evaluating cooling technologies is not the heat transfer coefficient, but the predicted external surface temperature.
CHAPTER 5. OVERALL EFFECTIVENESS AND FLOWFIELD MEASUREMENTS FOR AN ENDWALL WITH NON-AXISYMMETRIC CONTOURING

Abstract

Endwall contouring is a technique used to reduce the strength and development of three-dimensional secondary flows in a turbine vane or blade passage in a gas turbine. The secondary flows locally affect the external heat transfer, particularly on the endwall surface. The combination of external and internal convective heat transfer along with solid conduction determines component temperatures, which affect the service life of turbine components. A conjugate heat transfer model is used to measure the non-dimensional external surface temperature, known as overall effectiveness, of an endwall with non-axisymmetric contouring. The endwall cooling methods include internal impingement cooling and external film cooling. Measured values of overall effectiveness show that endwall contouring reduces the impingement effectiveness alone, but increases the effectiveness of film cooling alone. Given the combined case of both impingement and film cooling, the laterally averaged overall effectiveness is not significantly changed between the flat and contoured endwall. Flowfield measurements indicate that the size and location of the passage vortex changes as film cooling is added and as the blowing ratio increases. Because endwall contouring can produce local effects on internal cooling and film cooling performance, the implications for heat transfer should be considered in endwall contour designs.

5.1 Introduction

The gas turbine industry continues to require higher overall engine efficiency and therefore more effective and efficient turbine component designs. Turbine cooling performance continues to be important to maintain component life and reliability as well. It may be possible to achieve improvements in both efficiency and cooling performance with three-dimensional endwall contouring. Endwall contouring can reduce the strength of the passage vortex and other

secondary flows, which cause increased heat transfer on the endwall surface. Although contouring reduces secondary flows, the impact of contouring on the heat transfer and cooling performance is not straightforward.

The overall cooling performance of a contoured endwall relative to a flat endwall depends on the local change of heat transfer coefficients as well as the changes in film coolant distribution. It is important to understand how traditional cooling methods will perform when applied to contoured endwalls. In the case of film cooling jets, the changing gradient of the surface may increase or decrease film cooling attachment and effectiveness. For internal impingement cooling, the impingement plate to target spacing will vary leading to a range of impingement cooling effects.

The current study examines the thermal performance of a contoured endwall with a typical endwall cooling configuration with impingement cooling and angled film cooling holes. Both surface thermal measurements and flowfield measurements are used to learn how the passage secondary flows and film cooling interact to influence the cooling performance of contoured endwalls. These types of data will allow turbine designers to generate cooling arrangements and methods that are optimized for contoured endwalls.

5.2 Relevant Literature

A conjugate heat transfer approach can be applied in experiments and computational simulations to determine the non-dimensional external metal temperature, also known as overall effectiveness. Overall effectiveness, $\phi$, is defined as the difference between the mainstream temperature and the external metal temperature, divided by the overall mainstream to coolant temperature difference. The $\phi$ definition along with its relationship to the important non-dimensional parameters will be described in the next section. Albert et al. (2004) was the first to derive an equation for $\phi$ by considering one-dimensional heat transfer. The equation demonstrated it is necessary to match the Biot number, $\text{Bi}$, as well as the ratio of internal-to-external heat transfer coefficients, $h_\infty/h_i$, to acquire relevant scaled data in a conjugate simulation or experiment.

The practice of reporting the values of these non-dimensional parameters only exists for more recent studies, although conjugate studies without matched $\text{Bi}$ and $h_\infty/h_i$ still provide conjugate results to compare to computational predictions for model validation. The first
conjugate heat transfer experiments for turbine applications, by Hylton et al. (1983, 1988) and Turner et al. (1985), provided a benchmark for computational work and improved understanding of the thermal fields of a conducting vane. Papanicolaou et al. (2001) and Panda and Prasad (2012) both compared computational predictions to experimental measurements of conjugate heat transfer for a film cooled flat plate. Panda and Prasad (2012) also studied a configuration that included internal impingement cooling. The qualitative effects of heat conduction within the wall were apparent in both studies, but quantitatively, the temperature distribution and gradients require a matched Bi and $h_{\infty}/h_i$ configuration.

Properly scaled conjugate heat transfer models have been used to determine the relative importance of impingement and film cooling to the overall effectiveness. The first experiments to use an engine matched Bi experimental model were completed by Sweeney and Rhodes (2000) for a three-dimensional flat plate with internal impingement and film cooling. Between the impingement plate and the wall were heat transfer enhancement features in a Lamilloy® snowflake design. Their results showed that impingement cooling dominated over film cooling in the distribution of external wall temperatures. A series of matched Bi experiments and simulations studied a vane leading edge model with internal jet impingement and external film cooling (Dobrowolski et al., 2009; Maikell et al., 2011; Mouzon et al., 2005; Ravelli et al., 2010). There was little difference between the cases with and without impingement due to the high effectiveness of the showerhead film cooling. Another series of engine matched Bi studies (Dyson et al., 2012; Stewart & Bogard, 2014; Williams et al., 2014) measured the effectiveness and thermal fields above a vane suction side with film cooling and impingement. Williams et al. (2014) showed that for this part of the vane, the impingement cooling contributed significantly to the overall effectiveness, even more than the contribution from film cooling. They concluded that as blowing ratio increased, the improvement in impingement cooling compensated for the reduction in film cooling effectiveness. As with the flat plate and the vane, the importance of impingement cooling for the endwall of an airfoil was observed by Mensch and Thole (2014).

Unlike a flat plate or vane surface, endwall external heat transfer coefficients are influenced by three-dimensional secondary flow structures, including the horseshoe and passage vortices, (Lynch et al., 2011b; 1999; Radomsky & Thole, 2000; Kang & Thole, 2000). Endwall heat transfer increases near the leading edge, pressure side and wake of the Pack-B blade used in the present study, as shown in Figure 5.1a with contours of Nusselt number, Nu, for a flat
endwall (2011b). Figure 5.1b gives the heat transfer augmentation due to non-axisymmetric endwall contouring designed by Praisner et al. (2007) to minimize passage aerodynamic losses (Lynch et al., 2011b).

![Figure 5.1](image)

**Figure 5.1 Heat transfer measurements for the Pack-B cascade at Re_{exit} of 2\times10^5** (Lynch et al., 2011b), (a) Nusselt number contours for the flat endwall, (b) heat transfer augmentation contours due to endwall contouring with box for area-averaging.

Endwall contouring can reduce aerodynamic losses by reducing the cross-passage pressure gradient and the strength of the passage vortex compared to a flat endwall (Knezevici et al., 2010). The effect of endwall contouring on the Pack-B passage vortex was measured by Lynch (2011) using laser Doppler velocimetry. From the three-dimensional average flow measurements, it was found contouring slightly reduced the strength of the passage vortex and delayed its progression to the suction side of the passage. Although Lynch (2011) and others have measured the passage flowfield in detail with and without endwall contouring, the effects of film cooling on the passage vortex development and strength have not been measured. Considering the film cooling jets at high blowing ratios detach from the endwall, the interactions between the passage vortex and film cooling are investigated in the current study.

Contouring can also reduce overall endwall heat transfer by the same mechanism as the reduction in losses (Lynch et al., 2011b; Saha & Acharya, 2008). Figure 1b shows that the Pack-
B endwall contouring increased heat transfer at the leading edge and upstream portions of the passage. However, the overall area-averaged heat transfer coefficient across the entire passage was reduced by 3% (Lynch et al., 2011b). Also, contouring was able to reduce heat transfer in the regions with the highest heat transfer near the pressure side. Although heat transfer augmentation due to endwall contouring provides the heat transfer performance of a design, the primary metric for cooling performance is the overall effectiveness, which is obtained with a conjugate model in the current study.

5.3 Conjugate Endwall Model

The conjugate heat transfer in the contoured endwall is approximated as one-dimensional using the conjugate endwall model, depicted in Figure 5.2. The endwall model includes the combined effects of the internal and external convective heat transfer as well as the solid heat conduction to obtain the scaled endwall temperatures. The data are engine relevant if the appropriate physical parameters are matched between the experimental model and engine conditions.

![Figure 5.2 One-dimensional conjugate model of a conducting endwall with impingement and film cooling.](image)

The relevant non-dimensional parameters can be found from Equation (5.1), which is derived from the one dimensional heat transfer circuit in Figure 5.2. Equation (5.1) reveals that the endwall overall effectiveness is a function of the Bi, $h_\infty/h_i$, and the product, $\chi_\eta$, which represents an estimate for the non-dimensional film temperature (Williams et al., 2014).
dimensionless film temperature is approximated by the adiabatic effectiveness, $\eta$, and a scale factor, $\chi_\eta$, which is defined in Equation (5.2). The coolant warming factor, $\chi_\eta$, corrects $\eta$ for the warming of the coolant during impingement and flow through the film cooling holes (Williams et al., 2014). Adiabatic effectiveness and the coolant warming factor are assumed to be functions of the geometry, the mainstream flow parameters, and film cooling flow parameters, such as film cooling blowing ratio, $M$.

$$ \phi = \frac{T_\infty - T_w}{T_\infty - T_{c,in}} = \frac{1 - \chi_\eta \eta}{1 + Bi + h_\infty / h_i} + \chi_\eta \eta $$

(5.1)

$$ \chi_\eta = \frac{T_\infty - T_{c,exit}}{T_\infty - T_{c,in}} $$

(5.2)

Table 5.1 lists the non-dimensional parameters matched to an engine for both the experiments and simulations in this study. Engine matched Bi is achieved by thermoforming the contoured endwall from Corian®. The contoured endwall heat transfer, $h_\infty$, is obtained from heat transfer measurements by Lynch et al. (2011b). A Nu correlation for an impingement array with staggered coolant extraction (Hollworth & Dagan, 1980) is used to estimate the range of $h_i$ for each blowing ratio. The blowing ratio, $M_{avg}$, affects $\chi_\eta \eta$, and defines the average ratio of coolant to mainstream mass flux.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Model</th>
<th>Typical Engine</th>
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</thead>
<tbody>
<tr>
<td>$M_{avg}$</td>
<td>0.6, 1.0, 2.0</td>
<td>1.0–2.0</td>
</tr>
<tr>
<td>$k_w$, W/m-K</td>
<td>0.99–1.06</td>
<td>22</td>
</tr>
<tr>
<td>$t_w$, cm</td>
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<td>0.20</td>
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<tr>
<td>$Bi = h_\infty t_w / k_w$</td>
<td>0.3–0.7 (Lynch et al., 2011b)</td>
<td>0.27</td>
</tr>
<tr>
<td>$M = 0.6$</td>
<td>1.0–2.3 (Hollworth &amp; Dagan, 1980; Lynch et al., 2011b)</td>
<td></td>
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<tr>
<td>$h_\infty / h_i$</td>
<td>$M = 1.0$ 0.6–1.4 (Hollworth &amp; Dagan, 1980; Lynch et al., 2011b)</td>
<td>1.0</td>
</tr>
<tr>
<td>$M = 2.0$</td>
<td>0.4–1.0 (Hollworth &amp; Dagan, 1980; Lynch et al., 2011b)</td>
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</table>
5.4 Experimental and Computational Methods

5.4.1 Experimental Methods

Experiments to measure the contoured endwall overall effectiveness and the passage flowfield were completed using a large-scale, low speed, closed loop wind tunnel, shown in Figure 5.3a. In this facility, the flow was split upstream of the test section into a mainstream section in the center, and into coolant streams above and below the mainstream section. The mainstream flow passed through a heater bank and flow conditioning elements including a turbulence grid $\sim 11C_{ax}$ upstream of the test section. More details about the wind tunnel and flow conditioning can be found in Lynch et al. (2011b).

![Figure 5.3 Depiction of (a) the large-scale low-speed wind tunnel, (b) the test section containing the Pack-B linear blade cascade and conjugate endwall, and (c) the side view of the plenum and impingement channel for the flat endwall.](image-url)
The corner test section contained seven blades in a linear cascade as shown in Figure 5.3b. The low-pressure turbine Pack-B airfoil was used in the cascade because it has been used in several other studies, particularly studies of contoured endwall performance (Knezevici et al., 2010; Lawson et al., 2013; Lynch et al., 2011a, 2011b; Mensch et al., 2014; Mensch & Thole, 2014; Praisner et al., 2007). The non-axisymmetric contouring designed by Praisner et al. (2007) was implemented for the endwall. The blue colored portion of the endwall in Figure 5.3b was conducting and contained the contoured geometry. The rest of the endwall outside of the passages was made from insulating medium density fiberboard. Passages 3 and 4 were cooled by both internal impingement cooling and film cooling. Passages 1 and 2 had film cooling but no impingement cooling. Passages 5 and 6 had only impingement cooling, and the coolant was exhausted laterally from the side of the impingement channel. There was good periodicity between the blades in the cascade as shown in Figure 5.4, which gives the pressure distribution measured at the midspan of each blade as a function of axial distance. The computational prediction of pressure distribution from two previous studies (Lynch et al., 2011a; Mensch et al., 2014) is shown for comparison.

\[ C_p = \frac{P_s - P_{s,\infty}}{1/2 \rho \frac{U_{\infty}}{\sqrt{2}}} \]

Figure 5.4 Pack-B cascade static pressure distribution at the blade midspan compared to CFD predictions.
The inlet conditions to the passage were measured at multiple spanwise and pitchwise locations at 0.52C$_{ax}$ upstream of the blades in the axial direction. A thermocouple rake was used to measure the incoming flow temperature. The variation in temperature at different locations was less than ±0.6°C from the average T$_{\infty}$. The incoming flow velocity was measured using a Pitot probe at midspan of the test section. The mainstream inlet velocity, U$_{x,\text{in}}$, had a standard deviation of less than 1%. Table 5.2 provides a summary of the blade geometry and mainstream flow conditions. Additionally the inlet boundary layer was characterized at 2.85C$_{ax}$ upstream of the center blade to have δ/S = 0.061, and 6% freestream turbulence (Lynch et al., 2011b).

<table>
<thead>
<tr>
<th>Table 5.2 Flow Conditions and Blade Geometry</th>
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<tr>
<td>Scale factor</td>
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<tr>
<td>C$_{ax}$</td>
</tr>
<tr>
<td>p/C$_{ax}$</td>
</tr>
<tr>
<td>S/C$_{ax}$</td>
</tr>
<tr>
<td>Inlet Re</td>
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<tr>
<td>Exit Re</td>
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</table>

The coolant stream was extracted by a blower at the top of the wind tunnel as shown in Figure 5.3a. The coolant was dried, to remove any humidity, cooled and then diverted to separate plenums below the endwall as shown in Figure 5.3c. The coolant flowrate was adjusted to achieve the required blowing ratio and measured using a laminar flow element (LFE). The blowing ratio reported was an average of the local blowing ratio for each film cooling hole. The coolant flowrate uncertainty was estimated with sequential perturbation (Moffat, 1988) to be ±3% for a 95% confidence interval. The coolant temperature, T$_{c,\text{in}}$, was measured with two thermocouples suspended in each plenum 8.7D below the cooling holes in the impingement plate below the endwall when no impingement plate is used. The mainstream to coolant temperature difference was typically 40°C, which yielded a coolant to mainstream density ratio of about 1.15.

The contoured endwall was cooled by an impingement and film cooling configuration designed for comparison to the impingement and film cooled flat endwall used in Mensch et al. (2014) and Mensch & Thole (2014, 2015a). A flat impingement plate with an array of 28 staggered holes was located below the endwall as shown in Figure 5.3c. The coolant exited the impingement channel through 10 film cooling holes angled at 30° to the endwall surface at the hole exit. As mentioned previously, the coolant exited through a slot on the side of the
impingement channel when film cooling was not used. The impingement holes and film cooling holes all had a diameter $D = 4.4$ mm. The spacing between impingement holes was $4.65D$.

Special attention was given to the location of the film cooling holes for both the flat and contoured endwall designs. For the flat endwall the inlets to the film cooling holes were located between the rows of impingement holes. The $30^\circ$ angled film cooling holes were aligned in the $x$-$y$ plane to match the direction of the qualitative endwall streaklines. The endwall streaklines and film cooling hole locations for both flat and contoured endwalls are shown in Figure 5.5a and b (Lynch et al., 2011b). In the contoured endwall, the film cooling exit locations were at the same location as the flat endwall configuration in order to compare the film cooling behavior at the same location. Since the endwall streaklines differ between the flat and contoured endwalls, the orientation of the film cooling holes changed from the flat to contoured endwalls, especially in the row that extended downstream across the passage. Because of the changing hole orientation as well as the contour itself, the size and location of the film cooling hole inlets are different between the flat and contoured endwalls. For the case of the contoured endwall, a few of the film cooling hole inlets are larger and closer to the impingement hole locations than the flat endwall film cooling holes. The length of the film cooling holes, $L$, also varies between $4.2D$ and $8.0D$ for the contoured endwall, while all of the flat endwall film cooling holes have $L = 5.8D$.

A qualitative representation of the hills and valleys of the endwall contouring is given in Figure 5.5c. The design incorporated a hill on the pressure side and valley on the suction side in the upstream half of the passage to reduce cross-passage pressure gradients (Praisner et al., 2007). A ridge and valley in the downstream half of the passage helped displace the passage vortex and reduce exit flow angle deviation, as shown by the streaklines in Figure 5.5b compared to Figure 5.5a. The contoured endwall in this study was constructed with a constant thickness to retain a relatively uniform $Bi$ across the passage, which was similar to the flat endwall. A consequence of the constant thickness endwall was a varying distance between the flat impingement plate and the endwall, $H$. The nominal $H$ at the inlet and exit of the passage was the same $H$ used for the flat endwall, 2.9D. For the contoured endwall, however, $H$ varied between 0.6D to 3.4D.
Figure 5.5 Comparison of oil flow visualization of endwall streaklines (Lynch et al., 2011b) with film cooling hole inlet and outlet locations for the (a) flat and (b) contoured endwalls, and (c) qualitative representation of the contoured endwall height.

Temperatures on the surface were measured with infrared (IR) thermography using a FLIR P20 IR camera. The ceiling of the test section contained removable viewing ports to allow direct optical access and achieve an image resolution of 5.7 pixels/D. At least two thermocouples per image area were imbedded in the endwall with the bead at the surface. When the thermocouples indicated that steady-state was achieved, five images were acquired from each port. The images at each location were calibrated for emissivity and reflected temperature by minimizing the difference between the thermocouple measurements and the image temperatures at the thermocouple locations. The emissivity was typically 0.92 from the flat black paint on the surface. The calibrated images at each location are averaged together and assembled into a complete endwall temperature map. The uncertainty in the external effectiveness was estimated as ± 0.02 for a 95% confidence interval (Mensch & Thole, 2014).

5.4.2 Computational Methods

Conjugate RANS simulations for the contoured endwall were performed using commercial software (FLUENT 13.0.0, 2010) which uses the segregated pressure-based SIMPLE algorithm. The SST k-ω turbulence model (Menter, 1994) was used for closure with second-order spatial discretization schemes. The computational grid is depicted in Figure 5.6a, which
shows the boundary conditions. Periodic boundaries extending through the entire domain, were used to simulate a single blade passage. The specified velocity and boundary layer profile at the inlet were benchmarked to the measurements by Lynch et al. (2011b). An outflow boundary condition was applied at the outlet, and a symmetry boundary condition was applied at the top of the domain. A mass flow inlet boundary condition was applied at the bottom of the plenum to correspond to the flow and temperature conditions associated with each blowing ratio. Conjugate interfaces were thermally coupled, and all other surfaces were adiabatic. Convergence of a simulation was achieved when the normalized residuals were less than $1 \times 10^{-4}$ and the area-averaged endwall $\phi$ changed by less than 0.0015 over 500 iterations. Additional details regarding the solution methods and boundary conditions can be found in Mensch et al. (2014).

Unstructured grids were generated for the endwall and flow domains using (Pointwise 17.1r3, 2013). The flow domain grid contained wall-normal prism layers to resolve the boundary layers such that the $y^+$ of the first grid point was less than 1. Figure 5.6b shows a slice of the grid in the impingement channel, impingement hole and film cooling hole. The grid size was 10.6 million cells for the flow domain and 1.8 million cells for the endwall domain. A grid sensitivity study was conducted for the case of the flat endwall mesh generated in the same manner as the contoured endwall mesh (Mensch et al., 2014).

Figure 5.6 Depiction of (a) the computational domain and boundary conditions and (b) the prism layer volume grid in the holes and impingement channel.
5.4.3 Flowfield Measurement Methods

Time resolved particle image velocimetry (PIV) was used to measure the flowfield in three two-dimensional planes near the trailing edge of the passage suction side as shown in Figure 5.7. Planes A and B were parallel to the exit flow direction, 5D and 3D from the center of the centerline of the blade trailing edge, respectively. Plane C was parallel to the trailing edge of the passage at an axial location of 1.08C_{ax} from the leading edge. Plane C was at an angle of 30° from Planes A and B and crossed the other planes near the trailing edge as shown in Figure 5.7a. Figure 5.7b shows the approximate locations where Planes A and B cross Plane C while viewing the flat endwall predicted contours of turbulent kinetic energy in Plane C. Figure 5.7a also shows the relative locations of the film and impingement holes in the endwall.

![Figure 5.7](image)

**Figure 5.7** Planes measured with PIV (a) shown from above and (b) shown from the view of Plane C overlaid with flat endwall CFD tke contours for M_{avg} = 2.0.

During the PIV measurements, the flow was seeded with atomized liquid di-ethyl-hexyl-sebecat (DEHS), resulting in an average Stokes number much less than one. The particles in the measurement planes were illuminated with an Nd:Nd:YLF laser sheet and imaged with a high speed CMOS camera oriented normal to each measurement plane. Images at Planes A and B were taken for a window size of 1024 x 1024 at 1 kHz and a resolution of 22 pixels/D. Images at Plane C were taken for a window size of 1024 x 512 at 2 kHz and a resolution of 16 pixels/D.
The total number of image pairs recorded was 3000 at Planes A and B, and 6000 at Plane C. The total data acquisition time for all three planes was 3 s. The flow crossed the domain at least 90 times in the data acquisition period. The time delay between laser pulses was chosen based on an estimated bulk movement of 10 pixels, about 1/6 the initial interrogation window size.

Image processing and vector calculation were performed with LaVision software (*DaVis 8.1.4, 2012*). To increase the contrast of the raw images, the sliding minimum of the surrounding three images was subtracted from each image prior to vector calculation. The vector calculation was performed using a cross-correlation over four passes, with a decreasing interrogation window at each pass. Vector post-processing evaluated possible spurious vectors through two passes of a median filter using universal outlier detection for a 3 x 3 pixel region (*DaVis 8.1.4, 2012*). Empty spaces surrounded by at least two calculated vectors were filled by interpolating between the surrounding vectors.

5.5 Results and Discussion

5.5.1 Effects of Contouring on Overall Effectiveness

Measurements and computational predictions of the contoured endwall overall effectiveness $\phi$ are shown in Figure 5.8 for blowing ratios of $M = 1$ and 2. Overlaid on the contour plots are the locations of the film cooling holes, impingement holes, plenum boundaries and an elevation map for the height of the contoured endwall. Dotted lines indicate negative height values where there is a valley. The measured results are shown for Passage 4. The area used for averaging is shown by the box in Figure 5.1b.

In general there is good agreement between the predictions and the measurements of contoured endwall effectiveness except for the common failure of RANS to accurately predict film cooling jet mixing and attachment (Dyson et al., 2012; Foroutan & Yavuzkurt, 2014; Stewart & Bogard, 2014). A similar over-prediction of film jet attachment was found for the flat endwall simulation in Mensch et al. (2014). Downstream in the passage the predicted $\phi$ is about 0.05 less than the measurements. This difference between the measured and predicted $\phi$ downstream was also observed for the flat endwall and was attributed to heat losses in the experiments from boundaries assigned as adiabatic in the simulations (Mensch et al., 2014).
Figure 5.8 Contoured endwall overall effectiveness for (a) $M_{avg} = 1.0$ measured, (b) $M_{avg} = 1.0$ predicted, (c) $M_{avg} = 2.0$ measured, and (d) $M_{avg} = 2.0$ predicted.

Figure 5.9 compares the flat endwall to contoured endwall overall effectiveness with film and impingement cooling. The locations of the cooling holes and plenum boundaries are shown for all cases. The change in height of the contoured endwall is also shown with elevation lines in Figure 5.9d-f. Figure 5.10 shows the laterally averaged effectiveness as a function of the axial distance across the passage for both flat and contoured endwalls. The data in Figure 5.10a, for the cases with film and impingement cooling, correspond to the contour plots in Figure 5.9. Figure 5.10b and c compare the effectiveness for the cases with film cooling only and impingement only, respectively. The overall effectiveness peaks around the first row of film cooling holes and increases with blowing ratio as discussed in (Mensch & Thole, 2014).

Generally, the results for the contoured endwall are similar to those for the flat endwall as shown in Figure 5.9 and Figure 5.10. One difference seen in Figure 5.8 is that there is slightly increased effectiveness due to film cooling with contouring, especially for the upstream row of
film cooling holes at $M_{\text{avg}} = 1.0$. Improved film cooling effectiveness can also be seen in Figure 5.10b where the peak effectiveness is higher for the contoured endwall than for the flat endwall.

![Image of effectiveness measurements](image)

**Figure 5.9** Measured endwall overall effectiveness for the flat endwall (Mensch & Thole, 2014) at (a) $M_{\text{avg}} = 0.6$, (b) 1.0, and (c) 2.0, and for the contoured endwall at (d) $M_{\text{avg}} = 0.6$, (e) 1.0, and (f) 2.0.

Endwall contouring reduces the effect of the passage vortex on the endwall flow, as can be seen in the oil flow visualization of the endwall streaklines in Figure 5.5a and b. With a contoured endwall, the film cooling jets are less disturbed by the weaker passage vortex. Additionally, it is observed in Figure 5.9 that the holes in the first row are near a valley of the contour and may experience a more favorable pressure distribution coming out of the valley, which promotes jet attachment.
Figure 5.10 Laterally averaged overall effectiveness for (a) film and impingement, (b) film cooling only, and (c) impingement only, measured for the flat and contoured endwalls.

When comparing the contoured endwall to the flat endwall in a conjugate analysis, one must consider both the change in film cooling (local fluid driving temperature) and the change in the external heat transfer coefficient. In the case where the external heat transfer coefficient increases due to endwall contouring and the film coolant stays along the surface, one would expect a lower wall temperature (higher effectiveness value). In the case where the external heat transfer coefficient increases but the film coolant is separated from the surface, one would expect a higher wall temperature (lower effectiveness value). There are also possibilities in between that give competing effects.

In reviewing the effectiveness results in Figure 5.9 and Figure 5.10, it can be seen that the film cooling effectiveness improves with contouring. Because the external $h_x$ is higher with contouring in the upstream half of the passage (Figure 2.2b) there is an overall reduction of the wall temperature (higher effectiveness value). The boxed area in Figure 2.2b surrounds the region containing the impingement and film cooling holes. Within this area, the average heat transfer augmentation is +4% with contouring. Along the pressure side of the blade, there is a 20-30% reduction in $h_x$. Therefore, in the case with no external cooling, the contoured endwall increases the $\phi$ by 0.1-0.15 for $M_{avg} = 1.0$ and 2.0, which results in lower wall temperatures.

Although contouring improves the effectiveness of the external cooling, the internal cooling appears to be less effective with contouring. The contoured endwall laterally averaged effectiveness with impingement only, $\overline{\phi_{o}}$, in Figure 5.10c is similar qualitatively to the flat endwall, but the level of effectiveness is considerably less for the contoured endwall compared to
the flat endwall. Since the endwall thickness is uniform and the same as the flat endwall, the
differences arise from the changes that the contoured endwall has on the internal and external
heat transfer coefficients. The internal $h_i$ is affected by the contouring because the H/D varies
across the passage, which locally reduces $h_i$ from its peak performance at H/D ≈ 3 (Mensch &
Thole, 2015a). Although the variation in H/D contributes to lower $\bar{\phi}_o$, the H/D effect does not
fully account for the reduction because the area averaged $\bar{\phi}_o$ with contouring is less than the area
averaged $\bar{\phi}_o$ for a flat endwall with non-optimal H/D, which was measured by Mensch and Thole
(2015a). Reduced $\bar{\phi}_o$ is also caused by the 4% increase in external $h_\infty$ for the area surrounding
the cooling features. For impingement cooling only, the external driving temperature is $T_\infty$, and
a higher $h_\infty$ results in higher wall temperatures. In the other configurations with film cooling, the
presence of cooler fluid in the film jets reduces the external driving temperature. Therefore, with
film cooling, the increase in $h_\infty$ does not result in lower effectiveness.

5.5.2 Effects of Film Cooling on Contoured Flowfield

The average velocity vector field and turbulent kinetic energy (tke) fields are obtained
with time resolved PIV for the planes in Figure 5.7. The three trailing edge planes are of interest
because the passage vortex causes an increase in endwall heat transfer in this region (Lynch et
al., 2011b). The time-averaged in-plane streamlines are given in Figure 5.11 for no film cooling
and for two blowing ratios, $M_{\text{avg}} = 1.0$ and 2.0. The main component of velocity is in-plane for
Planes A and B, whereas Plane C has a significant out of plane component. The line of sight
location of the blade trailing edge is shown in each image for reference. The background of each
image is colored by the measured velocity magnitude normalized by the inlet $U_{\infty,\text{in}}$.

The low velocity region in Planes A and B in Figure 5.11 is assumed to be the location of
the top of the passage vortex as it circulates through the measurement planes, which are
approximately aligned with the inviscid flow direction. The streamlines within the low velocity
region converge downstream of the trailing edge line, indicating a significant out of plane
velocity component associated with the rotating vortex. The top of the vortex is moving away
from the endwall as the streamlines below the top of the vortex are turned upward. The passage
vortex appears to be very close to the blade, since the velocities in Plane B are much lower than
in Plane A. In Plane C, the passage vortex appears slightly to the right of the line of sight.
location of the trailing edge because the flow is exiting the passage at a 30° angle to Plane C and the flow exiting the passage is 6.9D to the right of its location at the blade trailing edge (see Figure 5.7a).

Figure 5.11 In-plane time-averaged streamlines measured with PIV, colored by velocity magnitude for the contoured endwall for (a-c) no film cooling, (d-f) $M_{avg} = 1.0$, and (g-i) $M_{avg} = 2.0$.

The approximate $z$ (spanwise) location of the top of the passage vortex, or the center of the low velocity area, is labeled in Figure 5.11a as $Z_p$. As film cooling is introduced in Figure 5.11d-i, $Z_p$ increases. Table 5.3 reports the non-dimensional $Z_p/S$ for all nine images in Figure 5.11. The endwall film cooling not only increases the height of the passage vortex, but also appears to fill some of the velocity deficit, especially with the highest blowing ratio. The low-
velocity region shrinks in Plane A with each increase in blowing ratio. In Planes B and C, the velocity increase is more significant for the change from $M_{\text{avg}} = 1.0$ to 2.0. There are other small changes in the shape of the low-velocity region with film cooling for all three planes. In Plane C, although the height of the passage vortex from the endwall is not significantly changed with blowing ratio, the width of the passage vortex grows as blowing ratio increases.

Table 5.3 Distance from the Endwall of the Center of the Low Velocity Region $Z_p/S$

<table>
<thead>
<tr>
<th></th>
<th>Plane A</th>
<th>Plane B</th>
<th>Plane C</th>
</tr>
</thead>
<tbody>
<tr>
<td>No Film Cooling</td>
<td>0.13</td>
<td>0.13</td>
<td>0.16</td>
</tr>
<tr>
<td>$M_{\text{avg}} = 1.0$</td>
<td>0.15</td>
<td>0.15</td>
<td>0.16</td>
</tr>
<tr>
<td>$M_{\text{avg}} = 2.0$</td>
<td>0.16</td>
<td>0.16</td>
<td>0.16</td>
</tr>
</tbody>
</table>

Contours of normalized turbulent kinetic energy (tke) are given in Figure 5.12 for no film cooling a-c, for $M_{\text{avg}} = 1.0$ d-f, and for $M_{\text{avg}} = 2.0$ g-i. The line of sight location of the blade trailing edge is shown in each image for reference. For a two-dimensional velocity field, tke is defined as $\frac{3}{2}(\overline{u^2} + \overline{v^2})$, which assumes the root mean square of the out of plane velocity fluctuations is an average of the other two components (DaVis 8.1.4, 2012). The tke is non-dimensionalized by the inlet velocity, $U_{\infty,\text{in}}^2$.

The contours of tke characterize the turbulence generated by shear layers as well as the unsteadiness of large flow structures such as the passage vortex and film cooling jets. In the mainstream flow outside of the boundary layer, tke/$U_{\infty,\text{in}}^2$ is quite low and only reflects the level entering from the upstream grid as expected. Consistent with the average velocity data, the shear layer at the top of the passage vortex clearly rises moving downstream. In Figure 5.12c, without film cooling, there are two distinct regions of maximum tke, indicating there is a counter rotating vortex located above the passage vortex. Figure 5.12b indicates the presence of another vortex, with two bands of maximum tke on top of one another. With the addition of film cooling, these bands become clearer in Figure 5.12e and h. The presence of a second vortex also explains the elongated region of low velocity observed in the average velocity contours of Figure 5.11c, f, and i. This vortex originates from the suction side leg of the horseshoe vortex (Goldstein & Spores, 1988), or the second vortex could be a vortex induced by the passage vortex (H. P. Wang et al., 1997). Goldstein and Spores (1988) noted that both this second vortex and the passage
vortex move away from the endwall along the blade on the suction side of the passage because the pressure is lower away from the endwall.

Figure 5.12 Turbulent kinetic energy measured with PIV for the contoured endwall for (a-c) no film cooling, (d-f) $M_{\text{avg}} = 1.0$, and (g-i) $M_{\text{avg}} = 2.0$.

Film cooling has a significant impact on the tke at the upstream side of the images in Figure 5.12d and g. The levels of tke at the upstream side of Plane A approximately double with each increase in blowing ratio. The peak level of tke in Figure 5.12g is 0.19. For both blowing ratios tested, the film cooling jets are not well attached and mix into the passage vortex. Although the last film cooling hole in the diagonal row exits at approximately 40D upstream of the Plane A images, film cooling has a significant impact on the turbulence levels downstream. Film cooling also increases tke levels of the passage vortex in the upstream half of Plane B. The other vortex is weakened with film cooling, as seen on the downstream half of Plane B and at the
trailing edge line in Plane C. With film cooling, Plane C also has a third peak in tke, located to the left of the trailing edge line and below the peaks associated with the vortices. This peak appears for $M_{avg} = 1$ and becomes more pronounced at $M_{avg} = 2$. This region is also just below the extended region of low velocity in Figure 5.11i.

Under the regions of peak tke in Figure 5.12, the tke levels increase as blowing ratio increases in all three measurement planes. Film cooling causes increased mixing near the wall, which promotes heat transfer. The thermal effect of this mixing can be seen in the non-dimensional wall temperature contours of Figure 5.9. As the secondary flows cross the passage, they entrain cooler flow from the film cooling as well as from the endwall thermal boundary layer. The cooler secondary flow cools the endwall along the suction side trailing edge of the passage in Figure 5.9. The cooling effect increases with blowing ratio because there is more coolant injected into the mainstream, having two effects. As blowing ratio increases, the temperature of the secondary flow is cooler, and there is increased near wall mixing downstream from the film cooling as demonstrated in Figure 5.12.

5.6 Conclusions

The current study examines the effects of endwall contouring on the overall effectiveness. Previous research has shown that endwall contouring reduces the area-averaged heat transfer coefficient by delaying the development of the passage vortex and other passage secondary flows. However the effects of contouring on thermal performance are more complicated than the effect on the average heat transfer coefficient. The overall cooling effectiveness depends on the local heat transfer coefficients and on the surrounding thermal fields, which are determined by the effectiveness of the cooling methods with endwall contouring.

Film cooling effectiveness improves with endwall contouring because the weakened passage vortex has less effect on the near endwall flow and cooling jet mixing with contouring. The flowfield measurements of the passage vortex show that the passage vortex convects away from the endwall towards the midspan. Film cooling significantly affects the trajectory of the passage vortex by increasing the convective motion towards the midspan. From the overall effectiveness measurements, it can also be seen that the influence of the passage vortex on the cooling of the endwall near the trailing edge is diminished for the contoured endwall compared
to the flat endwall. Another reason for improved film cooling effectiveness with contouring is that the film cooling jets can have better attachment to the surface compared to a flat endwall if there is a local favorable pressure gradient due to the local surface gradient in the contouring. Although the conjugate heat transfer simulations can satisfactorily predict area-averaged overall effectiveness, their failure to accurately predict local film cooling effectiveness limits the benefit of RANS predictions for when film cooling is important. Although film cooling effectiveness improves with contouring, the internal impingement effectiveness worsens with contouring because the varying H/D for the contoured endwall causes a reduction in the impingement performance.

Local changes in the external heat transfer coefficient with endwall contouring can either increase or decrease the local cooling performance depending on the presence or lack of film coolant above the endwall. Increased $h_\infty$ improves the overall effectiveness when film cooling is present, but decreases the overall effectiveness (raises the wall temperature) when there is no film cooling. The increased $h_\infty$ in the upstream half of the contoured passage, relative to the flat endwall, can improve cooling because the presence of the film coolant lowers the effective external driving temperature. However, an endwall without film cooling and only impingement cooling has worse performance with contouring due to increased $h_\infty$. Conversely, decreased $h_\infty$ improves the overall effectiveness when film cooling is not present but internal cooling is present. An example is the improved overall effectiveness along the pressure side where there is not film cooling and $h_\infty$ is reduced with contouring.

For the current contoured endwall cooling geometry, the performance for the flat and contoured endwalls is mostly the same when comparing the laterally averaged results, although local variations are present. In general, the cooling performance with contouring could increase or decrease depending on the local effects. In order to optimize contoured endwall performance, film cooling and internal cooling should be fully integrated into the design for contoured endwalls.
CHAPTER 6.  EFFECTS OF NON-AXISYMMETRIC ENDWALL CONTOURING AND FILM COOLING ON THE PASSAGE FLOWFIELD IN A LINEAR TURBINE CASCADE

Abstract

The time-resolved flowfield is measured in the passage of a linear turbine cascade to show the effects of passage film cooling and non-axisymmetric endwall contouring on the passage secondary flows. A particle image velocimetry system is used in three measurement planes: the plane at the exit of the passage, and two streamwise planes along the blade suction side. In the downstream half of the passage, the passage vortex moves away from the endwall toward the midspan, but closely follows the profile of the blade suction side. The secondary velocity vectors and vorticity fields in the passage exit plane indicate the large size of the passage vortex in the time mean flowfield. The measured velocities in the streamwise measurement planes reveal the trajectory of the passage vortex as well as steep gradients in the direction perpendicular to the blade surface. The turbulent kinetic energy and unsteadiness of the velocities are elevated in the passage vortex. Film cooling increases the passage vortex size, secondary velocities and exit plane turbulent kinetic energy, while endwall contouring reduces the passage vortex size, secondary velocities and exit plane turbulent kinetic energy. Film cooling and endwall contouring have fewer effects on the turbulent kinetic energy in the streamwise planes.

6.1 Introduction

Film cooling and endwall contouring are important aspects of the design of gas turbine airfoils and their associated endwalls. Film cooling is a frequently used technique to protect turbine airfoil surfaces from hot mainstream temperatures by injecting cooler air from the compressor through small angled holes in the airfoil walls to simultaneously provide internal convective cooling as well as a protective barrier between the external airfoil wall and the hot gas flow. Although film cooling has been extensively studied for optimizing the overall cooling

† Mensch, A. and Thole, K.A., “Effects of Non-Axisymmetric Endwall Contouring and Film Cooling on the Passage Flowfield in a Linear Turbine Cascade.” To be submitted to Experiments in Fluids.
performance and local flowfield effects, the effect of endwall film cooling injection on the endwall secondary flows is relatively unknown and generally neglected.

Gas turbine designers also use three-dimensional non-axisymmetric endwall contouring to reduce pressure losses through the passage by limiting the development of secondary flow structures such as the passage vortex and other vortices. The secondary flow structures generally produce negative effects for the turbine aerodynamic efficiency and endwall heat transfer. The vortices generate turbulent mixing and dissipation, reducing the energy of the flow. Increased mixing increases endwall heat transfer coefficients and reduces the effectiveness of film cooling jets. Endwall contouring has the potential to achieve improvements in both aerodynamic efficiency as well as cooling performance through non-axisymmetric endwall contouring.

The following systematic flowfield study provides time-resolved measurements of the passage secondary flows with and without endwall film cooling for both flat and contoured endwalls. A typical endwall cooling configuration is used to provide relevant flowfield results. The measurements provide insight into the complex three-dimensional flowfield that exists above a turbine airfoil endwall, and how the effects of film cooling and endwall contouring enhance or hinder the strength and size of secondary vortical flows.

6.2 Experimental Methods

The passage flowfield is measured in a large-scale, low speed, closed loop wind tunnel, shown in Figure 6.1a. The test section on the right side of Figure 6.1a contained a linear cascade of seven blades and six blade passages as shown in the top view of Figure 6.1b. Upstream of the test section, the flow is separated into mainstream and coolant sections where the wind tunnel cross section widens. The coolant flow is diverted into the top blue section of the tunnel away from the mainstream flow below. Before entering the test section, the mainstream flow passes through a heater bank and flow conditioning elements. A turbulence grid is located $11C_{ax}$ upstream of the test section to generate 6% freestream turbulence at the blade inlet (Lynch et al., 2011b).
Figure 6.1 Depiction of (a) the large-scale low-speed wind tunnel, (b) the test section containing the Pack-B linear blade cascade and conjugate endwall, and (c) the side view of the plenum and impingement channel for the flat endwall.

The profile of the blades in the corner test section comes from the low-pressure turbine Pack-B airfoil. This blade profile has been widely used in the literature for studying contoured endwall aerodynamic performance (Knezevici et al., 2010; Praisner et al., 2007) and endwall heat transfer performance (Lynch et al., 2011b, 2011a; Lawson & Thole, 2012a; Mensch & Thole, 2014, 2015b). Figure 6.1b shows the top view of the blade cascade with the blue portion of the endwall showing the region between the blades known as the blade passage. The axial direction is shown by the x-direction, and the blade pitch is in the y-direction. The blades have a constant profile in the span (z-direction). The flowfield measurements are obtained near the trailing edge of Passage 4. The endwall in Passage 4 is internally cooled by an array of impingement jets feeding angled film cooling holes in the endwall as shown in Figure 6.1c.
The flow through the linear blade cascade is ensured to be periodic and match the flow through an infinite cascade by adjusting flexible walls within the test section. Periodicity is verified by measuring the pressure distribution around each blade at the midspan. A typical result for the experimental pressure distribution of the blades is shown in Figure 6.2 along with two predictions of pressure distribution obtained from periodic CFD simulations (Lynch et al., 2011a; Mensch et al., 2014). Figure 6.2 shows that the linear cascade provides good periodicity in the flow between the passages.

![Figure 6.2 Pack-B cascade static pressure distribution at the blade midspan compared to CFD predictions.](image)

Passage inlet flow and temperature conditions are measured in front of each blade and across the blade span. A Pitot probe and thermocouple rake are each inserted into the test section at 0.52C_{ax} upstream of the blades in the axial direction. The mainstream flow velocity at the inlet midspan, U_{x,in}, is found by averaging the velocities in front of each blade obtained from the Pitot probe. The standard deviation of U_{x,in} is less than 1%. The thermocouple rake measures the temperature distribution at four spanwise locations and multiple pitchwise locations at the inlet plane. The average T_{x} is used to calculate the mainstream density for Re and blowing ratio, M, calculations. The variation in temperature at different locations was less than ±0.6°C from
the average $T_\infty$. The blade geometry and mainstream flow conditions are given in Table 6.1. The inlet boundary layer was measured at $2.85C_{ax}$ upstream of the center blade by Lynch et al. (2011b) to have a boundary layer thickness of $\delta/S = 0.061$.

<table>
<thead>
<tr>
<th>Table 6.1 Flow Conditions and Blade Geometry</th>
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<tr>
<td>Scale factor</td>
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<tr>
<td>$C_{ax}$</td>
</tr>
<tr>
<td>$p/C_{ax}$</td>
</tr>
<tr>
<td>$S/C_{ax}$</td>
</tr>
<tr>
<td>Inlet Re</td>
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<tr>
<td>Exit Re</td>
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</table>

A non-axisymmetric endwall contour geometry was designed for the Pack-B blade profile by Praisner et al. (2007). Figure 6.3c shows a qualitative representation of the elevation of the contour design. The design incorporates a hill on the pressure side and valley on the suction side in the upstream half of the passage to reduce cross-passage pressure gradients (Praisner et al., 2007). The downstream half of the passage contains a ridge in orange and a valley in blue to hinder the development of the passage vortex and reduce exit flow angle deviation. Both the top and the bottom of the contoured endwall follow this design, as the contoured endwall has a constant thickness equivalent to the flat endwall thickness, $t = 2.9D$.

The film coolant supply is extracted from the coolant section by a blower on the top of the wind tunnel as depicted in Figure 6.1a. The coolant flow is chilled by two heat exchangers, one in the top section of the wind tunnel and an external heat exchanger downstream of the blower in the coolant line. Next the coolant passes through a laminar flow element measuring the coolant mass flowrate. The coolant line feeds a plenum below the endwall where the coolant back pressure and temperature are measured. As the coolant enters the plenum, it encounters a splash plate to reduce the incoming velocity as shown in Figure 6.1c. The coolant temperature, $T_{c, in}$, is measured within the plenum by two thermocouples $8.7D$ below the impingement plate. With the coolant heat exchangers and the mainstream heaters, the facility achieved a coolant to mainstream density ratio of about 1.15.
Figure 6.3 Comparison of oil flow visualization of endwall streaklines (Lynch et al., 2011b) with film cooling hole inlet and outlet locations for the (a) flat and (b) contoured endwalls, and (c) qualitative representation of the contoured endwall height.

The amount of film cooling is quantified by the average blowing ratio or mass flux ratio of the coolant to the mainstream for each film cooling hole, $M_{avg}$. The local blowing ratio is calculated by scaling the total measured mass flowrate by the percent flow for each film cooling hole. The amount of flow through each hole varied based on the local exit pressure ratio. To obtain the local exit pressure at each hole location, pressure taps are installed in Passage 6, which does not have film cooling holes. The total mass flowrate is adjusted until the desired blowing ratio is achieved. The uncertainty is coolant flowrate is estimated to be ±3% for a 95% confidence interval using sequential perturbation (Moffat, 1988).

The cooling configuration included internal impingement cooling and film cooling as previously mentioned. The diameter of both the impingement holes and the film cooling holes is $D = 4.4$ mm. The impingement plate contains 28 staggered holes with a spacing of 4.65D in both the axial and pitchwise directions. The distance between the impingement plate and the bottom of the endwall, $H$, for the flat endwall is 2.9D. For the contoured endwall, the, $H$, varies between 0.6D to 3.4D across the passage, but matches the flat endwall $H$ at the leading edge of the passage where the contour height is level. The impingement jets supply cooling to ten angled film cooling holes which are staggered between the impingement jets. The locations of the holes in the impingement plate relative to the film cooling holes can be found in Mensch & Thole.
The layout of film cooling holes for the flat and contoured endwalls is given in Figure 6.3a and b.

The location of the film cooling holes for both the flat and contoured endwalls is carefully designed according to specific parameters. The film cooling holes are angled at 30° relative to the local surface at the film cooling hole exits. The film cooling holes are oriented in the x-y plane to align with the local endwall flow direction at the hole exit. The local flow direction is obtained from endwall streaklines from oil flow visualization (Lynch et al., 2011b) shown in Figure 6.3a and b. Although the local endwall flow direction and local surface angle are not the same between the flat and contoured endwalls, the exit location of the film cooling holes is maintained between the designs. The length and orientation of the film cooling holes changes between the flat and contoured endwalls as seen in Figure 6.3a and b. The length of the film cooling holes, L, also varies between 4.2D and 8.0D for the contoured endwall, while all of the flat endwall film cooling holes have L = 5.8D. The size and location of the inlets to the film cooling holes are also different between the flat and contoured endwalls.

The flowfield near the trailing edge of the passage suction side is measured using time resolved particle image velocimetry (PIV). The measurement planes labeled A, B and C are shown in Figure 6.4. The passage exit plane, C, lines up with the trailing edge of the passage at a constant axial location of 1.08C_{ax} as measured from the leading edge of the blade. The streamwise planes, A and B, are parallel to the exit flow direction of the passage, as determined by the angle of the blade trailing edge given in Table 6.1. Planes A and B are located distances of 5D and 3D, respectively, from the centerline of the blade trailing edge. The streamwise planes (A and B) are at an angle of 30° from the passage exit plane (C). Plane C crosses Planes A and B near the blade trailing edge as shown in Figure 6.4a. Figure 6.4b depicts the passage vortex within Plane C with a computational prediction of turbulent kinetic energy from Mensch et al. (2015a). The approximate location of the intersections of Planes A and B within Plane C are also shown in Figure 6.4b.
Figure 6.4 Planes measured with PIV shown (a) for the flat endwall from above, (b) for the contoured endwall from above, and (c) looking upstream from of Plane C overlaid with flat endwall CFD tke contours for $M_{av} = 2$ (Mensch & Thole, 2015a).

The PIV measurement setup uses a high speed CMOS camera to capture the motion of particles in the flow illuminated by a dual head Nd:Nd:YLF laser. The flow is seeded with small particles with atomized liquid di-ethyl-hexyl-sebecat (DEHS). The particles follow the flow since the average particle Stokes number is much less than one. The Nd:Nd:YLF laser beam passes through a cylindrical lens creating a laser sheet which is directed vertically down to the endwall surface. The high speed camera is oriented normal to each measurement plane to obtain a two-dimensional velocity field within each plane. The laser pulses and the camera images are synchronized with a high speed controller. The camera takes 6000 image pairs at a frequency of 2 kHz for the passage exit plane (C) and 3000 image pairs at a frequency of 1 kHz for the streamwise planes (A and B). The total data acquisition time for all three planes is 3 s, which is sufficient to obtain results that are statistically converged. The time delay between laser pulses is chosen based on an estimated bulk movement of 10 pixels, about 1/6 the initial interrogation window size. Plane C uses a camera window size of 1024 x 512, and the image resolution is 16 pixels/D. Planes A and B use a camera window size of 1024 x 1024, and the image resolution is 22 pixels/D.

The images are processed and vectors calculated using LaVision software ($DaVis 8.1.4$, 2012). The raw images are pre-processed to increase the contrast of the particles by subtracting...
the sliding minimum of the surrounding three images. Velocity vectors are calculated with a cross-correlation between the image pairs over an interrogation window that decreases in size over four passes. The vectors are post-processed with two passes of a median filter using universal outlier detection for a 3 x 3 pixel region (DaVis 8.1.4, 2012) to remove spurious vectors. Empty spaces surrounded by at least two calculated vectors are re-filled through interpolation.

The uncertainty in instantaneous velocity measurements is estimated based on the contribution of the instantaneous displacement gradients for each velocity component (Raffel et al., 2007). The maximum displacement gradients are 0.2 pixels/pixel in both directions for Planes A and B, resulting in an estimated uncertainty of the normalized velocities, $U/U_{\infty, \text{in}}$, of ±0.1, for a final interrogation window of 24 x 24 pixels (Raffel et al., 2007). For Plane C, the maximum displacement gradients are 0.2 pixels/pixel in the y-direction and 0.28 pixels/pixel in the z-direction, resulting in an estimated uncertainty of ±0.15 for $U_y/U_{\infty, \text{in}}$ and ±0.2 for $U_z/U_{\infty, \text{in}}$.

6.3 Results and Discussion

Figure 6.5 shows predicted average streamtraces of near endwall flow being swept into the passage vortex as obtained from RANS CFD simulations for a blowing ratio of two (Mensch & Thole, 2015a). The streamtraces are shown in two views for the flat endwall in Figure 6.5a and b and for the contoured endwall in Figure 6.5c and d. The streamwise plane closest to the blade, Plane B, is designated by the black line in Figure 6.5a and c, and by the light gray surface in Figure 6.5b and d. The predicted streamtraces allow visualization of the passage vortex and its evolution in the downstream half of the passage into the wake. The top views show that the passage vortex closely follows the profile of the blade suction side in the downstream half of the passage. Downstream of the blade trailing edge, the streamtraces are diverted to the right of the Plane B line, presumably due to interactions with the wake flow. Figure 6.5b and d show the upward trajectory of the passage vortex, which is maintained after the trailing edge of the blade. The contoured endwall streamtraces stay closer to the blade suction side compared to the flat endwall streamtraces which are more spread out and cross Plane B more often. Additional differences between the flat and contoured endwalls can be seen in the side views of Figure 6.5b and d. The flat endwall case shows stronger vortical motion of the streamtraces compared to the contoured endwall. The oscillations for the contoured endwall streamtraces are also smaller than
the flat endwall streamtraces. Although endwall contouring does not remove the passage vortex entirely, the streamtraces show that the vortex is weaker and more compact for the contoured endwall. Figure 6.5 provides context to understand the flow measurements in the plane at the exit of the passage and in the two streamwise planes presented below.

Figure 6.5 Flow streamtraces of near endwall flow swept into the passage vortex, from CFD predictions for $M_{\text{avg}} = 2$ (Mensch & Thole, 2015a) for the (a) flat endwall top view, (b) flat endwall side view, (c) contoured endwall top view, (b) contoured endwall side view.

6.3.1 Measurements at the Passage Exit Plane

Because the exit flow angle of the passage is 60°, the exit plane (Plane C) has a significant velocity component in the pitchwise (y) direction due to the inviscid mainstream flow. Therefore, the velocities measured in Plane C, $U_{\text{meas}}$, are decomposed into the bulk inviscid flow velocity, $U_{\text{inviscid}}$, plus the secondary flow velocity, $U_{\text{sec}}$. $U_{\text{inviscid}}$ is an estimate of the time-averaged velocity of the inviscid flow above the secondary flows, obtained by averaging the measured velocities across an area at the top of the measurement plane where the velocities
do not vary in the z direction. The width of the averaging region for \( U_{\text{inviscid}} \) is from -0.48p to 0.31p, which was 80% of the pitch. The height of the region is from 0.24S to 0.27S, 3% of the span. The y-components of the normalized area-averaged inviscid velocity, \( U_{\text{inviscid}}/U_{\infty,\text{in}} \), are about 1.3, and the z-components of the area-averaged inviscid velocity, \( U_{\text{inviscid}}/U_{\infty,\text{in}} \), are about 0.07, near zero as expected. The components of \( U_{\text{inviscid}} \) for each case are subtracted from the measured velocity vectors for that case, resulting in a secondary velocity, \( U_{\text{sec}} \), using Equation (6.1). The secondary velocity vectors are not the actual in-plane velocities of the flow, but represent a means to visualize the secondary flows.

\[
U_{\text{sec}} = U_{\text{meas}} - U_{\text{inviscid}}
\] (6.1)

The secondary velocity vectors calculated for the flat endwall are shown in Figure 6.6, which is viewed in the upstream direction, for no film cooling, and \( M_{\text{avg}} = 1 \) and 2. The y/p = 0 location marks the line of sight of the trailing edge of the blade within Plane C. Dotted lines indicate where the parallel Planes A and B intersect Plane C. Figure 6.6 also displays contour levels of the time-averaged vorticity fields normalized by \( U_{\infty,\text{in}}/S \). The vorticity fields are calculated based on the original measured velocities prior to the previously described decomposition. As the curl of the local two-dimensional velocity, the vorticity contours quantify the severity of velocity gradients associated with vortex motion. Positive vorticity is associated with counterclockwise rotation, and negative vorticity is associated with clockwise rotation.

As shown in Figure 6.6, the vertical line of negative vorticity at y/p = 0.1 to 0.15 represents the wake of the flow passing through the plane. To the left of the wake, the blue regions indicate the large passage vortex that dominates the secondary flow in the passage. The passage vortex persists after the passage as Plane C is beyond the trailing edge at x/C_{ax} = 1.08. It is apparent by the blue vorticity levels extending to the left edge of the measurement plane, that the passage vortex has an effect on the flow across at least half of the passage pitch. From y/p = -0.45 to the Plane B dotted line, the vectors close to the wall are directed down and toward the passage suction side. Above the passage vortex is a region of negative vorticity, which is most pronounced in Figure 6.6a without film cooling. Here, there is a smaller and weaker counter rotating vortex from what remains of the suction side horseshoe vortex (Goldstein & Spores, 1988). In the upper portions of the measurement plane to the left of Plane A, the relative velocity vectors are almost zero and the vorticity is not uniformly positive or negative. This region is part of the mainstream inviscid flow.
Figure 6.6 Contours of time-averaged vorticity overlaid with the secondary velocity vectors measured in Plane C for the flat endwall with (a) no film cooling, (b) $M_{\text{avg}} = 1.0$, and (c) $M_{\text{avg}} = 2.0$. 
As film cooling is added and the blowing ratio increases for the flat endwall in Figure 6.6, the peak vorticity in the passage vortex increases and moves away from the wake in the negative pitch direction. The center of rotation of the secondary velocity vectors also moves away from the endwall. The size of the relative velocity vectors also grows as blowing ratio increases. Although film cooling strengthens and increases the size of the passage vortex, the counter rotating vortex above the passage vortex is weakened for the cases with film cooling. The yellow region of negative vorticity shrinks as passage vortex becomes larger.

The secondary velocity vectors and the vorticity contours in Plane C are shown for the contoured endwall in Figure 6.7 for no film cooling, $M_{\text{avg}} = 1$ and $M_{\text{avg}} = 2$. The vorticity levels in the wake are similar between the flat and contoured endwalls. However, the contoured endwall passage vortex has greater peak vorticity than the peak vorticity in the corresponding flat endwall cases. As expected, the contoured endwall has a smaller area of positive vorticity and smaller secondary velocity vectors on the left and top of the passage vortex compared to the flat endwall. Lynch (2011) found the contoured endwall passage vortex is farther from the wake and closer to the endwall than the flat endwall passage vortex. This trend also is observed in the current measurements for the locations of peak vorticity, which is most visible by comparing the Figure 6.6c and Figure 6.7c for $M_{\text{avg}} = 2$.

Another difference between the flat and contoured endwalls is the extension of high vorticity below $z/S = 0.05$. Elevated vorticity below $z/S = 0.05$ is only seen for the flat endwall without film cooling in Figure 6.6a. In Figure 6.7, the contoured endwall vorticity and secondary vectors near the endwall are similar between the different coolant blowing ratios and are larger than for the flat endwall for all cases. Higher streamwise vorticity and secondary kinetic energy close to the endwall were also measured at the passage exit by Knezevici et al. (2010). In addition, Lynch (2011) measured higher secondary velocities and secondary kinetic energy close to the endwall for the contoured endwall just beyond the passage exit. Part of the reason for the higher vorticity and secondary velocities for the contoured endwall is because the passage vortex sits closer to the endwall with contouring. Additionally, Knezevici et al. (2010) showed that for the contoured endwall near the trailing edge of the passage, the pitchwise crossflow near the endwall suddenly increases as the contouring blends back to a flat surface. Upstream and within the passage, the endwall flow is less skewed for the contoured endwall compared to the flat endwall, but at the exit plane, there is a sharp acceleration and turning of the
contoured endwall flow toward the suction side of the passage (Knezevici et al., 2010). The local increase in near endwall flow energizes passage vortex in the exit plane and contributes to the higher vorticity measured near the endwall in Figure 6.7.

The counter rotating vortex above the passage vortex is stronger and larger for the contoured endwall in Figure 6.7 compared to the flat endwall in Figure 6.6. Lynch (2011) also found that the suction side horseshoe vortex remained stronger with contouring through velocity measurements near the blade endwall junction in the middle of the passage. Oil flow visualizations from Knezevici et al. (2010) and Lynch (2011) showed that the separation line of the suction side leg of the horseshoe vortex originates farther upstream from the suction side of the blade with contouring. This moves the suction side horseshoe vortex farther from the blade, resulting in a more coherent counter rotating vortex downstream (Lynch, 2011). Based on the vorticity measurements with film cooling and with endwall contouring, it can be seen that the size of the counter rotating vortex is inversely tied to the size of the passage vortex, with a larger passage vortex resulting in a smaller counter rotating vortex.

Similar to the flat endwall, the contoured endwall peak vorticity and size of the passage vortex increase with film cooling in Figure 6.7b and c. Also the vorticity and size of the counter rotating vortex decrease with film cooling. The peak in vorticity for the flat endwall around the Plane B dotted line in Figure 6.6a disappears as film cooling is added, but this vorticity peak for the contoured endwall in Figure 6.7a remains as film cooling is added. Overall, the vorticity fields are more changed by endwall contouring than by the film cooling blowing ratio.

The Plane C contours of turbulent kinetic energy (tke) normalized by $U_{in}^2$ are shown in Figure 6.8 and Figure 6.9 for the flat and contoured endwalls respectively. For the two-dimensional velocity field measured, tke is calculated by $\frac{1}{4}(u_1^2 + u_2^2)$, which assumes the root mean square of the out of plane velocity fluctuations is an average of the two measured components of velocity fluctuations (DaVis 8.1.4, 2012). The contours of tke indicate the variation in the measured velocity vectors and the unsteadiness in flow structures. The tke levels are lowest in the inviscid mainstream parts of the flow. The highest tke levels are measured at the top of the passage vortex, at the bottom of the counter rotating vortex, and within the wake. The secondary velocity vectors are overlaid on Figure 6.8 and Figure 6.9 to show the location of the passage vortex.
Figure 6.7 Contours of time-averaged vorticity overlaid with the secondary velocity vectors measured in Plane C for the contoured endwall with (a) no film cooling, (b) $M_{avg} = 1.0$, and (c) $M_{avg} = 2.0$. 
Figure 6.8 Contours of turbulent kinetic energy overlaid with the secondary velocity vectors measured in Plane C for the flat endwall with (a) no film cooling, (b) $M_{\text{avg}} = 1.0$, and (c) $M_{\text{avg}} = 2.0$. 
Figure 6.9 Contours of turbulent kinetic energy overlaid with the secondary velocity vectors measured in Plane C for the contoured endwall with (a) no film cooling, (b) $M_{\text{avg}} = 1.0$, and (c) $M_{\text{avg}} = 2.0$. 
The highest peak in tke without film cooling occurs at $z/S = 0.15$ for the flat endwall in Figure 6.8a and at $z/S = 0.13$ for the contoured endwall in Figure 6.9a. These locations are just below the corresponding vorticity peaks in Figure 6.6a and Figure 6.7a. At these locations, the secondary flow changes direction from having a significant upward component to having a significant negative pitchwise (-y) component. The contoured endwall tke peak is closer to the endwall than the tke peak for the flat endwall consistent with a smaller contoured endwall passage vortex. The levels of tke are about the same for the flat and contoured endwalls without film cooling, but the elevated tke region (> 0.04) for the flat endwall is larger than that for the contoured endwall. Lynch (2011) also measured a smaller region of tke greater than 0.04 for the contoured endwall in a similar measurement plane. An additional difference between the flat and contoured tke levels is that the contoured endwall in Figure 6.9a has two bands of high tke for the passage vortex and the counter rotating vortex, while the flat endwall has one distinct region because the counter rotating vortex is stronger for the contoured endwall measurements.

Film cooling increases the tke associated with the passage vortex for both the flat and contoured endwalls. With film cooling, a second peak in tke forms near the center of the passage vortex and grows with blowing ratio. The peaks in tke for $M_{avg} = 2$ in Figure 6.8c and Figure 6.9c line up with the peaks in vorticity in Figure 6.6c and Figure 6.7c. Film cooling has a significant effect on the tke levels for the flat endwall doubling the peak tke between no film cooling and $M_{avg} = 2$ in Figure 6.8c. The kinetic energy of the film cooling jets adds to the kinetic energy and unsteadiness to the passage vortex. Although film cooling also has a significant effect on tke for the contoured endwall, the effect is much less than for the flat endwall because the contoured endwall is able to suppress the size of the passage vortex. The contoured endwall also has better film cooling performance than the flat endwall (Mensch & Thole, 2015b), suggesting that the film cooling jets remain closer to the endwall with contouring.

6.3.2 Measurements along the Streamwise Direction

The measured time-averaged velocity vectors for the streamwise planes parallel to the trailing edge (Planes A and B) are shown for the flat endwall in Figure 6.10 for the cases of no film cooling, $M_{avg} = 1$ and $M_{avg} = 2$. The background of Figure 6.10 is colored by the average measured spanwise ($z$) velocity component, $U_z$. The primary flow direction is within the measurement planes along the s coordinate as defined in Figure 6.4a. Therefore, $U_z$, provides an
indication of the secondary flow associated with the passage vortex. The intersection with Plane C is shown by a dotted black line, and the location of $s/C_{ax} = 0$ is aligned with the line of sight of the blade trailing edge.

Figure 6.10 Contours of time-averaged spanwise ($z$) velocity overlaid with time-averaged velocity vectors in Planes A and B for the flat endwall for (a-b) no film cooling, (c-d) $M_{avg} = 1.0$, and (e-f) $M_{avg} = 2.0$. 
The spanwise velocities are positive everywhere in both measurement planes, as both planes are capturing the upward motion in the portion of the passage vortex closer to the suction side of the blade. The highest $U_z$ contours form a band that moves away from the endwall as $s/C_{ax}$ increases, indicating the size of the passage vortex is growing as it travels downstream. Comparing the $U_z$ contours in Planes A and B, it is apparent that the peak $U_z$ in Plane B is significantly higher than in Plane A, despite being only separated by 0.045$p$ in the pitch ($y$) direction. Plane A has lower spanwise velocities because Plane A is closer to the center of the passage vortex than Plane B. As seen by the streamtraces in Figure 6.5, the trajectory of the passage vortex is not oriented exactly with Planes A and B. Because the trajectory of the passage vortex follows the profile of the blade, the distance between the blade and the measurement plane changes the location within the passage vortex.

The spanwise velocities are highest in the upstream half of Plane B because Plane B is closest to the blade at $s/C_{ax} = -0.4$, as seen in Figure 6.4a. Downstream at $s/C_{ax} = 0$, the distance from the blade is greater, and $U_z$ decreases because Plane B is shifted toward to the center of the passage vortex. The intersection of Plane A and Plane C is closer to the center of the passage vortex, as can be seen in the Plane C vorticity contours in Figure 6.6. After $s/C_{ax} = 0$, $U_z$ increases again in both Planes A and B due to the turning of the passage vortex caused by interactions with the wake. The variation in $U_z$ in Planes A and B demonstrate the steep gradients in secondary velocities within the passage vortex.

With film cooling, in Figure 6.10c-d, the peak spanwise velocities increase compared to the cases without film cooling for both streamwise planes. The largest increase in $U_z$ with film cooling occurs in the upstream half of the measurement planes, where the planes are closest to the blade. For $M_{avg} = 1$ and 2, the cylindrical hole film cooling jets are detached from the flat endwall and introduce a significant upward velocity component to the passage vortex. This effect is strongest in the upstream half of the measurement planes closer to the film cooling jets, but film cooling also increases the peak $U_z$ in the wake around $s/C_{ax} = 0.2$. Similar to the increasing tke of the passage vortex with film cooling, the increasing secondary velocity in the wake indicates the film cooling kinetic energy helps to energize the passage vortex.

The time-averaged velocity vectors and contours of $U_z/U_{z,in}$ for the contoured endwall are shown in Figure 6.11 for the streamwise planes. The $s/C_{ax} = 0$ location marks the line of sight of the blade trailing edge, and the intersection with Plane C is indicated by a dotted black
In agreement with the measurements in Plane C, the passage vortex, indicated by the elevated \( U_z \) contours, is closer to the contoured endwall than the flat endwall. Without film cooling in Figure 6.11a and b, the contoured endwall \( U_z \) is zero above \( z/S = 0.24 \), which shows that the passage vortex does not affect the flow above this height. In comparison to the flat endwall in Figure 6.10, the passage vortex has an influence that extends past \( z/S = 0.34 \), where there is still a small \( U_z \) component. Without film cooling, the contoured endwall \( U_z \) measurements in Figure 6.11a and b are much less than the corresponding flat endwall \( U_z \) measurements. The \( U_z \) contours indicate a smaller passage vortex for the contoured endwall that is centered closer to the blade and to the endwall, as previously discussed. The contoured endwall \( U_z \) behaves similarly to the flat endwall when film cooling is added in Figure 6.11c-f. Film cooling increases \( U_z \), with the most significant increases at the far upstream portions of Plane B. Film cooling causes the influence of the passage vortex to expand farther into the mainstream, although an unaffected region remains even for \( M_{avg} = 2 \) in Figure 6.11e and f.

The contours of normalized tke and in-plane velocity vectors measured in Planes A and B are shown in Figure 6.12 for the flat endwall and Figure 6.13 for the contoured endwall. As before, the tke is calculated based on the two measured components of velocity fluctuations, assuming the out of plane component is an average of the other two components. Like Plane C, the tke levels are lowest in the mainstream flow and elevated in the passage vortex. The tke levels at the intersections with Plane C are generally similar to those measured within Plane C in Figure 6.8 and Figure 6.9. One exception is for the flat endwall at \( M_{avg} = 2 \) in Plane A (Figure 6.12e). The tke measured at the Plane C intersection in Figure 6.12e is less than the tke measured at the Plane A intersection in Figure 6.8c. The difference arises because Plane C uses different velocity fluctuations than Planes A and B to calculate the tke. Planes A, B and C are all calculated using the spanwise \( (U_z) \) velocity fluctuations. While the pitchwise velocity fluctuations are used for the tke calculations in Plane C, Planes A and B instead use the streamwise velocity fluctuations. Planes A and B do not consider unsteadiness in the passage vortex in the direction perpendicular to the blade surface. Therefore, the streamwise fluctuations of the passage vortex are less than the pitchwise fluctuations for the flat endwall at \( M_{avg} = 2 \) in Plane A.
Figure 6.11 Contours of time-averaged spanwise (z) velocity overlaid with time-averaged velocity vectors in Planes A and B for the contoured endwall for (a-b) no film cooling, (c-d) $M_{\text{avg}} = 1.0$, and (e-f) $M_{\text{avg}} = 2.0$. 
Figure 6.12 Contours of turbulent kinetic energy overlaid with time-averaged velocity vectors in Planes A and B for the flat endwall for (a-b) no film cooling, (c-d) $M_{\text{avg}} = 1.0$, and (e-f) $M_{\text{avg}} = 2.0$. 
Figure 6.13 Contours of turbulent kinetic energy overlaid with time-averaged velocity vectors in Planes A and B for the contoured endwall for (a-b) no film cooling, (c-d) $M_{avg} = 1.0$, and (e-f) $M_{avg} = 2.0$.

The tke levels without film cooling in Figure 6.12 and Figure 6.13 have similar trends to the $U_z$ contour levels, increasing as the distance from the blade decreases and increasing in the wake. Without film cooling, the tke in Plane B is greater than the tke in Plane A for both Figure 6.12a and Figure 6.13a. With film cooling, the tke increases in Plane A which is closer to the
center of the passage vortex. The tke in the center of the passage vortex also increased with film cooling for the measurements in Plane C. Figure 6.12 shows very large levels of tke for the flat endwall with film cooling in the wake. The high levels of tke are generated by large velocity variations measured intermittently in this region of the flow. It is believed these fluctuations are caused by larger out of plane velocities due to the passage vortex interacting with the wake.

6.3.3 Unsteadiness of the Passage Vortex

The unsteadiness of the flat and contoured endwall passage vortex without film cooling is compared in Figure 6.14 through histogram contours of velocity magnitude along the $s/C_{xx} = 0$ profile of Plane B. The vertical profile of average velocity is given by the solid black line in Figure 6.14. The contour levels surrounding the average profile show the prevalence of individual instantaneous velocities measured during the total measurement period. The bin size for the $|U|/U_{\infty,\text{in}}$ histogram levels is 0.02. The average velocity profiles have a negative peak at the top of the passage vortex. The velocity magnitude is lowest here because the flow has a larger out of plane component (Mensch & Thole, 2015b). The negative peak occurs at $z/S = 0.16$ for the flat endwall, and lower at $z/S = 0.12$ for the contoured endwall. Because the contoured endwall vortex is smaller, the average velocity gradient between the mainstream velocity and the negative peak is steeper than the gradient measured for flat endwall. Additionally the contoured endwall average velocity at the bottom of the profile nearly reaches the mainstream velocity at the top, while the flat endwall $|U|/U_{\infty,\text{in}}$ only reaches 1.4 at the bottom of the profile.

Figure 6.14 shows that the variation in the velocity magnitude is the greatest at the top of the passage vortex where the histogram is widest. The histogram is narrower at the bottom and top of the profile, in the mainstream. The widths of the histograms at the top of the passage vortex are similar between the flat and contoured endwalls, indicating the variation in the passage vortex streamwise and spanwise velocities are similar between the flat and contoured endwalls. This result is in agreement with the similar tke levels measured for the flat and contoured endwalls in Planes A and B.
Figure 6.14 Histogram contours of velocity magnitude in Plane B at the trailing edge with the time-averaged velocity profile in black for (a) the flat endwall and (b) the contoured endwall.

6.4 Conclusions

The separate and combined effects of endwall contouring and film cooling on the passage vortex flowfield were studied with time resolved two-dimensional PIV in three measurement planes near the trailing edge of the blade. The calculated average secondary velocity vectors and contours of vorticity in the passage exit plane confirmed the location of the passage vortex and a weaker counter rotating vortex situated above the passage vortex. The passage exit plane turbulent kinetic energy contours showed peak turbulent kinetic energy associated with the blade wake, the top of the passage vortex, and the center of the passage vortex with film cooling. In the two streamwise planes, the two-dimensional average velocity vectors and spanwise velocity contours showed upward velocities in the edge of the passage vortex closest to the blade suction side. The spanwise velocity contours were sensitive to the distance from the blade, indicating that the passage vortex has significant velocity gradients in the pitch direction. Turbulent kinetic energy in the streamwise planes was also elevated in the passage vortex. The spanwise velocities and turbulent kinetic energy increased as the distance between the measurement plane and the blade was reduced. The spanwise velocities and turbulent kinetic energy also increased after the flow exited the passage, where the passage vortex interacted with the wake.
Film cooling was a significant factor for the passage vortex size and strength in the exit plane measurements. As film cooling was added, the vorticity, turbulent kinetic energy, and secondary velocities of the passage vortex all increased with blowing ratio. Film cooling increased the size of the passage vortex, moving the center farther from the endwall and farther from the blade. As a result, the counter rotating vortex was diminished as film cooling increased. The spanwise velocities in the streamwise planes also increased with film cooling, and extended the influence of the passage vortex farther into the mainstream. Film cooling blowing ratios of 1 and 2 produced jets that were detached from the endwall surface. As the jets were swept into the passage vortex, the kinetic energy was transferred to the vortex, increasing the velocities and turbulent kinetic energy downstream. The increases in turbulent kinetic energy with blowing ratio were more significant in the exit plane compared to the increases in the streamwise planes, indicating that film cooling had a greater impact on the pitchwise velocity fluctuations compared to the streamwise velocity fluctuations in the passage vortex.

Endwall contouring generally had the opposite effect of film cooling on the passage vortex. Contouring reduced the size and turbulent kinetic energy of the passage vortex measured in the exit plane. The spanwise velocities measured in the streamwise planes were also much lower for the contoured endwall for the same film cooling conditions. The vortex was more compact for the contoured endwall, which resulted in a higher vorticity for the passage vortex as well as a more pronounced counter rotating vortex. The turbulent kinetic energy and velocity magnitude histograms indicated that the passage vortex streamwise and spanwise velocity fluctuations were similar in magnitude for the flat and contoured endwall. One exception was fewer pitchwise fluctuations for the contoured endwall passage vortex, as evidenced by the exit plane turbulent kinetic energy measurements. With film cooling, the contoured endwall spanwise velocities and exit plane turbulent kinetic energy were significantly increased, showing that the effects of film cooling and endwall contouring were of similar significance to the turbulent kinetic energy and unsteadiness of the passage vortex near the passage exit.
CHAPTER 7. CONJUGATE HEAT TRANSFER MEASUREMENTS AND PREDICTIONS OF A BLADE ENDWALL WITH A THERMAL BARRIER COATING**

Abstract

Multiple thermal protection techniques, including thermal barrier coatings (TBCs), internal cooling and external cooling, are employed for gas turbine components to reduce metal temperatures and extend component life. Understanding the interaction of these cooling methods, in particular, provides valuable information for the design stage. The current study builds upon a conjugate heat transfer model of a blade endwall to examine the impact of a TBC on the cooling performance. The experimental data with and without TBC are compared to results from conjugate CFD simulations. The cases considered include internal impingement jet cooling and film cooling at different blowing ratios with and without a TBC. Experimental and computational results indicate the TBC has a profound effect, reducing scaled wall temperatures for all cases. The TBC effect is shown to be more significant than the effect of increasing blowing ratio. The computational results, which agree fairly well to the experimental results, are used to explain why the improvement with TBC increases with blowing ratio. Additionally, the computational results reveal significant temperature gradients within the endwall, and information on the flow behavior within the impingement channel.

7.1 Introduction

The ongoing aim to increase power output in gas turbines results in increasing gas temperatures at the turbine inlet. Consequently, turbine parts must withstand thermal conditions that exceed the material allowable temperatures. For that reason cooling technologies are required to maintain part life. Coolant is extracted from the compressor and directed into turbine components. First, the coolant removes heat from internal surfaces, and then the coolant is ejected through film cooling holes providing a coolant layer on the outer surface. In addition,

insulating, high temperature ceramic coatings, known as thermal barrier coatings (TBCs), are often applied on external surfaces to further protect the parts that are exposed to the highest thermal loads. A critical need for engineers is the ability to accurately predict the overall thermal performance of configurations, which include a combination of the available cooling technologies.

The objective of this study is to evaluate the improvement in thermal performance when a TBC is applied to a typical gas turbine endwall configuration for different coolant flowrates. It is established that a TBC can provide dramatic reduction in metal temperatures and thus has been applied in many designs (Padture et al., 2002). Both experimental measurements and computational predictions of a scaled endwall model are used to quantify the benefit of a TBC for a fully cooled endwall. A second objective is to assess the predictions with experimental results. Once validated, the simulations provide additional insight into the endwall temperatures and heat transfer that cannot be easily measured. This study builds upon the experimental work by Mensch and Thole (2014) on a conjugate endwall with impingement and film cooling, to examine the thermal effects of a TBC.

7.2 Relevant Literature

There has been considerable emphasis recently on conjugate heat transfer and determining the scaled metal temperature, or overall effectiveness, of gas turbine surfaces both computationally and experimentally. Albert et al. (2004) showed that it is important to model the appropriate dimensionless parameters to obtain relevant measurements in these types of studies. The critical parameters are the Biot number, Bi; the ratio of external to internal heat transfer coefficients, $h_\infty/h_i$; the Reynolds number, Re; and the scaled geometry. The first studies to provide measurements of conjugate heat transfer on a conducting C3X vane were by Hylton et al. (1983, 1988) and Turner et al. (1985). The engine relevance of these data is limited because matched Bi and $h_\infty/h_i$ were not confirmed; however, these initial studies improved understanding of conjugate heat transfer in a vane and provided an important data set for comparison to computational work.

Endwall heat transfer can behave differently compared to a vane surface because of the presence of strong external secondary flows, namely the horseshoe and passage vortices. The passage vortices affect the endwall by skewing the direction of endwall flow and locally
increasing the external heat transfer coefficient as shown by Kang and Thole (2000). The first experimental study for a conjugate endwall exposed to passage flows was completed by Mensch and Thole (2014). Although the endwall was cooled with both impingement and film cooling, the impingement had a greater influence on the scaled wall temperature. Furthermore, significant temperature gradients were observed due to the use of engine matched Bi, which is the same value as the current study. Lynch et al. (2011a) compared experimental and computational results for an endwall with the same blade geometry for an adiabatic wall boundary condition, as well as for a constant wall heat flux boundary condition to measure the external heat transfer coefficients.

The thermal effect of TBC on a conjugate surface was considered by Maikell et al. (2011) in an experimental leading edge model with a Bi of 1-2.1, matched to an engine leading edge. The TBC case had cooler leading edge wall temperatures but hotter external TBC temperatures due to the insulating effect of the TBC. Davidson et al. (2014b) investigated the improvement in wall temperature when TBC was added to a matched Bi vane with Bi of 0.3-1.1. The cooling performance increased significantly with TBC to the point that increasing blowing ratio did not provide a further reduction in the vane wall temperature. A computational study of TBC applied to a conjugate wall with film cooling was completed by Na et al. (2007). The flat plate geometry contained a single 30° angled film cooling hole, and accounted for partial blockage of the film cooling hole by the TBC. Without TBC, the wall temperature was relatively uniform across the surface, indicating a low Bi, although the Bi was not stated. With TBC, decreased wall temperatures were reported.

Comparisons between experimental and computational conjugate heat transfer results are of increased interest since the experimental validation of the simulations provides an important designer tool. A study by Panda and Prasad (2012) involved experimental measurements and computational simulations for a flat plate with internal impingement and film cooling. The simulations showed excellent agreement with the experimental results along the centerline. Plates with different thermal conductivities (plastic and stainless steel) were considered, but the Bi of these configurations were not reported. Dobrowolski et al. (2009) performed conjugate simulations corresponding to the leading edge experiments without TBC by Maikell et al. (2011). The effect of impingement was applied by setting the internal wall temperature distribution to the temperatures measured in the experiments. Using the realizable k-ε turbulence
model, the simulations under-predicted the separation of the film cooling jets compared to the experiments. Similar results were found for the same leading edge model with shaped film cooling holes and impingement (Mouzon et al., 2005; Ravelli et al., 2010).

The conjugate heat transfer through the suction side of a vane model (engine matched Bi of 0.4-1.6) was simulated by Ledezma et al. (2011) using the standard k-ω turbulence model, and results were compared to experiments by Dees et al. (2013). Differences from the experiments were attributed to poor prediction of film cooling jet separation, the assumption of isotropic turbulence, and unsteady effects. The suction side of a vane model with a similar Bi and impingement cooling was simulated by Dyson et al. (2012) and compared to experiments in Williams et al. (2014). Film cooling jet diffusion was under-predicted by the k-ω SST turbulence model in the simulations. The insufficient diffusion led to over-predicted cooling effectiveness for attached jets and under-predicted effectiveness for detached jets. Ni et al. (2011, 2013) simulated a fully film cooled vane and endwall geometry under flow conditions consistent with a dual spool engine with a pressure ratio of 40. The conjugate simulations were performed using the standard k-ω turbulence model. The predicted surface heat flux and temperature data agreed with the measurements within experimental uncertainties. The results showed significant temperature variations along the vane chord at 62% span.

Although there have been several studies comparing experimental and computational conjugate heat transfer results for different configurations, this study fills two gaps in the literature. There are no studies comparing conjugate experiments and simulations for an endwall, and no studies which consider the effect of TBC on an endwall. The current study provides an experimental and computational comparison of a conjugate endwall with and without a TBC.

7.3 Conjugate Endwall Model

The endwall model in both the experimental and computational studies, as depicted in Figure 7.1, incorporates the convective and solid conduction heat transfer associated with the following features: external film cooling, a thermal barrier coating (TBC), a conducting endwall, and internal impingement jet cooling.
To obtain scaled temperature data that is engine relevant, the appropriate geometric, thermal, and flow parameters are taken into account. The relevant non-dimensional parameters for a conducting wall with TBC are obtained from an expression derived for the non-dimensional outer endwall temperature (at the interface between the endwall and the TBC) as the overall effectiveness with TBC, $\phi_{TBC}$. The overall effectiveness is an important parameter to assess thermal performance because the airfoil life is dependent upon this temperature. The equation for $\phi_{TBC}$, given in Equation (7.1), is derived from a one-dimensional consideration of the heat transfer through the wall in Figure 7.1. In the case of no TBC, the equation reduces to Equation (7.2) for $\phi$ without TBC, previously reported in (Mensch & Thole, 2014; Williams et al., 2014).

\[
\phi_{TBC} = \frac{T_{\infty} - T_w}{T_{\infty} - T_{c,in}} = \frac{1 - \chi_\eta \eta}{1 + \frac{\frac{h_\infty}{h_i}}{Bi + \frac{h_\infty}{h_i}}} + \chi_\eta \eta \quad (7.1)
\]

\[
\phi = \frac{T_{\infty} - T_w}{T_{\infty} - T_{c,in}} = \frac{1 - \chi_\eta \eta}{1 + BI + \frac{h_\infty}{h_i}} + \chi_\eta \eta \quad (7.2)
\]

The dimensionless parameters revealed in Equations (7.1) and (7.2) include the endwall Bi, the $h_\infty/h_i$, and the ratio of TBC to endwall thermal resistances ($R_{TBC}/R_w$), defined in Equation (7.3).

\[
R_{TBC} / R_w = \frac{R_{TBC} k_w}{t_w} \quad (7.3)
\]
The overall effectiveness is also a function of the product of the coolant warming factor, \( \chi \eta \), and the adiabatic effectiveness, \( \eta \). Adiabatic effectiveness is commonly reported to indicate the effectiveness of external cooling techniques such as film cooling. Adiabatic effectiveness is assumed to be a function of geometry, the mainstream flow parameters, and film cooling flow parameters such as the film cooling blowing ratio, \( M \). In Equations (7.1) and (7.2), \( \eta \) represents the non-dimensional driving temperature for external convection in the presence of film cooling. The \( \chi \eta \), defined in Equation (7.4), is a correction factor for \( \eta \) that corrects for the warming of the coolant during impingement and flow through the film cooling holes, from \( T_{c,\text{in}} \) to \( T_{c,\text{exit}} \). As a result, the product, \( \chi \eta \eta \), and \( \phi \) have the same normalizing temperature difference (Williams et al., 2014).

\[
\chi \eta = \frac{T_{\infty} - T_{c,\text{exit}}}{T_{\infty} - T_{c,\text{in}}} \tag{7.4}
\]

Another quantity of interest when considering surfaces with TBC, is the scaled external temperature of the TBC, also known as the TBC effectiveness, \( \tau \), which is defined in Equation (7.5).

\[
\tau = \frac{T_{\infty} - T_{\text{TBC}}}{T_{\infty} - T_{c,\text{in}}} \tag{7.5}
\]

Since \( T_{\text{TBC}} \) is on the same heat transfer circuit used to derive Equation (7.1), it follows that \( \tau \) depends on the same parameters as \( \phi_{\text{TBC}} \).

The non-dimensional parameters matched to an engine for the endwall and TBC layer for both the experiments and simulations in this study are given in Table 7.1. The blowing ratio, \( M_{\text{avg}} \), which affects \( \chi \eta \eta \), defines the average ratio of coolant to mainstream mass flux. The endwall \( h_{\infty} \) distribution is obtained from measurements made by Lynch et al. (2011b). An engine matched Bi is achieved by making the endwall from Corian®️, a DuPont material. To estimate \( h_i \) for each blowing ratio, a Nu correlation, derived by Hollworth and Dagan (1980) for an impingement array with staggered coolant extraction, is used. The thermal effect of a TBC is reproduced with a thin layer of cork. The conductivity and thickness of the cork have been chosen to replicate the \( R_{\text{TBC}}/R_w \) within the range of a typical engine.
Table 7.1 Conjugate Endwall and TBC Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Model</th>
<th>Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>(M_{\text{avg}})</td>
<td>0.6, 1.0, 2.0</td>
<td>1.0–2.0</td>
</tr>
<tr>
<td>(k_w), W/m-K</td>
<td>0.99–1.06</td>
<td>22</td>
</tr>
<tr>
<td>(t_w), cm</td>
<td>1.27</td>
<td>0.20</td>
</tr>
<tr>
<td>(\text{Bi} = h_\infty t_w/k_w)</td>
<td>0.3–0.7 (Lynch et al., 2011b)</td>
<td>0.27</td>
</tr>
<tr>
<td>(M = 0.6)</td>
<td>1.1–2.3 (Hollworth &amp; Dagan, 1980; Lynch et al., 2011b)</td>
<td></td>
</tr>
<tr>
<td>(M = 1.0)</td>
<td>0.7–1.4 (Hollworth &amp; Dagan, 1980; Lynch et al., 2011b)</td>
<td>1.0</td>
</tr>
<tr>
<td>(M = 2.0)</td>
<td>0.5–1.1 (Hollworth &amp; Dagan, 1980; Lynch et al., 2011b)</td>
<td></td>
</tr>
<tr>
<td>(R_{\text{TBC}}, m^2 K/W)</td>
<td>0.035 (Kistenmacher, 2013)</td>
<td>(8 \times 10^{-5}–9 \times 10^{-4}) (Bunker, 2009; Feuerstein et al., 2008; Padture et al., 2002; Soechting, 1999)</td>
</tr>
<tr>
<td>(R_{\text{TBC}}/R_w)</td>
<td>2.5</td>
<td>0.6–9.3</td>
</tr>
</tbody>
</table>

7.4 Experimental Methods

Overall effectiveness (\(\phi\)) and TBC effectiveness (\(\tau\)) were measured for the endwall of a linear cascade in the corner test section of the large-scale, low speed, closed loop wind tunnel shown in Figure 7.2a. This facility splits the flow upstream of the cascade into mainstream and coolant streams. The mainstream section is heated by a heater bank and passes through flow conditioning elements including a turbulence grid \(\sim 11C_{\text{ax}}\) upstream of the test section. A more detailed description of the wind tunnel and flow conditioning elements can be found in (Lynch et al., 2011b). The mainstream temperature is measured \(0.52C_{\text{ax}}\) in the axial direction upstream of the blades at multiple spanwise and pitchwise locations with a thermocouple rake. The mainstream temperature varies by no more than \(\pm 0.6^\circ\)C from the average \(T_\infty\). A Pitot probe, also inserted \(0.5C_{\text{ax}}\) upstream, is used to measure the inlet mainstream velocity, \(U_\infty\). The standard deviation over the mean \(U_\infty\) is less than 1%.

The coolant passes through a desiccant drier and two heat exchangers that chill the coolant, before entering the plenum located below the endwall. A laminar flow element, LFE, directly measures the total coolant flowrate, which is adjusted to achieve the necessary blowing ratios. The blowing ratios reported in this study, \(M_{\text{avg}}\), are an average of the local M at each film cooling hole. The uncertainty in coolant flowrate is estimated for a 95% confidence interval as
± 3%, using sequential perturbation (Moffat, 1988). To measure the internal coolant temperature, $T_{c,in}$, there are two thermocouples located approximately 8.7D below the impingement plate, which agree within ± 3°C. The experimental setup achieves a typical mainstream to coolant temperature difference of 40°C, which provides a coolant to mainstream density ratio of about 1.15.

Figure 7.2 Depiction of (a) the large-scale low-speed wind tunnel, and (b) the test section containing the Pack-B linear blade cascade and conjugate endwall.

The test section contains a seven blade linear cascade based on the low-pressure turbine Pack-B airfoil, which has been used in many other studies (Knezevici et al., 2010; Lake et al., 1999; Lawson et al., 2013; Lynch et al., 2011a, 2011b; Mahallati et al., 2007; Mensch & Thole, 2014; Murawski & Vafai, 2000; Popovic et al., 2006; Praisner et al., 2007, 2008; Zoric et al., 2007). Figure 7.2b shows the top view of the test section including the endwall constructed of Corian® colored in green. Outside of the passage, the endwall is constructed from medium density fiberboard (MDF). The center passages, 3 and 4, have film and impingement cooling
and were used for this study. A summary of the blade geometry and mainstream flow conditions is given in Table 7.2. At 0.5C₀ upstream of the blade leading edge in the axial direction, the δ₉₀/S is 0.089 and the freestream turbulence is 7%, as predicted by the CFD solution from this study. The details of the simulations are described in the following section.

**Table 7.2 Flow Conditions and Blade Geometry**

<table>
<thead>
<tr>
<th>Scale factor</th>
<th>Inlet U₀,₀₉₉</th>
<th>8.6</th>
<th>10.5 m/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>C₀</td>
<td>0.218 m</td>
<td>Inlet flow angle</td>
<td>35°</td>
</tr>
<tr>
<td>p/C₀</td>
<td>0.826</td>
<td>Exit flow angle</td>
<td>60°</td>
</tr>
<tr>
<td>S/C₀</td>
<td>2.50</td>
<td>Inlet Ma</td>
<td>0.029</td>
</tr>
<tr>
<td>Inlet Re</td>
<td>1.22x10⁵</td>
<td>Exit Ma</td>
<td>0.047</td>
</tr>
<tr>
<td>Exit Re</td>
<td>1.98x10⁵</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Periodicity of the cascade is determined by taking measurements of the pressure distribution at the midspan of all seven blades. In Figure 7.3, a typical set of data for the pressure coefficient, C₀, is plotted versus normalized axial distance. The measured C₀ agree well with the CFD results from this study, as well as an inviscid 2-D CFD prediction from (Lynch et al., 2011b).

The endwall configuration was designed with a generic internal and external cooling geometry to replicate the relevant non-dimensional parameters given in Table 7.1. Figure 7.4a shows the side view of the plenum, cooling features, endwall, and the layer of TBC. A splash plate is used in the experiments to spread and slow the plenum inlet flow. The coolant passes through an impingement plate with 28 staggered holes. The height between the impingement plate and the bottom of the endwall, is H/D = 2.9. Following impingement, the coolant flows into ten angled film cooling holes, which are staggered between the impingement holes. The diameter, D, of the impingement and film cooling holes is 4.4 mm. The film cooling holes are angled at 30° to the surface, resulting in a hole length to diameter ratio, L/D, of 5.8. The locations of the film holes and impingement jets are given in Figure 7.4b. The spacing of the impingement holes in the pitch (y) direction is 4.65D, and the spacing of the impingement rows in the axial (x) direction is 4.65D. The film cooling holes are oriented in the x-y plane to align with endwall streaklines (Lynch et al., 2011b).
Figure 7.3 Pack-B cascade static pressure distribution at the blade midspan compared to CFD predictions.

Figure 7.4 Schematic of internal and external cooling scheme from (a) the side view and (b) the top view showing the outline of the TBC and discrete thermocouple measurements taken on the endwall in the experiments.

In experiments with TBC, the cork layer extends $1/2C_{ax}$ upstream of the blade leading edge and $1/3C_{ax}$ downstream of the trailing edge as illustrated in Figure 7.4b. The thickness of
the cork, \( t_{TBC} \), is 0.45D. The cork is adhered to the endwall using several layers of contact cement to replicate the adhesive methods for which the thermal resistance for the combined cork and adhesive layer is given in Table 7.1.

The steady state external surface temperature is measured using infrared (IR) thermography with a FLIR P20 IR camera. This technique is capable of achieving a high spatial resolution of 5.7 pixels/D. Additional details regarding the IR measurement technique and calibration are available in Mensch and Thole (2014). The external surface temperatures are scaled to \( \phi \) for the cases without TBC, and scaled to \( \tau \) for the cases with TBC. The uncertainty in the external effectiveness is estimated to be ± 0.02 for a 95% confidence interval (Mensch & Thole, 2014).

To determine the outer endwall temperature for \( \phi_{TBC} \), 16 thermocouples are installed to measure the temperature at the interface between the endwall and the cork layer. A high conductivity epoxy, \( k = 4.3 \text{ W/m-K} \), is used to attach the thermocouples to the endwall at the locations shown as blue squares in Figure 7.4b. These points follow two inviscid streamlines referred to hereafter as suction side (SS) and pressure side (PS) streamlines. Using a 95% confidence interval, the uncertainty in \( \phi_{TBC} \) is estimated to be ± 0.01, since temperature measurements were made directly with thermocouples.

### 7.5 Computational Methodology

Conjugate simulations for the endwall with and without TBC were performed using commercial computational fluid dynamics (CFD) software (FLUENT 13.0.0, 2010). The segregated pressure-based SIMPLE algorithm is used to solve the steady-state RANS and energy equations using the SST \( k-\omega \) turbulence model (Menter, 1994) for closure with second-order spatial discretization schemes. The SST \( k-\omega \) model was chosen because it has shown reasonable agreement with experimental results in turbomachinery applications (Dyson et al., 2012; Lynch et al., 2011a; Panda & Prasad, 2012; Schwänen & Duggleby, 2009; Snedden et al., 2009).

The computational grid begins 3.5\( C_{ax} \) upstream of the blade leading edge in the inlet flow direction to capture the flow development upstream of the cascade, as shown in Figure 7.5a. At this location, a velocity inlet is applied with a freestream velocity of 10.5 m/s and a boundary layer profile benchmarked to the measurements by Lynch et al. (2011b). The inlet velocity, turbulent kinetic energy, and specific dissipation profiles are generated using the boundary layer
code TEXSTAN (Crawford, 2009) to match the measured momentum thickness Reynolds number, $Re_0 = 1330$, at the measurement location $2.85C_{ax}$ upstream of the blade leading edge (Lynch et al., 2011b). The temperature at the inlet reflects the typical $T_{\infty}$ in the experiments. An outflow boundary condition is applied $1.5C_{ax}$ downstream of the blade trailing edge in the axial direction. Symmetry is imposed at the top of the domain, which is located at the midspan of the blade in the experiments. A single blade passage is simulated using periodic boundaries that extend vertically through the entire domain, cutting through the mainstream section, the TBC and endwall, four film cooling holes, the impingement channel, and the plenum.

A mass flow inlet boundary condition is applied at the bottom of the computational plenum, which is located $65D$ below the impingement plate to reflect the dimensions of the plenum in the experiments. The mass flow rate and temperature applied at the boundary match the conditions in the experiment for that blowing ratio. Blowing ratios of 1 and 2 are simulated. Air properties used for the flow were incompressible-ideal gas for density, polynomial fits to temperature for thermal conductivity and specific heat, and Sutherland’s law for viscosity. The properties used for the solids are listed in Table 7.1.

A thermally-coupled wall interface is used at all conjugate solid/fluid boundaries. For the simulation with TBC, a cork layer extends along the entire endwall surface of the computational domain except for the film cooling holes. A thermally-coupled wall interface is also used at the boundary between the TBC and the endwall. All other wall surfaces in the domain are modeled as adiabatic.

Separate unstructured grids were generated for the conducting endwall solid, the conducting TBC solid, and the flow domain. A commercial grid generation program, (Pointwise 17.1r3, 2013) was used to generate the unstructured grids for the endwall and TBC geometries, shown in Figure 7.5b. The endwall grid contains 1.5 million cells, and the cork grid contains 0.3 million cells. For the flow domain, the Advancing-Front/Local-Reconnection unstructured grid generation software, AFLR3 (Marcum & Gaither, 1999), was used to create a high-fidelity tetrahedral grid with wall-normal prism layers to resolve the boundary layer on key surfaces (the blade, the entire external endwall surface, the internal endwall surface, the film cooling holes, and the impingement holes) with a wall spacing such that $y^+ < 1$. The unstructured grid for the flow is depicted in Figure 7.5c, showing a slice in the $y$-plane through a film cooling hole inlet and an impingement hole, and in Figure 7.5d, showing a slice in the $x$-plane through the
mainstream, film cooling holes, impingement channel, impingement holes, and plenum. The initial grid size for the flow domain was 9.8 million cells.

Convergence of a simulation was achieved when the normalized residuals were $< 1 \times 10^{-4}$ and the area-averaged endwall $\phi$ changed by less than 0.0015 over 500 iterations. To ensure grid independence, the initial grid without TBC that contained a total of 11.3 million cells was uniformly refined to a grid containing a total of 18.5 million cells. The difference in the solution for $M_{avg} = 1.0$ from the refined grid relative to the initial grid for area-averaged $\phi$ over the endwall was $1 \times 10^{-4}$. Also, the total heat flux at the internal endwall surface varied by less than 0.25% for the refined grid relative to the initial, nominal grid. Thus, it was concluded that the nominal grid was of sufficient resolution for the present conjugate heat transfer predictions and that the CFD solutions were grid insensitive.

Figure 7.5 Depiction of (a) the computational domain and boundary conditions, (b) the surface grid for the endwall and TBC, (c) the prism layer volume grid in the holes and impingement channel, and (d) the volume grid in the mainstream, channel, and plenum.
7.6 Results and Discussion

The effects of blowing ratio, M, and the TBC are examined through measurements and steady state RANS predictions of overall effectiveness without TBC ($\phi$) and with TBC ($\phi_{TBC}$), in addition to the TBC effectiveness, $\tau$. The predictions reveal heat flux and temperature trends that can be used to understand the physical mechanisms involved in the conjugate endwall heat transfer.

7.6.1 Measured and Predicted Temperatures Without TBC

The contours of overall effectiveness, $\phi$ (non-dimensional endwall temperatures) for two blowing ratios of $M_{avg} = 1.0$ and 2.0 are shown in Figure 7.6 and Figure 7.7. The contours show the non-dimensional endwall temperatures measured without TBC (Figure 7.6a and Figure 7.7a) and predicted without TBC (Figure 7.6b and Figure 7.7b). Figures 6c and 7c also show the predicted endwall temperatures under the TBC, $\phi_{TBC}$, which will be discussed in the following section. The film cooling holes and plenum boundaries are shown for reference in Figure 7.6 and Figure 7.7. The horizontal lines just below each blade indicate a wall in the impingement channel that prevents coolant from crossing into another passage after passing through the impingement plate. Note that the experimental results are shown for two passages, indicating good periodicity. To compare, repeated images are shown for the predictions.

For $M_{avg} = 1.0$ the area-averaged $\bar{Nu}$ across entire internal endwall surface is predicted to be 3.2 without TBC. With TBC, the difference is only 0.1%, showing that the CFD is consistent with the internal flow predictions. Likewise the predicted internal $\bar{Nu}$ for $M_{avg} = 2.0$ is nearly the same with and without TBC, with a value of 5.2 and a difference of 0.4%. It is expected that the external heat transfer coefficients do not change between the cases with and without TBC since the flowfield did not change.

The comparison of the measured and predicted endwall temperature ($\phi$) contours without TBC (Figure 7.6a and b; and Figure 7.7a and b) at both blowing ratios are very good. The simulations correctly predict the extent of lateral conduction into the uncooled areas. As seen in Figure 7.6a and b at $M_{avg} = 1.0$, there are some discrepancies between the measured and predicted $\phi$ in the areas with film cooling. As many other studies have found, RANS CFD often has difficulty predicting mixing in the film cooling jet shear layer and difficulty predicting film attachment. For $M_{avg} = 2.0$ in Figure 7.7a and b, a small difference can be seen in the influence
of the film cooling jets closest to the suction side. Although the CFD correctly predicts the jet detachment, the predicted $\phi$ here is less than in the measurements. The simulations are underpredicting the cooling influence that the detached jets have on the wall, which was also observed by Dyson et al. (2012). The strong effect of in-hole convection at $M_{avg} = 2.0$ is well predicted in the high $\phi$ at the exits of the film cooling holes.

![Overall effectiveness contours for $M_{avg} = 1.0$](image)

Figure 7.6 Overall effectiveness contours for $M_{avg} = 1.0$ (a) measured without TBC, (b) predicted without TBC, and (c) predicted under the TBC.

A small difference in $\phi$ is also seen downstream in the passage for both blowing ratios. In the downstream region of Figure 7.6b and Figure 7.7b, the simulations show a slightly lower $\phi$ (by about 0.05) and a warmer endwall than the experiment. These differences are attributed to small heat losses present in the experiment that are not captured in the simulation. An example of a surface modeled as adiabatic that loses heat to the surrounding environment is the downstream side of the Corian® endwall. The endwall is surrounded on the side and bottom by medium density fiberboard shown in Figure 7.4a. The conduction heat loss from the downstream sides of the endwall into the fiberboard ($k = 0.3 \text{ W/m}^2$) is estimated to be on the same order as the convective heat transfer into the endwall from the mainstream. The lateral
conduction in this part of the endwall is estimated to be an order of magnitude less because the temperature gradients in this region of the endwall are small.

Figure 7.7 Overall effectiveness contours for $M_{\text{avg}} = 2.0$ (a) measured without TBC, (b) predicted without TBC, and (c) predicted under the TBC.

The data in Figure 7.8a-f give the local $\phi$ along the length of the PS and SS streamlines shown in Figure 7.4b. The experimental $\phi$ without TBC and the predicted $\phi$ (in red) have been extracted from the measured and predicted contours in Figure 7.6 and Figure 7.7. The measured and predicted $\phi_{\text{TBC}}$ will be discussed in the next section. For $M_{\text{avg}} = 0.6$, the experimental data are shown in Figure 7.8a for the PS and Figure 8d for the SS. (No CFD simulations were done at the low blowing ratio.) Similar graphs are shown for $M_{\text{avg}} = 1.0$ in Figure 7.8b and e and for $M_{\text{avg}} = 2.0$ in Figure 7.8c and f. A repeated data set is shown for $M_{\text{avg}} = 1.0$ in Figure 7.8b and e for the experiments with and without TBC. The excellent agreement between repeated experiments indicates the experiments are reproducible.

As expected from the contours, the predicted $\phi$ shows good agreement with the measurements, especially on the PS (Figure 7.8b and c). The PS streamline crosses a few film cooling holes at or just upstream of the hole exits, which show up as sharp peaks in the graphs.
Other than at these peaks, the endwall along the PS streamline is influenced by internal cooling only. The effects of both internal impingement and in-hole convection increase with blowing ratio, which is well captured by the predictions.

![Graphs showing measured and predicted values](image)

**Figure 7.8** Comparison of overall effectiveness with and without TBC, showing measured and predicted values, along inviscid streamlines, PS for a–c and SS for d–f.

The agreement on the SS is not as close as on the PS because the SS streamline crosses the downstream path of several film cooling jets. The SS $\phi$ data, shown in Figure 7.8d-f, have a sharp drop around $s/C_{ax} = 0.2$ following a film cooling hole. At $M_{avg} = 0.6$ (Figure 7.8d), the SS line continues to slowly decrease consistent with the behavior of an attached film cooling jet. As blowing ratio increases (Figure 7.8e and f), the behavior reflects that of a detached and reattached jet because measured $\phi$ slightly increases again following the sharp drop. For $M_{avg} = 1.0$ between $s/C_{ax}$ of 0.1–0.5 (Figure 7.8e), the prediction shows the trend of detachment and reattachment, but the resulting $\phi$ is over-predicted. For $M_{avg} = 2.0$, there is a slight under-prediction of $\phi$ on the SS (Figure 7.8f) around $s/C_{ax}$ of 0.1–0.3, which is related to the deficiency of RANS in predicting the diffusion of detached film cooling jets.
The data obtained in the simulations can be used to provide more insight into the conjugate heat transfer in locations where it is difficult to obtain experimental measurements. The predicted distributions of dimensionless temperature within the endwall and the impingement channel are revealed in Figure 7.9 through two slices of the domain. Figure 7.9a shows the slices at the leading edge of the blade, $x/C_{ax} = 0$, which also line up with the first row of impingement holes. The slices shown in Figure 7.9b are located at $x/C_{ax} = 0.09$, which passes through the second row of impingement holes and also partway through five film cooling holes. Both sets of slices display in white the impingement plate and the separating wall in the impingement channel since these features were non-conducting.

Figure 7.9 Conjugate CFD prediction of non-dimensional temperature in the fluid and the solid at different two slices (a) at the first row of impingement holes and (b) at the second row of impingement holes.

From Figure 7.9, it is observed that impingement and in-hole convection become more effective with the increase in blowing ratio. Figure 7.9a shows that the first row of impingement holes is very effective at cooling the endwall. Just above the bottom of the endwall, the
temperature contours show the effect of individual impingement jets, but at the top of the endwall, the cooling has spread and the wall temperature is uniform. In Figure 7.9b, the effect of the in-hole convective cooling on the wall is shown. The temperature contours bend as they encounter a film cooling hole because there is significant heat transfer at the hole surface. The temperature contours also give an indication of the flow patterns of the impingement jets in the channel. The coolant impinges on the internal side of the endwall, and then flows outward, and down towards the impingement plate as it encounters the flow from the adjacent jets. The flow moving towards the impingement plate has picked up heat from the endwall and is warmed slightly.

7.6.2 Measured and Predicted Temperatures With TBC

The layer of TBC prevents optical access to the endwall surface so that the IR camera cannot measure $\phi_{\text{TBC}}$ on the endwall. Therefore, the measured $\phi_{\text{TBC}}$ is compared to the case without TBC in Figure 7.8a-f for the discrete thermocouple measurements made below the TBC along inviscid streamlines. With TBC, higher cooling performance compared with the no TBC cases is observed along the inviscid streamlines for the three blowing ratios. TBC increases the effectiveness on the PS line at all measurement locations by nearly a constant amount. The increase from $\phi$ to $\phi_{\text{TBC}}$ varies on the SS along the length of the streamline, especially between $0.2 < s/C_{ax} < 0.7$. The local effects of film cooling seen on the endwall without TBC have been smeared out in $\phi_{\text{TBC}}$ because the TBC is insulating the endwall from the external flow, which includes film cooling.

The predictions of $\phi_{\text{TBC}}$ in Figure 7.6c, Figure 7.7c, Figure 7.8, and Figure 7.9a-b also show that the TBC provides a significant cooling effect on the endwall. The predicted contours of $\phi_{\text{TBC}}$ for $M_{\text{avg}} = 1.0$, in Figure 7.6c and for $M_{\text{avg}} = 2.0$ in Figure 7.7c, are significantly higher at all locations on the endwall relative to $\phi$ without TBC. When the blowing ratio is increased from $M_{\text{avg}} = 1.0$ (Figure 7.6c) to $M_{\text{avg}} = 2.0$ (Figure 7.7c), the area of high effectiveness under the TBC increases. The predicted $\phi_{\text{TBC}}$ along the inviscid streamlines in Figure 7.8b-c and e-f (dashed red lines) compare fairly well with the discrete $\phi_{\text{TBC}}$ measurements. The predictions show some influence of the film cooling on $\phi_{\text{TBC}}$, which is not observed in the measurements. Both $\phi_{\text{TBC}}$ and $\phi$ are under-predicted downstream in the passage for $s/C_{ax} > 0.7$. These discrepancies are

139
attributed to the conduction losses at the endwall sides previously discussed in reference to the $\phi$ contours. When TBC is included in Figure 9 in the slices of non-dimensional temperature, the endwall temperatures are greatly reduced, even in the areas not around the impingement and film cooling holes. In addition, the fluid in the channel and the film cooling holes remains cooler with TBC than the cases without TBC. When TBC is added, there is less warming of the coolant because the TBC reduces the internal wall temperature.

The average increases in $\phi$ due to changes in the blowing ratio and due to the addition of TBC are compared in Table 7.3. The change due to $M$ is an area-average across the impingement area from Mensch and Thole (2014). The $\Delta\phi_{\text{TBC}}$ is the average difference between $\phi_{\text{TBC}}$ and $\phi$ (without TBC) at the measurement locations along the streamlines. The values in the table show that the improvement due to the addition of TBC is greater than the improvement from increasing blowing ratio. Changing blowing ratio is less effective than TBC at reducing endwall temperatures, which is consistent with the findings on the vane surface (Davidson et al., 2014b).

**Table 7.3 Measured Improvement in Overall Effectiveness Due to an Increase in Blowing Ratio and Due to the Addition of TBC**

<table>
<thead>
<tr>
<th>$\Delta M$</th>
<th>$\Delta\phi_M$ (Mensch &amp; Thole, 2014)</th>
<th>$M_{\text{avg}}$</th>
<th>$\Delta\phi_{\text{TBC}} = \phi_{\text{TBC}} - \phi$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.6 – 1.0</td>
<td>0.05</td>
<td>0.6</td>
<td>0.13</td>
</tr>
<tr>
<td>1.0 – 2.0</td>
<td>0.04</td>
<td>1.0</td>
<td>0.14</td>
</tr>
<tr>
<td></td>
<td></td>
<td>2.0</td>
<td>0.17</td>
</tr>
</tbody>
</table>

The improvement in $\phi$ due to the TBC for the measurements and simulations has been plotted in Figure 7.10. The values for the experiment are those given in Table 7.3. The predicted data are an average across the entire conducting endwall surface. Figure 7.10 demonstrates that $\Delta\phi_{\text{TBC}}$ slightly increases with blowing ratio. The measured and predicted values for improvement with TBC agree very well, indicating that the discrete thermocouple locations used to measure the endwall temperatures under the TBC provide a good indication of the effect throughout the passage.
Figure 7.10 Measured and predicted improvement with TBC, $\Delta \phi_{TBC}$, and the predicted $\Delta q_r$ for the external endwall surface plotted as a function of $M_{avg}$.

The net heat flux reduction with TBC, $\Delta q_r$, can be calculated at the endwall outer surface using Equation (7.6).

$$\Delta q_r = \frac{q_w - q_{w,TBC}}{q_w}$$ (7.6)

The predicted $\Delta q_r$ is plotted in Figure 7.10 for the y-axis on the right side of the figure. The simulations predict that $\Delta q_r$ also increases with blowing ratio. Similar trends are found for $\Delta q_r$ when applied to the other endwall surfaces (the internal endwall surface and the film cooling holes). Therefore, adding TBC gives a greater reduction in $q_w$, and a greater improvement in $\phi$ at higher blowing ratios. With TBC, the endwall is insulated from the hot mainstream, but still influenced by the internal cooling. In general, internal cooling becomes more effective as blowing ratio increases, generating higher heat transfer coefficients. The insulating effect of the TBC allows the internal impingement and in-hole convection to cool the endwall more effectively, as can be seen in the temperature slices in Figure 7.9.

The measured and predicted dimensionless temperatures on the outer TBC surface, $\tau$, are shown in Figure 7.11. Experimental measurements are shown for three blowing ratios including
Figure 7.11a at $M_{\text{avg}} = 0.6$, Figure 7.11b at $M_{\text{avg}} = 1.0$, and Figure 7.11c at $M_{\text{avg}} = 2.0$. Similarly, computational predictions are given in Figure 7.11d for $M_{\text{avg}} = 1.0$ and Figure 7.11e for $M_{\text{avg}} = 2.0$. Because the TBC has a higher thermal resistance than the endwall, $\tau$ is generally lower than $\phi$ and is closer in appearance to the effectiveness for an adiabatic wall, $\eta$. However, as blowing ratio increases, impingement and in-hole convection begin to have a greater effect on $\tau$. The insulating effect of TBC is also observed in the lower temperatures measured at the exit of the holes compared to the cases without TBC (seen in Figure 7.6 and Figure 7.7). The reduction in coolant warming with TBC is also observed in the temperature slices in Figure 7.9.

![Image of TBC effectiveness contours](image)

Figure 7.11 Contours of TBC effectiveness for (a) $M_{\text{avg}} = 0.6$ measured, (b) $M_{\text{avg}} = 1.0$ measured, (c) $M_{\text{avg}} = 2.0$ measured, (d) $M_{\text{avg}} = 1.0$ predicted, and (e) $M_{\text{avg}} = 2.0$ predicted.

The corresponding predictions for $\tau$ show reasonable agreement to the experiments. Like the measurements, the simulations predict temperatures on the outside of the TBC to be hotter
than the bare endwall. However, the predicted $\tau$ is slightly lower than measured, especially for $M_{\text{avg}} = 2.0$. Also, the simulation for $M_{\text{avg}} = 1.0$ in Figure 7.11d shows more jet attachment than the experiments, which is consistent with the $\phi$ comparison at the same blowing ratio. The fully detached jets for $M_{\text{avg}} = 2.0$ in Figure 7.11e have less influence on $\tau$ compared to the experiments, a trend that was also observed for $\phi$.

The external TBC temperatures, $\tau$, along the inviscid SS and PS streamlines are given in Figure 7.12a for $M_{\text{avg}} = 0.6$, Figure 7.12b for $M_{\text{avg}} = 1.0$, and Figure 7.12c for $M_{\text{avg}} = 2.0$. The data within the film cooling hole outlets have been removed. As blowing ratio increases, the SS data (dashed lines) stay about the same or slightly decrease because the film cooling jets become detached from the TBC surface. The decreased cooling by the film jets is balanced by the increased cooling by internal impingement. In contrast, the PS data increase with each increase in blowing ratio because the PS streamline does not cross the path of the film cooling jets, and is influenced by internal cooling. As discussed in reference to the contours, the simulations predict lower $\tau$ than measured and less influence on $\tau$ by the internal cooling than the experiments indicate. This is more apparent on the PS after $s/C_{\text{ax}} > 0.25$, and on the SS after $s/C_{\text{ax}} > 0.7$, downstream of the film cooling jets. Despite the under-prediction, the trends of the data in the passage are well captured by the simulations.

![Graph](image)

**Figure 7.12** Comparison of TBC effectiveness with film and impingement cooling, showing measured and predicted values, along inviscid streamlines, for (a) $M_{\text{avg}} = 0.6$, (b) $M_{\text{avg}} = 1.0$, and (c) $M_{\text{avg}} = 2.0$. 
7.7 Conclusions

Overall and TBC effectiveness measurements were compared to conjugate CFD predictions for the cases with and without TBC at different blowing ratios. The improvement in overall effectiveness due to TBC was evaluated and found to be significant. Adding TBC produced a greater improvement in overall effectiveness than the improvements achieved by increasing blowing ratio alone. The TBC protected the endwall from the hot mainstream, reduced heat transfer, and allowed the internal cooling to be more effective. As blowing ratio increased, a greater improvement in overall effectiveness was observed because the TBC was more effective at reducing heat transfer. The reduction in heat transfer with TBC also caused the outer TBC temperature to be higher in comparison to the endwall temperature without TBC.

The predicted overall effectiveness with no TBC showed good agreement with the measurements. The main differences were observed in the prediction of film cooling jet attachment for $M_{\text{avg}} = 1.0$ and diffusion for $M_{\text{avg}} = 2.0$, which were consistent with previous literature findings. The predictions of TBC effectiveness were reasonably close to the measurements, and the correct trends were captured by the simulations. In conclusion, the predictive model in conjunction with a suitable grid gave reasonable predictions of conjugate heat transfer on an endwall.

The simulations were used to reveal the temperature behavior within the impingement channel and the endwall. The temperature contours showed that the heat transfer was highly three-dimensional, and steep temperature gradients existed in the wall. The effect of individual impingement jets and film cooling holes were observed inside the wall showing the thermal gradients. The predicted temperature distributions inside the wall and impingement channel demonstrated the importance of matching the relevant non-dimensional parameters in a conjugate model.
CHAPTER 8. SIMULATIONS OF MULTIPHASE PARTICLE DEPOSITION ON A GAS TURBINE ENDWALL WITH IMPINGEMENT AND FILM COOLING††

Abstract
Replacing natural gas fuels with coal derived syngas in industrial gas turbines can lead to molten particle deposition on the turbine components. The deposition of the particles, which originate from impurities in the syngas fuels, can increase surface roughness and obstruct film cooling holes. These deposition effects increase heat transfer to the components and degrade the performance of cooling mechanisms, which are critical for maintaining component life. The current study dynamically simulated molten particle deposition on a conducting blade endwall with the injection of molten wax. The key non-dimensional parameters for modeling of conjugate heat transfer and deposition were replicated in the experiment. The endwall was cooled with internal impingement jet cooling and film cooling. Increasing blowing ratio mitigated some deposition at the film cooling hole exits and in areas of coolest endwall temperatures. After deposition, the external surface temperatures and internal endwall temperatures were measured and found to be warmer than the endwall temperatures measured before deposition. Although the deposition helps insulate the endwall from the mainstream, the roughness effects of the deposition counteract the insulating effect by decreasing the benefit of film cooling and by increasing external heat transfer coefficients.

8.1 Introduction
An alternative power generation method in recent years uses coal derived fuels in a gas turbine power system known as integrated gasification combined cycle (IGCC). Coal is converted to hydrogen and carbon monoxide synthesis gas (syngas), which is used as the fuel instead of natural gas in a conventional industrial gas turbine. One of the challenges associated

with the use of this alternative fuel technology comes from residual solids that remain through the gasification process. These particles can become molten upstream of the turbine due to the extreme temperatures experienced in the combustor section. Downstream of the combustor, the particles encounter relatively cooler surfaces along the cooled airfoil components and tend to deposit on those surfaces. The deposition can potentially interfere with critical turbine cooling technologies designed to maintain turbine component life. Without proper cooling, the turbine components are at risk of exposure to gas temperatures above the material limits.

There are two objectives of this study on an internally and externally cooled endwall of a turbine blade. The first is to evaluate the influence of the endwall cooling on the distribution of deposition. The deposition of molten particles in the turbine is a complicated three phase flow process that depends on many dynamic parameters. It is necessary to understand the relevant parameters controlling the deposition to be able to accurately simulate the deposition in experiments.

Once the deposition has been applied to the endwall, the second objective is to determine the change in endwall cooling performance and the causes for the change. The cooling performance is quantified through temperature measurements of the surfaces. The current study considers the effects of coolant blowing ratio as well as cooling configuration, with and without internal impingement cooling.

### 8.2 Relevant Literature

While many studies in the literature have studied the effects of deposition geometry and roughness on gas turbine cooling, more recent studies have dynamically simulated the deposition process in a laboratory environment as well. Dynamic molten particle deposition methods capture a variety of effects, most notably particle Stokes number (Stk) and molten character of the particles, both which determine the amount and character of the deposition.

Molten coal ash particles were dynamically deposited on stationary hardware by combustion gases in accelerated blowdown facilities used by Ai et al. (2012), Webb et al. (2012), and Casaday et al. (2013). Ai et al. (2012) optically measured the external temperature on a film cooled surface and found that the temperature increased as more particles deposited. Increasing the blowing ratio mitigated the accumulation of deposition and rise in temperature. Webb et al. (2012) tested the consequences of injecting different types of coal ash at different mainstream
temperature conditions and discovered that there was a threshold temperature for deposition that varied depending on the type of ash. As the gas temperature was increased beyond the threshold, the deposit size and thickness increased. Casaday et al. (2013) also found levels of increased deposition due to higher gas temperatures from simulated hot streaks. In both studies, the deposition was thickest on the leading edge and pressure side of the vanes. Film cooling provided some benefit in reducing deposition for the types of ash that had the mildest deposition amounts (Webb et al., 2012).

A method to dynamically simulate molten particle deposition with wax in low speed wind tunnel facilities has been employed by several researchers (Albert & Bogard, 2012, 2013a; Davidson et al., 2014a; Kistenmacher et al., 2014; Lawson et al., 2013; Lawson & Thole, 2011, 2012a, 2012b). Lawson et al. (2013; 2011, 2012a, 2012b) simulated deposition on adiabatic endwall geometries with film cooling. Initially the degradation of the film cooling effectiveness increased over time, but then reached an equilibrium, when the rate of deposition was balanced by the rate of erosion. Lawson et al. (2013; 2011, 2012a, 2012b) varied the molten character of the particles by changing the mainstream temperature. A Thermal Scaling Parameter (TSP) was defined to quantify the molten character of the particles. For lower values of TSP, corresponding to more solid than liquid particles, the deposition had increased surface roughness and greater reduction in adiabatic film cooling effectiveness.

A similar wax deposition facility was employed by Albert and Bogard (2012, 2013a), Davidson et al. (2014a) and Kistenmacher et al. (2014) for conducting vane geometries with internal cooling and film cooling. In conducting wall experiments, the appropriate dimensionless parameters must be engine-matched to model the correct non-dimensional temperature distributions (Albert et al., 2004). The relevant dimensionless parameters for conjugate studies include the Biot number, Bi; the ratio of external to internal heat transfer coefficients, $h_\infty/h_i$; the Reynolds number, Re; and the scaled geometry. Albert and Bogard (2012, 2013a) observed significant amounts of deposition on the leading edge and pressure sides of the vane. However, there was threshold for surface temperature above which a significant amount of deposition was observed. When the vane surface temperature was below the threshold, deposition was partially mitigated. In light of the importance of surface temperature to deposition, Davidson et al. (2014a) and Kistenmacher et al. (2014) included the thermal effect of a thermal barrier coating (TBC) on the vane in their deposition simulations. Both wax and TBC thermal conductivities
were appropriately scaled to the vane conductivity. Measurements of the vane temperature, under the TBC, showed lower temperatures for the case with deposition relative to no deposition due to the insulating effect of the deposition layer.

To the authors’ knowledge there are no studies in the literature which focus on the deposition on an endwall with a conducting wall boundary condition to determine the effect on endwall temperature. The studies mentioned above either focused on the deposition on the vane surface or used an adiabatic boundary condition for the endwall. The deposition behavior and effect on cooling performance for the endwall is different than the vane because of the orientation of the surface to the mainstream and the presence of passage secondary flows. The vortices associated with endwall flow have been shown to locally increase deposition where the flow of the vortex impacts the endwall, as well as locally mitigate deposition where the flow of the vortex pulls away from the endwall (Lawson & Thole, 2012a, 2012b; B. Casaday et al., 2012). The vortices in the passage also locally increase the external heat transfer coefficient as shown by Kang and Thole (2000). The distribution of the external endwall heat transfer coefficients was investigated by Lynch et al. (2011a) through both experiments and computational predictions of an endwall with a constant wall heat flux boundary condition. The thermal performance of a conjugate endwall with passage secondary flows was investigated by Mensch and Thole (2014) without deposition. For this previous study, the endwall was cooled with the same cooling features as the current study. It was found that the internal impingement cooling had a greater influence than film cooling on the scaled endwall temperatures.

8.3 Conjugate Endwall and Deposition Model

A simplified model of an impingement and film cooled endwall with a layer of deposition is depicted in Figure 8.1. Figure 8.1 identifies the critical parameters and temperatures for the conjugate model development, the experiments and the analysis. From the one-dimensional heat transfer through the endwall in Figure 8.1, Equation (8.1) can be derived for the dimensionless wall temperature with deposition, \( \phi_{dep} \). Equation (8.1) is written in non-dimensional form, revealing the dimensionless parameters affecting heat transfer. By matching these parameters and the relative dimensions of all the geometric features to those in the engine, the experiments can provide engine relevant dimensionless temperatures. Equation (8.1) is written for the case with deposition, \( \phi_{dep} \) whereby this is the dimensionless outside wall temperature. In the case of
no deposition, the equation for the outside wall temperature, $\phi$, reduces to Equation (8.2), previously reported in (Mensch & Thole, 2014; Williams et al., 2014).

$$
\phi_{dep} = \frac{T_w - T_{c,in}}{T_\infty - T_{c,in}} = \frac{1 - \chi_1 \eta}{1 + \frac{\chi_1 \eta}{Bi + \frac{h_\infty}{h_i}}} + \chi_1 \eta
$$  \hspace{1cm} (8.1)

$$
\phi = \frac{T_\infty - T_w}{T_\infty - T_{c,in}} = \frac{1 - \chi_1 \eta}{1 + Bi + \frac{h_\infty}{h_i}} + \chi_1 \eta
$$  \hspace{1cm} (8.2)

The dimensionless parameters revealed in Equations (8.1) and (8.2) include the endwall Bi, the $h_\infty/h_i$, the coolant warming factor, $\chi_1 \eta$, the adiabatic effectiveness, $\eta$, the ratio of deposition thickness to endwall thickness $t_{dep}/t_w$, and the ratio of deposition conductivity to wall conductivity $k_{dep}/k_w$.

![Figure 8.1 Configuration of a conjugate endwall with impingement and film cooling and simulated deposition.](image)

Adiabatic effectiveness is commonly reported in the literature to indicate the effectiveness of external cooling techniques such as film cooling. In Equations (8.1) and (8.2), $\eta$ represents the non-dimensional driving temperature for external convection, which varies due to film cooling. The geometry as well as the mainstream and film cooling flow parameters (film cooling blowing ratio, $M$, for example) all influence $\eta$. The coolant warming factor, $\chi_1 \eta$, defined in Equation (8.3), is a correction factor for $\eta$ to account for the warming of the coolant from the plenum through impingement and in-hole convection to the film cooling hole exit, from $T_{c,in}$ to
As a result, the product, $\chi_\eta \eta$, and $\phi$ have the same normalizing temperature difference (Williams et al., 2014).

$$\chi_\eta = \frac{T_\infty - T_{c,\text{exit}}}{T_\infty - T_{c,\text{in}}}$$ (8.3)

The thickness ratio, $t_{\text{dep}}/t_w$, is a parameter that represents the size and distribution of deposits on the endwall. Lawson and Thole (2012a) have shown that the size and distribution of multi-phase particle deposition depend on the particle Stk and TSP. The particle Stk characterizes the trajectory of the particles. The TSP quantifies the molten character of the particles and influences the particle sticking behavior (Lawson & Thole, 2012a). The TSP is the ratio of particle solidification time, from a lumped capacitance analysis, to the particle travel time from the combustor to the turbine surface. For TSP > 1, the particles are mostly molten. For TSP < 1, the particles are mostly solid.

The non-dimensional endwall and deposition parameters for the experiments in this study matched to engine conditions are given in Table 8.1. The blowing ratio, $M_{\text{avg}}$, is the average ratio of coolant to mainstream mass flux across the film cooling holes. The distribution of endwall $h_e$ is obtained from measurements by Lynch et al. (2011b). An engine matched Bi is achieved by using a Corian® endwall. To estimate $h_i$, a Nusselt number correlation for an impingement array with staggered coolant extraction (Hollworth & Dagan, 1980) is used. A molten wax spray technique is used to achieve the deposition parameters in the laboratory. Details regarding this technique can be found in the following section and in Lawson and Thole (2012a). The conductivity of the paraffin wax deposits ($T_{\text{melt}} = 60^\circ\text{C}$) is estimated assuming an air porosity of 0.6 in the deposits, similar to the porosity of engine deposition (Richards et al., 1992). The thermal conductivity of coal ash deposition depends on deposition temperature, chemical composition and porosity, which is reflected in the range of $(k_{\text{dep}}/k_w)_{\text{engine}}$ given in Table 8.1. Table 8.1 also shows a wide range of Stk and TSP for engines due to the range of possible particle diameters (Bons et al., 2007). The median values of Stk and TSP in the experiment are chosen to simulate the distribution of these parameters in the engine. The Stk is provided for definitions based on the film cooling diameter, $D$, as well as the mainstream length scale, $C_{\text{ax}}$. 

150
Table 8.1 Conjugate Endwall and Deposition Parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Experimental Model</th>
<th>Typical Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>$M_{avg}$</td>
<td>0.6, 1.0, 2.0</td>
<td>1.0–2.0</td>
</tr>
<tr>
<td>$k_w$, W/m-K</td>
<td>0.99–1.06</td>
<td></td>
</tr>
<tr>
<td>$t_w$, cm</td>
<td>1.27</td>
<td></td>
</tr>
<tr>
<td>Bi = $h_w t_w/k_w$</td>
<td>0.3–0.7 (Lynch et al., 2011b)</td>
<td>0.27</td>
</tr>
<tr>
<td></td>
<td>$M = 0.6$</td>
<td>1.1–2.3 (Hollworth &amp; Dagan, 1980; Lynch et al., 2011b)</td>
</tr>
<tr>
<td></td>
<td>$h_{w}/h_i$</td>
<td>M = 1.0 0.7–1.4 (Hollworth &amp; Dagan, 1980; Lynch et al., 2011b)</td>
</tr>
<tr>
<td>$k_{dep}/k_w$</td>
<td>0.1</td>
<td>0.07–0.1 (Richards et al., 1992; Rezaei et al., 2000)</td>
</tr>
<tr>
<td>StkD</td>
<td>median: 6 (Lawson &amp; Thole, 2012a)</td>
<td>0.004–40 (Bons et al., 2007)</td>
</tr>
<tr>
<td>StkCax</td>
<td>median: 0.1 (Lawson &amp; Thole, 2012a)</td>
<td>8x10$^{-5}$–0.7 (Bons et al., 2007)</td>
</tr>
<tr>
<td>TSP</td>
<td>median: 0.3(Lawson et al., 2013)</td>
<td>1x10$^{-4}$–1.2 (Bons et al., 2007; Q. Wang et al., 2008; Li et al., 2007; Dennis et al., 2007)</td>
</tr>
</tbody>
</table>

Besides the scaled outer wall temperature, two other temperatures of interest in Figure 8.1 are the scaled internal wall temperature, $\phi_i$, and the scaled external temperature of the deposition, $\omega$, defined in Equations (8.4) and (8.5). As wax is used for deposition in the experiments, $\omega$ is referred to as the wax effectiveness. Since $T_{w,i}$ and $T_{wax}$ are on the same heat transfer circuit used to derive Equation (8.1), it follows that $\phi_i$ and $\omega$ depend on the same parameters as $\phi_{dep}$.

$$\phi_i = \frac{T_{\infty} - T_{w,i}}{T_{\infty} - T_{c,in}}$$ (8.4)

$$\omega = \frac{T_{\infty} - T_{wax}}{T_{\infty} - T_{c,in}}$$ (8.5)

8.4 Experimental Methods

The large-scale, low speed, closed loop wind tunnel depicted in Figure 8.2a is used to perform the deposition simulation and thermal measurements on the endwall of the linear blade cascade shown in Figure 8.2b. The procedure of the experiments involves first the dynamic simulation of the deposition through a steady state wax spray issuing from small nozzles in the
turbulence grid. Then a second steady state experiment is performed to measure the temperature distribution on the wax deposits and the endwall. In each experiment the flow and thermal conditions are controlled such that the Re and the important parameters in Table 8.1 are achieved.

![Figure 8.2 Depiction of (a) the large-scale low-speed wind tunnel, and (b) the test section containing the Pack-B linear blade cascade and conjugate endwall.](image)

The wind tunnel facility allows for a temperature difference between the mainstream and coolant streams. The coolant flow is separated from the mainstream flow upstream of the cascade where the tunnel widens in Figure 8.2a. The mainstream section is heated by a heater bank and passes through flow conditioning elements including a turbulence grid 10C_{ax} upstream of the test section. A more detailed description of the wind tunnel and flow conditioning elements can be found in (Lynch et al., 2011b). The mainstream temperature is measured 0.52C_{ax} upstream of the blades in the axial direction at multiple spanwise and pitchwise locations with a thermocouple rake. The mainstream temperature varies by no more than ± 0.6°C from the
average $T_x$. A Pitot probe, also inserted $0.5C_{ax}$ upstream, is used to measure the inlet mainstream velocity, $U_x$. The standard deviation over the mean $U_x$ is less than 1%.

The coolant passes through a desiccant drier and two heat exchangers that chill the coolant, before entering the plenum located below the endwall. A laminar flow element directly measures the total coolant flowrate, which is adjusted to achieve the necessary blowing ratios. The blowing ratios reported in this study, $M_{avg}$, are an average of the local $M$ at each film cooling hole. The local mass flow through each film cooling hole is weighted according to the local pressure drop across each hole. The uncertainty in coolant flowrate is estimated for a 95% confidence interval to be $\pm 3\%$, using sequential perturbation (Moffat, 1988). To measure the internal coolant temperature, $T_{c,in}$, two thermocouples are located approximately 8.7D below the impingement plate, which agree within $\pm 3^\circ$C. The mainstream to coolant temperature difference is about 40°C, which provides a coolant to mainstream density ratio of about 1.15.

The linear blade cascade contains seven blades based on the low-pressure turbine Pack-B airfoil, which has been used in many studies (Lake et al., 1999; Lawson et al., 2013; Lynch et al., 2011a, 2011b; Mensch et al., 2014; Mensch & Thole, 2014; Praisner et al., 2008). Although heat transfer and deposition are most severe in the high-pressure stages, a low-pressure turbine blade profile captures the relevant passage flow effects of pressure gradient and curvature. In Figure 8.2b, the green section of the endwall in the passage is constructed from Corian®, while outside the passage the endwall is constructed from medium density fiberboard. Passage 2, film cooling only, and passage 3, film and impingement cooling, are used for the experiments in this study. A summary of the airfoil geometric parameters and mainstream flow conditions is provided in Table 8.2. The test section inlet boundary layer parameters, were measured at $2.85C_{ax}$ upstream of the center blade previously by Lynch et al. (2011b). At this location upstream, the boundary layer thickness/span, $\delta/S$, was 0.061, and the freestream turbulence was 6% (Lynch et al., 2011b). Periodicity of the cascade is confirmed by taking measurements of the pressure distribution at the midspan of all seven blades and comparing to CFD predictions (Mensch et al., 2014).

The passage of the endwall is cooled with a generic configuration of internal impingement cooling and film cooling. Coolant is supplied by a plenum located beneath the endwall. In passages with impingement cooling, an impingement plate is located a gap height of $H = 2.9D$ below the endwall. The coolant enters the passage through film cooling holes in the
endwall, angled 30° to the surface. The locations of the 28 impingement holes and ten film cooling holes are shown in Figure 8.3. The spacing of the impingement holes is 4.65D in both the x and y directions, and two rows of film cooling holes are staggered between the impingement jets. The film cooling holes are oriented in the x-y plane to align with the local endwall streaklines found by Lynch et al. (2011b). Additional details regarding the cooling geometry can be found in Mensch and Thole (2014).

**Table 8.2 Flow Conditions and Blade Geometry**

<table>
<thead>
<tr>
<th>Scale factor</th>
<th>8.6</th>
<th>Inlet $U_\infty$</th>
<th>10.5 m/s</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{ax}$</td>
<td>0.218 m</td>
<td>Inlet, flow angle</td>
<td>35°</td>
</tr>
<tr>
<td>$p/C_{ax}$</td>
<td>0.826</td>
<td>Exit flow angle</td>
<td>60°</td>
</tr>
<tr>
<td>$S/C_{ax}$</td>
<td>2.50</td>
<td>Inlet Ma</td>
<td>0.029</td>
</tr>
<tr>
<td>Inlet Re</td>
<td>$1.22 \times 10^5$</td>
<td>Exit Ma</td>
<td>0.047</td>
</tr>
<tr>
<td>Exit Re</td>
<td>$1.98 \times 10^5$</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

![Figure 8.3 Schematic of internal and external cooling scheme from the top view, also showing the area average outline and locations of internal thermocouples.](image)

The deposition is dynamically simulated with a molten wax spray technique designed by Lawson and Thole (2012a). The two-nozzle wax injection system depicted in Figure 8.4 is
integrated into the turbulence grid upstream of the test section. The nozzle heads, located at one third span, contain a center jet for liquid wax and two air jets directed at the liquid stream to atomize the liquid into a mist of particles. Separate pressure regulators are used to independently control the liquid wax flowrate and the atomizing air pressure. The liquid flowrate for each nozzle is set to 1.9 g/s for an estimated particle loading of 0.8 ppm (570 ppmw). The air pressure is set to 138 kPa (20 psi) to generate particles with a median size of 34 μm, as discussed in (Lawson et al., 2013), achieving a median $St_kD$ of 6. The liquid wax and air lines are heated such that the injection temperature, measured by a thermocouple in the nozzle, is about 84°C. To achieve a TSP of 0.3, the mainstream temperature during deposition, $T_\infty$, is 47°C. The deposition is performed once steady state is reached for these thermal conditions as well as the mainstream and coolant flow conditions of $Re$ and $M_{avg}$. The wax spray lasts more than four minutes, which is beyond the equilibrium point identified by Lawson and Thole (2012a). Following the deposition, photographs are taken of the wax deposition, and a coat of flat black spray paint is applied for the thermal measurements in the second phase of the experiment. The thermal resistance of the paint is estimated to be less than 5% of the thermal resistance of the wax or the thermal resistance of the endwall.

![Figure 8.4 Schematic of two-nozzle wax injection system located in the turbulence grid.](image)

In the second phase of the experiment, the same mainstream $Re$, coolant flowrate, and $\Delta T$ of 40°C are generated for steady state measurements of external wax temperature, $\omega$. These measurements are made with infrared (IR) thermography using a FLIR P20 IR camera. Using IR imaging to obtain the temperature distribution on the surface allows for a high spatial resolution.
of 5.7 pixels/D. Additional details regarding the IR measurement technique and calibration are available in Mensch and Thole (2014). The uncertainty in the external effectiveness is estimated to be ± 0.02 for a 95% confidence interval.

Because access to the outer surface of the endwall is obstructed by the wax deposition, measurements of $\phi$ cannot be obtained with IR imaging, and thermocouples on the outer surface of the endwall would affect the deposition and the flow. Instead, the internal wall temperature, scaled to $\phi_i$, can be used to assess the endwall cooling performance after deposition at discrete locations in the passage. To determine $\phi_i$, four flat ribbon-type thermocouples are attached to the internal surface of the endwall with a high conductivity epoxy, $k = 4.3 \text{ W/m-K}$, at the locations shown in red in Figure 8.3. Using a 95% confidence interval, the uncertainty in $\phi_i$ is estimated to be ± 0.01 since temperature measurements are made directly with thermocouples.

8.5 Results and Discussion

The effect of simulated contaminant deposition is examined through measurements of external wax temperature, used to compute wax effectiveness ($\omega$), and internal endwall surface temperature, used to compute internal overall effectiveness ($\phi_{i,\text{dep}}$). The experiments considered the cooling configurations of film cooling only and combined film plus impingement for blowing ratios of 0.6 and 1.0.

8.5.1 Effects of Deposition on External Temperatures

Figure 8.5 - Figure 8.8 compare the external endwall overall effectiveness before deposition, $\phi$, to the external wax effectiveness after deposition, $\omega$. The film cooling holes and plenum boundaries are shown on the contours for reference. The horizontal lines just below each blade indicate a wall in the plenum and in the impingement channel that prevents coolant from crossing from one passage to another. Below the $\omega$ contours are photographs of the deposition patterns on the endwall prior to the application of black paint for thermal measurements. The pitchwise laterally averaged $\phi$ and $\omega$ are compared in Figure 8.7, which is plotted as a function of the axial direction, $x/C_{ax}$. The area averaged $\phi$ and $\omega$ are plotted in Figure 8.8 along with the average $\phi_i$ and $\phi_{i,\text{dep}}$, which are discussed in the next section. The area used for averaging the external effectiveness is shown in Figure 8.3.
Film cooling only (no internal impingement) results are shown in Figure 8.5a for $M_{avg} = 0.6$ and 5b for $M_{avg} = 1.0$. As the can be seen in the photographs, the deposition generates roughness elements with the size of 0.2 – 0.5D. The deposition is relatively uniform on the endwall, except near the pressure side of the passage. Particles with a $St_{C_{ax}}$ greater than one account for 8% of the total particle mass from the particle size distribution measurements reported in Lawson et al. (2013). These larger particles are carried into the pressure side instead of turning with the flow. Toward the upstream portion of the pressure side there are mostly small deposits, and toward the downstream half of the pressure side there are more large deposits. The effect of blowing ratio can be seen by comparing the photographs of the film cooling holes in Figure 8.5a to b. Some deposits form around the edges of the holes for $M_{avg} = 0.6$, but at $M_{avg} = 1.0$, the coolant jets prevent deposits around the hole edges and downstream of the hole for about 1D. Farther downstream of the first row of film cooling holes in Figure 8.5b, it appears there is increased deposition from vortices induced by the film cooling jets bringing particles down to the wall. The deposition downstream of the film cooling holes is consistent with the deposition and flow visualization reported by Lawson and Thole (2012b). In Figure 8.5b there is slightly less deposition just upstream of the first row of film cooling holes compared to farther upstream. As the $\phi_f$ contours on the left of Figure 8.5b show, the in-hole convection cools the endwall just upstream of the film cooling holes, and increased cooling prevents deposition, as also reported in Albert and Bogard (2013a).

The effectiveness contours and deposition photographs for combined film and impingement cooling are shown in Figure 8.6. The deposition photographs in Figure 8.6 have similar features as Figure 8.5. There are increased levels of deposition near the pressure side, with smaller deposits concentrated toward the upstream pressure side, and larger particles concentrated toward the downstream pressure side and into the wake. Again, the higher blowing ratio mitigates some deposition just downstream of the film cooling holes, but slightly increases deposition farther downstream. It also appears that for $M_{avg} = 1.0$, there are fewer large deposits across the entire area cooled by impingement cooling, indicating that the local temperature of the endwall has an effect on the accumulation of deposition. The height of deposition was observed to correlate with surface temperature in Albert and Bogard (2013a) with larger thicker deposits for higher surface temperatures.
Figure 8.5 Film cooling only contours of (a) \( \phi_f \) without deposition for \( M_{\text{avg}} = 0.6 \), (b) wax effectiveness, \( \omega_f \), and deposition photographs for \( M_{\text{avg}} = 0.6 \), (c) \( \phi_f \) for \( M_{\text{avg}} = 1.0 \), and (d) \( \omega_f \), and deposition photographs for \( M_{\text{avg}} = 1.0 \).

Figure 8.6 Film and impingement cooling contours (a) \( \phi \) without deposition for \( M_{\text{avg}} = 0.6 \), (b) wax effectiveness, \( \omega \), and deposition photographs for \( M_{\text{avg}} = 0.6 \), (c) \( \phi \) for \( M_{\text{avg}} = 1.0 \), and (d) \( \omega \), and deposition photographs for \( M_{\text{avg}} = 1.0 \).
Figure 8.7 Laterally averaged overall effectiveness without deposition, $\phi$, and wax effectiveness, $\omega$, across the passage for $M_{\text{avg}} = 0.6$ and 1.0 for (a) film cooling only, and (b) film and impingement.

The contours in Figure 8.5 and the laterally averaged data in Figure 8.7a can be used to compare the external effectiveness levels for film cooling only. For both blowing ratios, the temperatures measured on the deposition surface are the same or warmer than the temperatures measured on the endwall before deposition. The $\omega_f$ contours are also not as smooth as the $\phi_f$ contours, reflecting the roughness elements of the deposition. The roughness appears to degrade the performance of the film cooling jets, since the distances of the jet footprints on the surface are reduced. The roughness increases coolant jet mixing with the mainstream and degrades the effectiveness of the coolant as a protective barrier from the mainstream. The effect is more apparent when the pre-deposition coolant jets are attached, as with $M_{\text{avg}} = 0.6$. Also, the mitigation of deposition just downstream of the film cooling holes allows the film coolant to be more effective at $M_{\text{avg}} = 1.0$ after deposition.

The effectiveness levels for film and impingement cooling with and without deposition are compared in Figure 8.6, Figure 8.7b, and Figure 8.8. The $\omega$ is significantly less than $\phi$ for both blowing ratios, meaning the external wax temperatures are warmer than the endwall temperatures without deposition. The deposition causes a greater reduction in effectiveness for the case of combined film and impingement cooling compared to the case of film cooling only. The combined impingement and film cooling performance is degraded due to roughness induced coolant jet mixing like the case of film cooling only discussed earlier. An additional reason $\omega$ is
less than $\phi$ is because the deposition has an insulating effect on the endwall. Although the layer of deposition is thin and not uniform, $k_{\text{dep}}$ is much less than $k_w$, and the estimated conduction thermal resistance of the deposition is on the same order of magnitude as the conduction resistance of the endwall. The conduction resistances are also on the same order as the internal convection resistance for impingement cooling. Without impingement, the internal convection resistance is much higher than either of the conduction resistances. Before deposition, film cooling only has lower rates of overall heat transfer compared to combined film and impingement. Therefore without impingement, the insulating effect of the deposition is less significant. Because the deposition acts as an insulating layer to the impingement cooled endwall, the wax temperature is warmer than the endwall under the wax ($\omega < \phi_{\text{dep}}$). This conclusion however does not reveal whether the external surface of the endwall itself is cooler or warmer after deposition.

Figure 8.8 Area average overall effectiveness without deposition, $\phi$, area average wax effectiveness $\omega$, and average internal effectiveness with and without deposition, $\phi_i$, $\phi_{i,\text{dep}}$, for film cooling only and combined film and impingement at different blowing ratios.
8.5.2 Effect of Deposition on Internal Temperatures

Because measurement access to the external endwall temperatures was prevented by the deposition, the endwall cooling performance could not be directly calculated; however, the internal endwall effectiveness, $\phi_i$ and $\phi_{i,dep}$, can be directly compared with measurements on the inner side of the endwall. The average internal effectiveness from the four locations shown in Figure 8.3 is plotted in Figure 8.8 along with the area averaged $\phi$ and $\omega$ for comparison. If the overall endwall heat transfer is unchanged before and after deposition, the presence of the deposition layer would insulate the endwall temperatures from the mainstream, and $\phi_{i,dep}$ (with deposition) would be higher than $\phi_i$ (without deposition). Although the wax does act as an insulator, $\phi_{i,dep}$ is lower than $\phi_i$ for all four cases by 0.01 – 0.03. This result means that the overall heat transfer must increase with deposition. An increase in the overall heat transfer can be attributed to roughness effects, which degrade the performance of the film cooling and increase the external heat transfer coefficients. It is worth noting that if the internal wall temperatures are warmer with deposition for both cooling configurations, it is expected that the external wall temperatures are warmer with deposition also.

To understand the effect of deposition roughness on the external heat transfer coefficients, $h_{\infty,dep}$, the $h_{\infty,dep}$ are estimated with a simple 1-D analysis using the measured temperatures. The location closest to the pressure side in Figure 8.3 is chosen for this analysis, since the external driving temperature there is not affected by film cooling. The heat transfer analysis at this location uses $T_{\infty}$ as the external driving temperature. The heat transfer is assumed to be primarily in the direction through the thickness of the endwall. Although the lateral surface temperature gradients and endwall vertical temperature gradient are approximately equal at this location in the passage, the lateral heat flux is limited by the lower thermal conductivity of the wax. Equating the external convective heat transfer without film cooling to the internal convective heat transfer results in the following equation for $h_{\infty,dep}$.

$$h_{\infty,dep} = h_i \left( 1 - \frac{\phi_{i,dep}}{\omega} \right) \quad (8.6)$$

The quantities of $\phi_{i,dep}$ and $\omega$ are directly measured in the experiment. The internal heat transfer coefficient, $h_i$, is unchanged from the experiments without deposition. Therefore, $h_i$ can be calculated from the internal and external endwall temperature measurements without deposition using Equation (8.7).
The calculation of $h_{\infty,\text{dep}}$ at the pressure side location is compared to the local $h_{\infty}$ measured by Lynch et al. (2011b), which is 41 W/m$^2$-K. The estimated $h_{\infty,\text{dep}}$ ranges from 54 – 81 W/m$^2$-K (30 – 100% increase) for the different cooling configurations and blowing ratios. The variation in predictions does not correlate with either the cooling configuration or the blowing ratio, but reflects the variation in the deposition surface as well as the uncertainty in the one-dimensional analysis and the measurements.

\[ h_i = \frac{k_w (\phi_i - \phi)}{t_w (1 - \phi_i)} \]  \hspace{1cm} (8.7)

8.6 Conclusions

Molten particle deposition was experimentally simulated on the endwall of a gas turbine blade cascade. The experiments matched the relevant thermal and flow parameters that influence the deposition behavior. The cooling performance of the endwall with deposition was compared to the clean endwall through measurements of wax effectiveness, $\omega$, endwall overall effectiveness without deposition, $\phi$, and endwall internal effectiveness with and without deposition, $\phi_{i,\text{dep}}$ and $\phi_i$. The cases included two different cooling configurations: film cooling only and combined film and impingement cooling, as well as different blowing ratios. For the higher blowing ratio, $M_{\text{avg}} = 1.0$, some mitigation of deposition was observed at the film cooling hole exits and in the areas cooled by in-hole convection and impingement. This result indicates that the correct flow and thermal boundary conditions are needed to accurately simulate the deposition itself. In addition, an effective strategy to mitigate deposition may be to increase or improve turbine cooling.

In addition to the thermal behavior of the endwall affecting the amount of deposition, the deposition affects the thermal performance of the endwall. Decreased effectiveness was measured after deposition for $\omega$ compared to $\phi$, and $\phi_{i,\text{dep}}$ compared to $\phi_i$. The deposition added an insulating layer to the endwall but also degraded the performance of the cooling systems. The reasons for higher endwall temperatures with deposition are attributed to two roughness effects. There is a reduction in the film cooling performance because of the additional mixing that occurs from the roughness of the surface. Second, the roughness increases the external heat transfer coefficients, which increases the overall endwall heat transfer.
CHAPTER 9. CONCLUSIONS

Conjugate heat transfer for a gas turbine endwall was studied using properly scaled experiments and simulations. This dissertation demonstrated the conjugate methodology for endwall heat transfer, and the effects of cooling configuration, endwall contouring, a thermal barrier coating, and particle deposition. The flowfield measurements and the computational simulations revealed the previously unstudied interactions between the passage vortex, film cooling, and endwall heat transfer. The thermal and flow measurements and comparisons to predicted conjugate results fill a gap in the literature data for gas turbine endwalls and the secondary flow effects.

The relative influence of impingement and film cooling for the flat endwall base case was compared with properly scaled conjugate experiments. With regards to the overall effectiveness, film cooling was most effective around the film cooling holes due to in-hole convection. The film cooling jet attachment was generally poor at high blowing ratios, thereby not providing a significant cooling benefit relative to impingement cooling. The dominant factor in endwall cooling was from impingement, which generated a uniform distribution of overall effectiveness and increased with coolant flowrate. The combined film and impingement cooling case demonstrated high effectiveness from in-hole convection as well as uniformly high effectiveness across the impingement array. The measured overall effectiveness was compared to the calculated overall effectiveness based on a one-dimensional heat transfer analysis of the individual cooling features. Reasonable comparisons were achieved although the analysis necessarily simplified the heat transfer with several assumptions including negligible three-dimensional and lateral temperature gradients. The flat endwall conjugate experiments showed the value of investigating internal and external cooling schemes together to capture conjugate effects.

The effect of varying the impingement channel height for the case with the flat endwall indicated that reducing the channel height resulted in reduced overall effectiveness levels. The reduction was more significant for the case of impingement only than for combined film and impingement. Computational predictions of conjugate heat transfer provided additional results for a wider range of impingement channel heights. The simulations showed a small peak in overall effectiveness at the channel height to cooling hole diameter ratio of 2.9 due to changes in the impingement channel flow for different channel heights. The small channel heights restricted
the spreading of the cooling outside the area above the impingement array. The larger channel heights had slower impingement jet velocities leading to a more three-dimensional channel flow. The small channel heights had the highest effectiveness directly above the jets, but the larger channel heights had higher effectiveness in the regions of the endwall adjacent to the impingement jets. On an area-averaged basis, the overall effectiveness with combined film and impingement cooling was not very sensitive to the impingement channel height.

Non-axisymmetric endwall contouring affected the endwall conjugate heat transfer through changes to both the internal and external flow. A varying impingement channel height inherently resulted for the contoured endwall design. For impingement cooling only the contoured endwall overall effectiveness was decreased compared to the flat endwall due the varying impingement height as well as increased external heat transfer coefficients in the upstream half of the passage. When film cooling was included for the contoured endwall the increased heat transfer coefficients did not have a detrimental effect on the overall effectiveness because the presence of film coolant above the endwall reduced the effective external driving temperature. The film cooling jets also had better attachment to the surface where the local surface gradient generated a local favorable pressure gradient. Film cooling effectiveness was also improved with contouring because the weaker passage vortex, relative to the flat endwall, generated less mixing of the coolant into the mainstream. External heat transfer coefficients and film cooling attachment were affected by endwall contouring. However, these effects combined to result in unchanged laterally averaged overall effectiveness between the flat endwall and the contoured endwall with combined film and impingement.

Near wall secondary flows, such as the passage vortex, affected endwall heat transfer by increasing coolant and mainstream mixing. When the film cooling jets were detached from the endwall surface, increased mixing from the passage vortex improved overall effectiveness. The overall effectiveness measurements showed that the overall effectiveness of the endwall near the trailing edge was slightly increased for the flat endwall compared to the contoured endwall with a weaker passage vortex.

Measurements of the passage secondary flows showed that the passage vortex size, location and strength were influenced by endwall contouring as well as by film cooling. In agreement with previous studies, endwall contouring reduced the size and strength of the passage vortex by reducing the secondary velocities and turbulent kinetic energy. A more compact
passage vortex resulted in increased vorticity and turbulent kinetic energy measured for the counter rotating vortex above the passage vortex. Although the effect of passage film cooling on secondary flows was generally neglected in previous studies, the measurements in the current study showed that film cooling increased the vorticity, turbulent kinetic energy and secondary velocities of the passage vortex for both the flat and contoured endwalls. The kinetic energy of the film cooling jets was transferred to the passage vortex. Film cooling also increased the size of the passage vortex, moving the center of rotation farther away from the endwall and farther away from the blade compared to without film cooling. As a result, the counter rotating vortex diminished as film cooling blowing ratio increased. The contoured endwall with a blowing ratio of two produced similar levels of turbulent kinetic energy and secondary velocities as the flat endwall without film cooling.

The cooling performance of a conjugate endwall with a thermal barrier coating was quantified with measurements and predictions of endwall overall effectiveness and TBC effectiveness. The improvement in effectiveness for the flat endwall with TBC was found to be more significant than the effectiveness improvement achieved by increasing blowing ratio. The TBC accomplished a substantial increase in the effectiveness by insulating the endwall from external heating and reducing the endwall surface temperature. A larger improvement in overall effectiveness was achieved as blowing ratio increased because the insulating effect of the TBC allowed the internal impingement cooling to be more effective. Due to the insulating effect of the TBC, the outer TBC temperature was warmer than the endwall without TBC, which caused TBC effectiveness data to resemble adiabatic effectiveness.

Computational predictions of conjugate heat transfer were compared to the experimental results for both the flat and contoured endwalls. Generally there was good agreement between the predictions and the measurements except for the common failure to accurately predict film cooling jet attachment and a 5% under-prediction of effectiveness downstream of the cooling area. The conjugate simulations with TBC in predicted the correct trends across the passage for both the endwall and the TBC surface. Overall, the conjugate heat transfer simulations in conjunction with a suitable unstructured grid provided reasonable temperature predictions that were used to analyze the endwall heat transfer.

Molten particle deposition was experimentally simulated on the flat endwall with impingement and film cooling. Both film cooling and internal cooling were found to locally
reduce the amount of deposition by reducing the endwall temperature which decreased the sticking probability of the particles. Increased cooling may be an effective deposition mitigation strategy, and the correct flow and thermal boundary conditions are necessary to accurately simulate the deposition process. In addition to the thermal behavior of the endwall affecting the amount of deposition, the deposition affected the thermal performance of the endwall. The deposition reduced effectiveness on both the external and internal endwall surfaces. Although the deposition added an insulating layer to the endwall, it also degraded the cooling performance through roughness effects. The rough deposition surface increased external heat transfer coefficients and increased mixing of the film cooling jets.

9.1 Recommendations for the Turbine Designer

The various trends found from the experimental and computational studies of this dissertation lead to the following recommended considerations in the design of gas turbine endwall cooling configurations to optimize performance.

1. Based on the comparisons between the cooling configurations, using only impingement cooling is possible where the external heat transfer coefficients are low, such as in the upstream portions of the flat endwall. For the flat endwall, the overall effectiveness with impingement only was not much lower than the overall effectiveness with combined film and impingement. The coolant could be released through film cooling in a different region of the endwall to increase the residence time of the coolant in the impingement channel.

2. Film cooling should be used in regions where the heat transfer coefficients are high, such as the pressure side of the flat endwall and the suction side trailing edge where the passage vortex increases heat transfer. Even if the film cooling jets detach from the surface, the presence of film cooling reduces heat transfer by decreasing the local driving temperature for external convection. For the contoured endwall, impingement cooling could be strategically located on the upstream portions of hills where a positive surface gradient thins the boundary layer and increases heat transfer. These regions are also more likely to maintain film cooling attachment due to the favorable pressure gradient.
3. The distance between the impingement plate and the endwall (height of the impingement channel) is not as significant of a factor to area averaged overall effectiveness as the method of coolant extraction from the impingement channel. If the coolant is extracted through film cooling holes in the vicinity of the impingement jets, the residence time of the coolant within the channel is less than if the coolant is forced to flow through the channel to be extracted at a different endwall location. A smaller residence time of the coolant reduces the coolants potential to pull heat from the internal side of the endwall.

4. Endwall contouring has the potential to reduce aerodynamic pressure losses, reduce average endwall heat transfer, improve film cooling effectiveness, and mitigate the size of the passage vortex. The increase in the turbulent kinetic energy and secondary velocities from adding film cooling to the flat endwall can be reduced back to the levels without film cooling by using endwall contouring.

5. A film cooling blowing ratio of one is recommended based on multiple performance considerations of overall effectiveness, cost in terms of pressure ratio, and influence to passage secondary flow. Although overall effectiveness increases with blowing ratio, the increase from $M = 0.6$ to $1.0$ was greater than the increase from $M = 1.0$ to $2.0$. The required increase in pressure drop from $M = 0.6$ to $1.0$ was also less than the increase in pressure drop from $M = 1.0$ to $2.0$. Additionally, $M = 1.0$ would result in fewer passage aerodynamic losses by generating a smaller passage vortex compared to $M = 2.0$.

6. A thermal barrier coating achieves a substantial improvement in overall effectiveness across the endwall surface. The improvement in area-averaged overall effectiveness slightly increases with coolant blowing ratio, but is approximately 0.15.

7. In situations when the turbine is at risk for molten particle deposition, both internal cooling and film cooling have the ability to reduce the amount of deposition by reducing the potential for particles to stick to the surface. Additionally, internal cooling can counteract the increased heat transfer from increased surface roughness with deposition.
9.2 Recommendations for Future Work

The conjugate heat transfer scaling in this dissertation established that the endwall overall effectiveness depended on non-dimensional parameters, such as the Bi, ratio of heat transfer coefficients, and geometric scaling, as given in Equation (1.1). However, this dependence has not been experimentally validated for cases with distinctly varying conditions, such as laboratory experiments and engine conditions. Additionally, the conjugate heat transfer analysis should be compared against metal temperature predictions obtained from a decoupled heat transfer analysis using adiabatic effectiveness and heat transfer coefficients. The performance of the conjugate heat transfer analysis and the decoupled analysis may vary across the endwall or for different engine conditions. It would be valuable to understand when each type of model succeeds and fails to inform the required complexity of future endwall heat transfer studies.

Additional efforts could be made in computational modeling of the conjugate heat transfer comparisons with experimental results. The cases with film cooling only and impingement cooling only could be simulated using a similar approach to generate the computational meshes, and specify the boundary conditions and thermal coupling of the solid and fluid boundaries. The film cooling only data would provide spatially resolved internal endwall temperatures as well as internal heat transfer coefficients to compare to the calculations and assumptions from the one-dimensional analyses. The impingement cooling only results would reveal information about the flow path of the impingement jets with lateral extraction. The trend of higher internal heat transfer coefficients for impingement only versus combined film and impingement could be justified from additional flow and temperature data within the impingement channel. The external heat transfer coefficients predicted by the conjugate simulations could also be compared between cases with and without film cooling to measure the heat transfer augmentation of film cooling. The conjugate predictions of heat transfer coefficient could also be compared to the heat transfer coefficients obtained for a constant heat flux endwall in experiments and computational simulations.

Although there was good agreement between the measurements and simulations, the computational predictions could be improved particularly in the prediction of the diffusion of the film cooling jets and attachment to the surface. Efforts to improve predictions could investigate the effects of local grid refinement, other turbulence models tuned to film cooling flows, more resolved turbulence methods such as LES or DNS, or other methods.
The passage flowfield measurements in this work were obtained using planar PIV, which provided the two in-plane velocity components. Due to the pitchwise gradients in the passage vortex, the data were sensitive to the orientation and precise location of the streamwise measurement planes. Also, the calculations of turbulent kinetic energy only reflected the two components of fluctuations available. Three components of velocity and fluctuations would be possible using a two-camera setup in Stereo-PIV. With a two-camera setup, it would be possible to access measurements in planes that are perpendicular to the blade surface. These planes would allow visualization of the cross section of the passage vortex at the same streamwise location, as well as measurements of all three velocity components.

Further analysis and experiments for an endwall with contaminant deposition could be completed. Chapter 8 showed that deposition decreased the internal overall effectiveness of the endwall by increasing heat transfer from surface roughness. However, on a vane with TBC, Davidson et al. (2014a) found that deposition slightly increased the overall effectiveness by adding an insulating layer to the external side of the TBC. It is likely that deposition is causing both increased heat transfer and insulation on the endwall and vane with TBC, but the relative importance of each is not understood. The different results could be caused by differences in the heat transfer with and without TBC, or the external heat transfer coefficients and the amount of deposition on the surface for the endwall and the vane. Additional experiments for an endwall with a TBC would improve understanding of the mechanisms. Detailed measurements could be made of the deposition distribution using three-dimensional scanners. This information would allow more detailed computational analysis and improvement of deposition models.
REFERENCES


Crawford, M. E. (2009). *TEXSTAN (academic version).* Austin, TX: University of Texas.


Pu, J., Yu, J., Wang, J., Yang, W., Zhang, Z., & Wang, L. (2014). An experimental investigation of secondary flow characteristics in a linear turbine cascade with upstream converging...


178
A.1 Thermal Barrier Coating Assembly

In experiments with TBC, sixteen ribbon type-thermocouples were attached to the endwall before the application of the cork layer to measure the temperature at the interface between the endwall and the cork layer. A high conductivity epoxy, \( k = 4.3 \text{ W/m-K} \), is used to attach the ribbon end of the thermocouple to the endwall. To allow the cork to lay as flat as possible over the thermocouples, 36 gauge wire is used, and the outer layer of clear insulation on the wires is removed. To prevent the wires from the endwall thermocouples from overlapping, a 64 \( \mu \text{m} \) thick Kapton\textsuperscript{®} tape with a silicon adhesive is applied as shown in Figure A.a. Additionally, five thermocouples are embedded in the cork layer for IR image calibration of the outer TBC surface. These thermocouples are bead-type, but the wires are treated the same manner as described for the ribbon thermocouples.

The cork is adhered to the endwall using two to three layers of DAP\textsuperscript{®} Weldwood\textsuperscript{®} Original Contact Cement. The contact cement provides a strong and even adhesion across the surface. Although the thermocouples are beneath the endwall surface, the top surface of the cork is smooth and level (as shown in Figure A.1b) due to the thin wire and layer precautions described above. The thermal resistance for the combined cork and adhesive layer, given in Table 7.1, was measured by Kistenmacher (2013) for the same cork and adhesive method. When the TBC is applied to the contoured endwall, slits are cut into the cork to allow the cork to lay uniformly over the non-developable contoured surface. The pattern for the slits cut into the cork was obtained from the contoured endwall constant heat flux heater design for the Pack-B endwall, which is discussed in Lynch (2011). The slight openings in the slits are filled with wood filler to smooth the surface and maintain a similar conductivity in the seams.
Figure A.1 Photographs of the assembly of endwall with TBC: (a) ribbon thermocouples adhered to the endwall with epoxy and Kapton® tape, and (b) cork layer adhered to the flat endwall with holes cut for the film cooling holes.

A.2 Particle Image Velocimetry Detailed Experimental Methods

The particle image velocimetry system arrangement is shown in Figure A.2. The front of 60 W dual cavity laser (30 W per cavity) is shown at the top of the schematic covered in plastic. The laser beam emerges from the laser head in the horizontal direction, where it enters a focusing lens, followed by a 10 mm focal length cylindrical lens, and a 90 degree mirror. The cylindrical lens expands the beam into a plane, and the mirror directs the plane downward onto the endwall. The approximate location of the laser beam relative to the blades and the flow is shown in Figure A.2. The DEHS tracer particles are generated using a Laskin nozzle aerosol generator (Raffel et al., 2007), which are injected into the mainstream flow upstream of the primary blower in the wind tunnel. The laser current required to sufficiently illuminate the particles is given in Table A.1 for each measurement plane (Figure 6.4). The 2 kHz CMOS camera is shown at the bottom left of Figure A.2 with a 50 mm lens. The camera is oriented perpendicular to the laser sheet such that the image on the sensor corresponds approximately to the dotted square in Figure A.2. The camera has 1024 x 1024 pixel resolution, which corresponds to an image scale of 4 – 5 pixels/mm, with the exact values given in Table A.1. The Δt between laser pulses is also shown in Table A.1. The required Δt was chosen based on an a priori estimate of the period of time for a particle to move ~10 pixels while traveling at the bulk
in-plane velocity. The actual bulk particle displacement is given in Table A.1. In the vector
calculation, the an adaptive multiple pass processing scheme is used where the interrogation
windows are progressively decreased for each pass (*DaVis 8.1.4*, 2012). With four passes in
total, the initial interrogation window is 64 x 64 pixels with 50% overlap, and the final
interrogation window is 24 x 24 pixels with 75% overlap. The length of the final interrogation
window in mm is given for each measurement plane in Table A.1.

![Diagram of laser and high speed camera setup](image)

**Figure A.2** Diagram of laser and high speed camera setup for the particle image
velocimetry measurements.

**Table A.1 PIV Setup and Calculation Parameters**

<table>
<thead>
<tr>
<th></th>
<th>Flat Endwall</th>
<th>Contoured Endwall</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Plane A</td>
<td>Plane B</td>
</tr>
<tr>
<td>Laser Current (A)</td>
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<td>28</td>
</tr>
<tr>
<td>Δt Between Images (μs)</td>
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<td>100</td>
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<tr>
<td>Image Scale (pixels/mm)</td>
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<td>Bulk Displacement (mm)</td>
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</tr>
<tr>
<td>Final Interrogation Window (mm)</td>
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<td>5.0</td>
</tr>
</tbody>
</table>
APPENDIX B. ONE- AND TWO-DIMENSIONAL HEAT TRANSFER CALCULATIONS

B.1 Derivation of Overall Effectiveness Equations

The derivation of Equation (1.1) for overall effectiveness as a function of $Bi$, $h_{\infty}/h_i$, and $\chi_\eta \eta$ is given below. The derivation is based on the one-dimensional heat transfer through a constant thickness wall with internal and external convection as shown in Figure 1.5. First, the external convective heat transfer is set equal to the overall heat transfer from $T_{aw}$ to $T_{c,in}$ in Equation (B.1).

$$ q_{\text{conv},e} = h_{\infty} (T_{aw} - T_w) = \frac{T_{aw} - T_{c,in}}{1/h_{\infty} + t/k_w + 1/h_i} = q_{\text{overall}} $$  \hspace{1cm} (B.1)

Multiplying both sides by $1/[h_{\infty} (T_{aw} - T_{c,in})]$ results in Equation (B.2), reported in Albert et al. (2004). The $Bi$ is defined according to $h_{\infty}$ as given in the nomenclature.

$$ \frac{T_{aw} - T_w}{T_{aw} - T_{c,in}} = \frac{1}{1 + Bi + h_{\infty}/h_i} $$  \hspace{1cm} (B.2)

$T_{\infty}$ is introduced to the temperature ratio on the left hand side of Equation (B.2) by adding and subtracting $T_{\infty}$ to the numerator. Rearranging leads to an expression of entirely dimensionless temperatures: $\frac{\phi - \chi_\eta \eta}{1 - \chi_\eta \eta}$, where $\phi$, $\chi_\eta$, and $\eta$ are defined in the nomenclature. The last step in deriving Equation (1.1) is to solve for $\phi$, which gives Equation (1.1).

A two-dimensional conduction analysis can show that the endwall thickness to width ratio is also important to the overall effectiveness for an endwall. The endwall thickness to width ratio is matched in the experiments, which use full geometric scaling. The two-dimensional analysis considers a two dimensional wall with adiabatic sides and known convective coefficients on the top and bottom edges as shown in Figure B.1. Film cooling is not included in the two-dimensional analysis for simplicity.
Figure B.1 Schematic of two-dimensional conduction in the endwall.

The steady two-dimensional conduction equation with no heat generation is given in terms of non-dimensional temperature at any location within the wall, \( \phi(x,y) \), in Equation (B.3), with the directions of \( x \) and \( y \) shown in Figure B.1. The corresponding \( \phi \) boundary conditions are also given in Equation (B.3).

\[
\begin{align*}
\frac{\partial^2 \phi}{\partial x^2} + \frac{\partial^2 \phi}{\partial y^2} &= 0, \\
\frac{\partial \phi}{\partial x} \bigg|_{x=0,c_{ax}} &= 0, \\
k \frac{\partial \phi}{\partial y} \bigg|_{y=0} &= h_\infty \phi \big|_{y=0}, \\
k \frac{\partial \phi}{\partial y} \bigg|_{y=t} &= h_i \left( t - \phi \big|_{y=t} \right)
\end{align*}
\]  

(B.3)

Equation (B.3) can be solved analytically using the method of separation of variables. It is assumed that the solution for \( \phi(x,y) \) is the product of two functions, \( \phi(x, y) = F(x)G(y) \).

After applying this form to Equation (B.3), the result is \( \frac{F''}{F} = -\frac{G''}{G} = -\beta^2 \), where \( \beta \) must be a constant. The family of solutions for \( F(x) \), given in Equation (B.4), can be found by applying the boundary conditions at \( x = 0 \) and \( C_{ax} \), where \( A \) is an unknown constant and \( n = 1, 2, 3 \ldots \)

\[
F(x) = A_n \cos(\beta_n x),
\]

\[
\beta_n = \frac{\pi(n-1)}{C_{ax}}
\]  

(B.4)

The solution for \( G(y) \), given in Equation (B.5), can be found by applying the homogeneous boundary condition at \( y = 0 \), where \( E \) is an unknown constant.
Combining $F(x)$ and $G(y)$ gives a family of solutions which are summed to give the final solution, given in Equation (B.6). The only remaining unknown constants are the particular $K_n$ values which are the product of the constants $A_n$ and $E_n$.

$$
\phi(x, y) = \sum_{n=1}^{\infty} K_n \cos(\beta_n x) \left[ \sinh(\beta_n y) + \frac{\beta_n t}{\text{Bi}} \cosh(\beta_n y) \right].
$$

(B.6)

By applying the final boundary condition at $y = t$, it can be shown that $K_n$ is a function of the Bi, $h_x/h_i$, $t/C_{ax}$, and $\beta_n t$. The product $\beta_n t$, which also appears in Equation (B.6) can be rewritten as $\pi(n - 1)t/C_{ax}$. The products $\beta_n x$ and $\beta_n y$ can be rewritten similarly, revealing the geometric ratios, $x/C_{ax}$ and $y/C_{ax}$. Therefore, the only additional non-dimensional groups in the two-dimensional analytical solution are geometric ratios, $t/C_{ax}$, $x/C_{ax}$ and $y/C_{ax}$, which are maintained when the geometry is properly scaled.

### B.2 One Dimensional Calculation of Internal Heat Transfer Coefficients

The average internal heat transfer coefficient, $\overline{h_i}$, is calculated for all three cooling arrangement using the assumption of one-dimensional heat transfer between the top of the endwall and the internal convection. The measurements of $\phi$ and $\phi_i$ at the four locations in Figure 4.4b are applied to Equation (4.3). The $\overline{h_i}$ are non-dimensionalized to Nusselt number based on $D$ and the thermal conductivity of the coolant. The coolant thermal conductivity is determined from a polynomial fit of the air property to temperature, and applying the measured internal coolant temperature, $T_{c,i}$. Figure B.2 shows the measured $\overline{\text{Nu}_{D,i}}$ for the three configurations at each blowing ratio in solid bars.

In Figure B.2, the film cooling only case produces the lowest $\overline{\text{Nu}_{D,i}}$, since the plenum providing the coolant flow is nearly stagnant due to the large cross plenum area relative to the film cooling holes. Natural convection is not expected due to the stable coolant temperature distribution in the plenum. For film cooling only, the measured $\overline{\text{Nu}_{D,i}}$ are compared to the $\text{Nu}_{D,i}$ calculated assuming a stagnant plenum with strictly conduction heat transfer in the air. Since the coolant temperature is measured at a distance $L_{c,in}$ below the endwall, the definition simplifies to $\text{Nu}_{D,i} = D/L_{c,in} = 0.11$ for all three blowing ratios. The measured $\overline{\text{Nu}_{D,i}}$ is higher than $D/L_{c,in}$.
because of the lateral flow that is induced by the flow entering the film cooling holes. This induced flow effect would also explain why the $\overline{\text{Nu}}_{D,i}$ for film cooling only also increases slightly with blowing ratio.

![Graph showing average Nusselt numbers based on internal wall temperature measurements for different geometries and flow conditions.](image)

**Figure B.2 Average Nusselt numbers based on internal wall temperature measurements for the different geometries and flow conditions.**

The highest $\overline{\text{Nu}}_{D,i}$ is measured for the case of impingement cooling only with lateral coolant extraction. In an impingement array with lateral coolant extraction, the amount of crossflow, or buildup of coolant, degrades the heat transfer coefficients compared to a single impingement jet (Florschuetz et al., 1981). However, crossflow is not expected to be significant in the impingement array tested because the estimated crossflow velocity ratio was estimated to be less than 9% of the jet velocity for all three blowing ratios. The impingement array in the current study only has an outlet on one side, but there is space on the pressure side and the downstream side of the channel for the impingement flow to exit the impingement array.

The case of combined film and impingement cooling is plotted in black in Figure B.2. Although the internal cooling scheme does not change from the case of impingement only, there is a large difference in measured $\overline{\text{Nu}}_{D,i}$. This result indicates that the method of coolant extraction influences the impingement flow and heat transfer. With combined film and impingement, the coolant is directed only into the ten film cooling holes. In contrast, the coolant
in the impingement only case is exhausted outside the impingement area after it has flowed through the channel. This difference in the path of the impingement flow results in lower heat transfer coefficients for the film cooling extraction case. The bars with diagonal stripes display the area averaged $\text{Nu}_{D,i}$ obtained from the computational predictions of combined film and impingement, which are in good agreement with the measured average $\text{Nu}_{D,i}$ as discussed in Chapter 4.

**B.3 Measurements of Internal Coolant Warming Factor**

The calculated overall effectiveness in Equation (3.6) is a function of the internal coolant warming factor, $\chi_f$, which corrects for the difference in normalizing coolant temperatures between $\phi$ and $\phi_f$. In Chapter 3, the coolant warming factor was estimated from a total energy balance on the case of impingement only case using Equation (3.7). Later, measurements of $T_{c,\text{inlet}}$ were obtained for the middle film cooling hole in both rows shown in Figure 4.4b. These values are averaged together and non-dimensionalized to obtain $\chi_f$, shown in Figure B.3 for film cooling only in blue and for combined film and impingement in black. The maximum deviation of any single measurement from the average shown is 0.05. Note that the expected value for film cooling only is $\chi_f = 1$ by definition, since it is assumed that $T_{c,\text{inlet}} = T_{c,\text{in}}$, and there is negligible heating of the coolant between these two measurement locations. The horizontal solid line across the graph denotes this value of 1. The measured values for film and impingement (solid black) are compared to the estimates from the energy balance (crosshatched black). The $\chi_f$ as predicted by the computational simulations is shown with the diagonally striped bar. The measurements of $\chi_f$ are very close to the predicted values in both cases of film cooling only and combined film and impingement. The good agreement between the measured, predicted, and assumed $\chi_f$ supports the validity of the assumed values for use in the analysis.
Figure B.3. Average measured coolant warming factors, $\chi_f$, for the cases of film cooling only and film and impingement cooling, compared to predictions from an energy balance on the case of impingement only.
APPENDIX C. CONTOURED ENDWALL EFFECTIVENESS MEASUREMENTS AND PREDICTIONS

C.1 Contoured Endwall Overall Effectiveness Spatially Resolved Measurements

The spatially resolved measurements for the contoured endwall with impingement and film cooling are given in Chapter 5. Below are the spatially resolved measurements for the contoured endwall with film cooling only and with impingement cooling only. Chapter 5 only provides these results on a laterally averaged basis.

C.1.1 Film Cooling Only Measurements

The overall effectiveness contours for the contoured endwall are shown for film cooling only in Figure C.1. These figures can be compared to the flat endwall $\phi_f$ contours given in Figure 3.7. The laterally averaged $\phi_f$ are shown in Figure 5.10b, which compares the contoured and flat endwall data. Generally, the results for the contoured endwall are similar to those for the flat endwall, but slightly higher. Compared to the flat endwall, the film cooling jets, appear to be more attached with endwall contouring. The better film jet attachment with contouring provides better downstream cooling than the corresponding jets on the flat endwall. The two film cooling jets closest to the suction side of the leading edge show the most change from the flat endwall. The reasons for the better film cooling performance with endwall contouring, discussed in Chapter 5, were related to locally higher $h_\infty$ with contouring compared to the flat endwall as well as the more favorable pressure gradient for the jets coming out of the valley.

C.1.2 Impingement Cooling Only Measurements

The contours of $\phi_o$ for the contoured endwall are shown in Figure C.2, compared to the flat endwall $\phi_o$ contours in Figure 3.8. Figure 5.10c plots the laterally averaged $\phi_o$ as a function of the axial distance across the passage for both flat and contoured endwall cases. Like the flat endwall, the effectiveness from impingement shows a significant increase with each increase in blowing ratio. However, the level of $\phi_o$ is considerably less with the contoured endwall compared to the flat endwall. Since the endwall thickness is uniform and the same as the flat endwall, the differences arise from the changes to the internal and external heat transfer.
coefficients with the contoured endwall as discussed in Chapter 5. Because the contoured endwall weakens the strength of the passage vortex, the cooling by the vortex downstream of the impingement area is diminished compared to the flat endwall case.

\[
\phi_I = \frac{T_\infty - T_{w,f}}{T_\infty - T_{c,in}}
\]

(a) Contoured
\[M_{avg} = 0.6, \ I_{avg} = 0.3\]
DR = 1.12

(b) Contoured
\[M_{avg} = 1.0, \ I_{avg} = 0.9\]
DR = 1.16

(c) Contoured
\[M_{avg} = 2.0, \ I_{avg} = 3.5\]
DR = 1.18

Figure C.1 Contoured endwall overall effectiveness measured for film cooling only (a) \(M_{avg} = 0.6\), (b) \(M_{avg} = 1.0\) and (c) \(M_{avg} = 2.0\).

\[
\phi_O = \frac{T_\infty - T_{w,o}}{T_\infty - T_{c,in}}
\]

(a) Contoured
\[M_{avg} = 0.6, \ I_{avg} = 0.3\]
DR = 1.12

(b) Contoured
\[M_{avg} = 1.0, \ I_{avg} = 0.9\]
DR = 1.15

(c) Contoured
\[M_{avg} = 2.0, \ I_{avg} = 3.5\]
DR = 1.17

Figure C.2 Contoured endwall overall effectiveness measured for impingement cooling only (a) \(M_{avg} = 0.6\), (b) \(M_{avg} = 1.0\) and (c) \(M_{avg} = 2.0\).
C.2 Contoured Endwall Overall Effectiveness With TBC

C.2.1 Film Cooling Only Measurements with TBC

The film cooling overall effectiveness with and without TBC along the two streamlines is plotted in Figure C.3a-f. The flat endwall data are in black, and the contoured endwall data are in red. The data for the PS streamline are in Figure B.1a-c, and for the SS streamline in Figure C.3d-f. The lines with open symbols are the continuous $\phi_f$ measurements without TBC, and the filled symbols are the discrete $\phi$ measurements with TBC obtained from the streamline thermocouples in Figure 7.4b. Without TBC, the endwall $\phi_f$ for the flat and contoured endwalls are almost the same on the PS, but have some differences on the SS. On the SS, the contoured $\phi_f$ is higher than the flat $\phi_f$ between $0.1 < s/C_{ax} < 0.6$ for $M_{avg} = 1.0$ and 2.0. This region of the SS streamline crosses the first row of film cooling holes and the jet paths. Therefore, the improvement with contouring shows the better attachment of these film cooling jets with contouring. For $M_{avg} = 0.6$, the film cooling jets are mostly attached for the flat endwall, so the contouring has almost the same performance as the flat endwall.

With TBC, the film cooling overall effectiveness improved for all cases. Again, the flat and contoured $\phi_{TBC}$ is very similar on the PS, but there are some differences on the SS from better film cooling attachment for the contoured endwall cases. However, $\phi_{TBC}$ is less affected by film cooling attachment than $\phi$ or $\tau$ because the TBC layer is between the film coolant and the $\phi_{TBC}$ measurement. Therefore, the differences between the flat and contoured endwall $\phi_{TBC}$ are expected to be smaller than for the flat and contoured endwall $\phi$ or $\tau$. Although there was no difference between the flat and contoured endwall $\phi$ for $M_{avg} = 0.6$, the contoured $\phi_{TBC}$ is slightly higher than the flat $\phi_{TBC}$ for $M_{avg} = 0.6$. The jet attachment for the flat endwall at this blowing ratio may be diminished when TBC is included because of the vertical cuts made in the TBC for the film cooling holes. In that case, endwall contouring can provide some improvement in the film cooling attachment over the flat endwall with TBC even at the lowest blowing ratio.
Figure C.3 Comparison of film cooling only overall effectiveness with and without TBC, for both flat and contoured endwalls, along inviscid streamlines, PS (a)–(c) and SS (d)–(f) (streamlines shown in Figure 7.4b).

C.2.2 Impingement Only Measurements with TBC

The impingement only overall effectiveness is also plotted along the streamlines in Figure C.4a-f. The flat endwall data are in black, and the contoured endwall data are in red. Figure C.4a-c is for the PS streamline, and Figure B.2d-f is for the SS streamline. The \( \phi_0 \) data without TBC are in the open symbols connected by the continuous lines. The contoured endwall \( \phi_0 \) is consistently lower than the flat endwall \( \phi_0 \), which is consistent with the observations from the contour plots and laterally averaged plots based on the trends with H/D. The reasons for the decrease in \( \phi_0 \) with contouring is because both \( h_i \) and \( h_c \) are reduced with contouring as discussed previously. The discrete filled symbols are the \( \phi_0 \) under the TBC for the flat and contoured cases. With TBC, the effectiveness is significantly higher than without TBC. The peak overall effectiveness increases by about 0.2 with TBC. Although the contoured \( \phi_0 \) was consistently lower than the flat \( \phi_0 \) without TBC, the differences between the flat and contoured
lines with TBC are not significant for any of the cases at either streamline. Therefore, the reduction in impingement effectiveness is not as significant as the improvement with TBC. Additionally, the effect of increased $h_\infty$ with contouring does not matter as much when TBC is added because the TBC provides insulation between the endwall temperature measurement and the mainstream convective heating.

**Figure C.4** Comparison of impingement only overall effectiveness with and without TBC, for both flat and contoured endwalls, along inviscid streamlines, PS (a)–(c) and SS (d)–(f).

Although the impingement only TBC effectiveness increases slightly for each increase in blowing ratio, the effect of internal cooling on the external TBC temperature is limited by the insulation of the TBC. Additionally, the contoured endwall $\tau_o$ is even less than the flat endwall $\tau_o$. This trend is of reduced impingement effectiveness with contouring is consistent with the results of contoured and flat $\phi_o$ without TBC, discussed in Chapter 5. The contoured effectiveness is consistently less than the flat endwall because the internal heat transfer coefficients are reduced, and because the external heat transfer coefficients are increased within the impingement array area. The $h_i$ is reduced from the flat endwall because the distance from the impingement plate varies with contouring, and values of H/D higher and lower than 2.9 have
decreased $h_i$. Additionally, the measured $h_i$ with contouring is 10-30% higher than the flat endwall for much of the impingement area (Lynch et al., 2011b) as shown in Figure 2.2b.

### C.2.3 Combined Film and Impingement Results with TBC

Computational predictions for the contoured endwall overall effectiveness data with combined film and impingement cooling with TBC are shown in Figure C.5. Elevation lines of the contoured endwall height, the cooling holes and the plenum boundaries are overlaid. Comparing Figure C.5 to the corresponding flat endwall predictions in Figure 7.6 and Figure 7.7, there are not many differences. The contoured endwall is slightly warmer than the flat endwall downstream in Figure C.5a. However, the contoured endwall is approximately 5% cooler than the flat endwall in the valley of the contour in the upstream half toward the suction side. Since the TBC is insulating the endwall from most of the effects of film cooling, the cooler endwall is due to locally improved impingement cooling. Therefore, the contoured endwall impingement effectiveness can actually improve locally compared to the flat endwall, when the coolant is exhausted through film cooling holes.

$$
\phi_{TBC} = \frac{T_\infty - T_w}{T_\infty - T_{c,in}}
$$

![Figure C.5 Predicted contoured endwall overall effectiveness with TBC for (a) $M_{avg} = 1.0$ and (b) $M_{avg} = 2.0$.](image)

The measured overall effectiveness for the contoured endwall with combined film and impingement cooling is compared to the flat endwall along the PS and SS streamlines in Figure C.6a-f. The contoured endwall measurements are in red, and the corresponding flat endwall
results are shown in black. The data for the PS streamline are in Figure C.6a-c, and SS streamline in Figure C.6d-f. The solid lines and open symbols are the continuous $\phi$ measurements without TBC, and the filled symbols are the discrete $\phi_{TBC}$ measurements obtained from the thermocouples under the TBC, shown in Figure 7.4b. Without TBC the endwall $\phi$ for the flat and contoured endwalls are almost the same as discussed previously. With TBC, the flat and contoured $\phi_{TBC}$ are also similar on the SS, but larger differences are observed between the flat and contoured $\phi_{TBC}$ on the PS in Figure C.6a-c. Upstream of the first row of film cooling holes, which have a peak around $s/C_{ax} = 0.1$, the contoured endwall $\phi_{TBC}$ is less than the flat endwall. Upstream of the film cooling hole exits, impingement and in-hole convection are the only cooling mechanisms. In addition, this upstream pressure side region of the contoured endwall passage has increased $h_\infty$ over the flat endwall by up to 30%, as shown in Figure 2.2b (Lynch et al., 2011b). Although the TBC is insulating the endwall to some extent, locally increased $h_\infty$ upstream of film cooling increases heating from the mainstream. Downstream of the film cooling holes on the PS ($s/C_{ax} > 0.2$) the opposite trend is found; the contoured endwall $\phi_{TBC}$ is greater than the flat endwall. This local increase with contouring is aligned with the region of higher effectiveness in the predictions previously discussed. In this downstream pressure side region of the contoured endwall passage, the $h_\infty$ is 10-30% less than the $h_\infty$ for the flat endwall, from Figure 2.2b. Locally reduced $h_\infty$ helps to further protect the endwall from the mainstream heating. The changes in $h_\infty$ from the flat to contoured endwall along the SS streamline are not as significant, ranging from about -10% to 10%, from Figure 2.2b.
Figure C.6 Comparison of overall effectiveness with and without TBC, for both flat and contoured endwalls, along inviscid streamlines, PS a–c and SS d–f.

C.3 Contoured Endwall TBC Effectiveness

TBC effectiveness, $\tau$, for the contoured endwall with film and impingement cooling is shown for measurements in Figure C.7a-c and predictions in Figure C.7d-e. Just like the flat endwall, the effects of in-hole convection and impingement cooling are more limited for $\tau$ compared to $\phi$. In areas upstream of the film cooling hole exits, the $\tau$ for the contoured endwall is even less than $\tau$ for the flat endwall. This trend between the flat and contoured endwalls is consistent with the $\phi_{TBC}$ results in Figure C.6a-c for the upstream portion of the PS streamline. High values of $\tau$ are found in areas where there is film cooling attachment. Compared to the flat endwall, the contoured endwall has better attachment of the film cooling jets at the two highest blowing ratios. This finding is also consistent with the $\phi$ results for the contoured endwall without TBC in Figure 5.9. The predictions in Figure C.7d and e show a similar extent of film cooling attachment as the measurements. However, the effect of film cooling is slightly over-
predicted for $M_{\text{avg}} = 1.0$ and under-predicted for $M_{\text{avg}} = 2.0$, similar to the flat endwall cases in Figure 7.11. Because of the reduced impact of the internal cooling and the increased film cooling attachment with the contoured endwall, there is not much change in the contoured endwall $\tau$ as blowing ratio increases.

Figure C.7 Contoured endwall TBC effectiveness, $\tau$, with internal impingement plus film cooling, for: (a) $M_{\text{avg}} = 0.6$ measured, (b) $M_{\text{avg}} = 1.0$ measured, (c) $M_{\text{avg}} = 2.0$ measured, (d) $M_{\text{avg}} = 1.0$ predicted, and (e) $M_{\text{avg}} = 2.0$ predicted.

C.4 Conclusions

Measurements and predictions of the TBC effectiveness and overall effectiveness with TBC were also completed for the contoured endwall for all of the cooling arrangements. Like the comparisons between the flat and contoured endwall without TBC, there was improved film cooling attachment with contouring, and decreased impingement effectiveness with contouring.
because the contouring reduced internal heat transfer and increased external heat transfer. Although the TBC effectiveness of the flat and contoured endwalls behaved as expected based on the overall effectiveness without TBC, the overall effectiveness with TBC had some differences. With TBC, the impingement only overall effectiveness was not reduced with contouring, but was the same between the flat and contoured endwalls. Because the TBC insulates the endwall from the external effects, the local increases in external heat transfer with contouring are mitigated with TBC. Additionally the improvement with TBC is much more significant than the difference between the flat and contoured overall effectiveness without TBC.
APPENDIX D. UNCERTAINTY ANALYSIS

D.1 Uncertainty in Overall Effectiveness

The uncertainty in overall effectiveness was calculated according to the propagation of uncertainty equation, which uses the partial derivatives as the sensitivity coefficients with respect to the measured quantities (Moffat, 1988). Equation (1.1) was used to determine the partial derivatives of \( \phi \) with respect to the measured temperatures, \( T_\infty \), \( T_{c,in} \), and \( T_w \). Equation (D.1) shows the uncertainty calculation for \( \phi \) as well as analytical expressions for the partial derivatives, where \( \delta \) represents the uncertainty in a quantity.

\[
\delta \phi = \sqrt{\left( \frac{\partial \phi}{\partial T_\infty} \delta T_\infty \right)^2 + \left( \frac{\partial \phi}{\partial T_{c,in}} \delta T_{c,in} \right)^2 + \left( \frac{\partial \phi}{\partial T_w} \delta T_w \right)^2}
\]

\[
\frac{\partial \phi}{\partial T_\infty} = \frac{T_w - T_{c,in}}{(T_\infty - T_{c,in})^2}, \quad \frac{\partial \phi}{\partial T_{c,in}} = \frac{T_\infty - T_w}{(T_\infty - T_{c,in})^2}, \quad \frac{\partial \phi}{\partial T_w} = \frac{-1}{T_\infty - T_{c,in}}
\]

The values of the partial derivatives, and the bias, precision and total uncertainties in the measured \( T_\infty \), \( T_{c,in} \), and \( T_w \) for two values of \( \phi \) are shown in Table D.1. The locations were chosen based on the locations of the warmest and coldest thermocouples for the condition of \( M_{avg} = 0.6 \), which was the condition which had the smallest mainstream to coolant temperature difference. The bias uncertainties were estimated as 0.2 K for direct thermocouple measurements for \( T_\infty \) and \( T_{c,in} \). The bias uncertainty of \( T_w \) shown in Table D.1 was found from the root mean square of the bias uncertainty of the IR camera value and the bias uncertainty of the calibration thermocouple. The bias uncertainty of the IR camera was the difference between the temperature of the thermocouple and the IR measurement at the same location (0.08 K for \( \phi = 0.26 \) and 0.8 K for \( \phi = 0.06 \)). The bias of the calibration thermocouple measurement was 0.2 K.

The precision uncertainties were calculated from Equation (D.2) based on the number of individual measurements taken, \( N_x \), the standard deviation of the measurements, \( \sigma_x \), and the relevant \( z \) or \( t \) value corresponding to a 95% confidence level. For the thermocouple measurements, sample sizes were greater than 30, so the \( z \) value of 1.96 was used. For the IR temperature measurements, the sample size was 5, which corresponded to a \( t \) value of 2.77. Like the bias uncertainty for \( T_w \), the precision uncertainty for \( T_w \) is found by taking the root mean
square of the individual precision uncertainties from the IR camera measurements and the thermocouple measurements, resulting in the precision value shown in Table D.1.

\[
\text{Precision} \delta x = \left[ \frac{z = 1.96}{t_{N_x = 5} = 2.77} \right] \sigma_x \div \sqrt{N_x} \quad \text{(D.2)}
\]

<table>
<thead>
<tr>
<th></th>
<th>Low ( \phi = 0.06 )</th>
<th>High ( \phi = 0.26 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( T )</td>
<td>Avg. ( T ) (K)</td>
<td>( \frac{\partial \phi}{\partial T} ) (1/K)</td>
</tr>
<tr>
<td>( T_\infty )</td>
<td>326.2</td>
<td>0.025</td>
</tr>
<tr>
<td>( T_{c, in} )</td>
<td>289.0</td>
<td>0.002</td>
</tr>
<tr>
<td>( T_w )</td>
<td>323.9</td>
<td>-0.027</td>
</tr>
</tbody>
</table>

The total uncertainty in \( \phi \) is found by plugging the values in Table D.1 into Equation (D.1). The uncertainty was \( \delta \phi = 0.023 \) for \( \phi = 0.06 \), and \( \delta \phi = 0.008 \) for \( \phi = 0.26 \) at a 95% confidence level for local effectiveness measured using the IR camera. For cases when the effectiveness was measured based on only thermocouple measurements (\( \phi \) \(_{TBC} \) and \( \phi_i \)), the uncertainty was \( \delta \phi = 0.008 \) for \( \phi = 0.06 \), and \( \delta \phi = 0.007 \) for \( \phi = 0.26 \). In some parts of this dissertation, the average uncertainty over multiple surface locations (including area averaged results) was reported. The propagation of uncertainty analysis was applied using the maximum \( \delta \phi \) reported above and the partial derivatives of the average \( \phi \) with respect to \( \phi \) measured at the individual locations. The resulting equation simplifies to \( \delta \bar{\phi} = \delta \phi / \sqrt{N_\phi} \), where \( N_\phi \) is the number of \( \phi \) measurements used in the average. The uncertainty in average \( \phi \) is less than or equal to 0.01 for averages across four or more locations.

### D.2 Uncertainty in Coolant Flowrate

The uncertainty in coolant flowrate was dependent on the uncertainties in the measurements of the pressure drop across the LFE, the temperature of the flow upstream of the LFE, and the atmospheric pressure. The propagation of uncertainty equation for the coolant flowrate is given in Equation (D.3). The partial derivatives could not be determined analytically.
due to the complexity of the dependence on the measured parameters. Therefore, the partial derivatives were estimated using sequential perturbation (Moffat, 1988). The inputs to the measured quantities were slightly perturbed and the resulting change in \( \dot{m}_c \) was divided by the perturbation amount. The uncertainty in \( \Delta P_{LFE} \) was dominated by the bias uncertainty (0.005 in H\(_2\)O), which was 1\% of full scale of the pressure transducer (Setra 264 0.5 in H\(_2\)O Differential Pressure Transducer). The uncertainty in \( T_{LFE} \) was also dominated by the bias uncertainty, which was conservatively estimated as 1 K. The precision uncertainties for \( \Delta P_{LFE} \) and \( T_{LFE} \), calculated from the standard deviations across 200 measurements, were orders of magnitude less than the bias uncertainties. The uncertainty in \( P_{atm} \) measured by a Setra 370 Barometer was estimated as 2 Pa. The sequential perturbation analysis was applied to the flowrate for \( M = 0.6 \) and 2.0, with a result of 3\% uncertainty in \( \dot{m}_c \) for both cases.

\[
\delta \dot{m}_c = \sqrt{\left( \frac{\partial \dot{m}_c}{\partial (\Delta P_{LFE})} \delta (\Delta P_{LFE}) \right)^2 + \left( \frac{\partial \dot{m}_c}{\partial T_{LFE}} \delta T_{LFE} \right)^2 + \left( \frac{\partial \dot{m}_c}{\partial P_{atm}} \delta P_{atm} \right)^2} \quad (D.3)
\]

### D.3 Uncertainty in Velocity Measurements with PIV

The uncertainty in the PIV velocity measurement was estimated using the error dependence on the maximum instantaneous particle displacement gradients (Raffel et al., 2007), which was the largest contribution to error in the velocity calculation. The displacement gradients were calculated by multiplying the time delay between laser pulses (\( dt \)) by the maximum instantaneous velocity gradients in the transverse direction. The velocity gradients for Planes A and B were \( \partial U_z / \partial z \) and \( \partial U_z / \partial s \) for the horizontal and vertical velocities respectively. For Plane C, the velocity gradients were \( \partial U_y / \partial z \) and \( \partial U_z / \partial y \). The maximum instantaneous displacement gradients, and the corresponding uncertainties in pixels (RMS error), in normalized instantaneous velocity and in normalized time averaged velocity are given in Table D.3. The RMS error was determined for a final interrogation window of 24 x 24 pixels (Raffel et al., 2007). The uncertainty in time averaged velocities was dependent on the number of image pairs recorded for each case, \( N_U \), according the formula for the uncertainty in the mean,

\[
\delta \overline{U} = \delta U / \sqrt{N_U} .
\]

There were 3000 image pairs recorded for Planes A and B and 6000 image pairs recorded for Plane C.
<table>
<thead>
<tr>
<th>Planes A and B</th>
<th>Max. Displ. Gradient [pixels/pixel]</th>
<th>RMS error [pixels]</th>
<th>Instantaneous Velocity Uncertainty (normalized by $U_{x,in}$)</th>
<th>Time Averaged Velocity Uncertainty (normalized by $U_{x,in}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Horizontal Velocity, $U_x$</td>
<td>0.2</td>
<td>0.6</td>
<td>0.12</td>
<td>0.002</td>
</tr>
<tr>
<td>Vertical Velocity $U_z$</td>
<td>0.2</td>
<td>0.6</td>
<td>0.12</td>
<td>0.002</td>
</tr>
<tr>
<td>Plane C</td>
<td>Max. Displ. Gradient [pixels/pixel]</td>
<td>RMS error [pixels]</td>
<td>Instantaneous Velocity Uncertainty (normalized by $U_{x,in}$)</td>
<td>Time Averaged Velocity Uncertainty (normalized by $U_{x,in}$)</td>
</tr>
<tr>
<td>Horizontal Velocity, $U_y$</td>
<td>0.2</td>
<td>0.6</td>
<td>0.15</td>
<td>0.002</td>
</tr>
<tr>
<td>Vertical Velocity $U_z$</td>
<td>0.28</td>
<td>0.9</td>
<td>0.23</td>
<td>0.003</td>
</tr>
</tbody>
</table>
VITA

Amy Mensch

Amy Mensch completed her Bachelor of Science in Mechanical Engineering from the University of Maryland Baltimore County (UMBC) in May 2007. After graduation, Amy attended graduate school at the Pennsylvania State University (Penn State) where she studied soot formation in surrogate jet fuels under the guidance of Dr. Robert Santoro. Upon completing her Master of Science degree in Mechanical Engineering in 2009, Amy began a research position in the Fire Research Division of the National Institute of Standards and Technology (NIST). While at NIST Amy performed research to evaluate the thermal performance of fire fighting respirator masks. In 2011, Amy returned to Penn State to pursue her doctoral degree in the Experimental and Computational Convection Laboratory (ExCCL) with Dr. Karen Thole. Amy’s research has focused on using a method to simulate the conjugate heat transfer and cooling relevant to gas turbine airfoils, with funding support from the University Turbine Systems Research (UTSR) program of the National Energy Technology Laboratory (NETL) of the Department of Energy. In 2014, Penn State awarded Amy the College of Engineering Distinguished Teaching Fellowship, which involved teaching a required junior level applied math course. Amy will complete her Doctor of Philosophy in Mechanical Engineering in May 2015, and then will be working on a Post-Doc at the National Institute of Standards and Technology.