THERMALLY INDUCED FLOWS AND VENTILATION WITHIN AN AIRCRAFT WING LEADING EDGE COMPARTMENT

Daithí Moore, BEng(Hons) Aero. Eng.

A thesis submitted for the degree of Doctor of Philosophy at the Faculty of Science & Engineering, University of Limerick, Ireland

Supervisors

Dr. Vanessa Egan & Dr. David Newport
Stokes Institute, Department of Mechanical, Aeronautical & Biomedical Engineering
University of Limerick

Dr. Vesna Lacarac
AIRBUS, Filton BS997AR, United Kingdom

Submitted to the University of Limerick, 2012
Declaration

The substance of this thesis is the original work of the author, and due reference and acknowledgement has been made, where necessary, to the work of others. No part of this thesis has been submitted in canditure for any degree.

__________________________________________
Daithí Moore (Candidate)

__________________________________________
Dr. Vanessa Egan (Supervisor)

__________________________________________
Dr. David Newport (Supervisor)

This thesis was defended on the 2nd of November, 2012.

Examination Committee

<table>
<thead>
<tr>
<th>Role</th>
<th>Name</th>
<th>Institution</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chairman</td>
<td>Dr. Jeremy Robinson</td>
<td>University of Limerick</td>
</tr>
<tr>
<td>External Examiner</td>
<td>Prof. Oronzio Manca</td>
<td>Second University of Naples</td>
</tr>
<tr>
<td>Internal Examiner</td>
<td>Dr. Reena Cole</td>
<td>University of Limerick</td>
</tr>
</tbody>
</table>
Abstract

Modern day airliners are subjected to a wide range of environmental conditions when operational across the globe. In order to ensure reliable aircraft performance and avoid thermal failure of heat sensitive components, consideration must be given to the wide range of temperatures that the aircraft will encounter when designing aircraft internal enclosures. This thesis presents an investigation into the fluid flow and heat transfer within an aircraft wing leading edge compartment due to solar loading and the presence of an internal bleed duct. A detailed experimental investigation into the thermal distribution, bleed duct heat transfer and flow structure was performed for $1 \times 10^5 < \text{Grashof No.} < 4 \times 10^5$ in a representative wing leading edge enclosure constructed for this purpose. Temperature measurements were recorded along the interior surfaces along with the internal air temperature in the cavity. Particle Image Velocimetry (PIV) was used to quantify the flowfield present in the leading edge. There were three main aspects to this thesis:

- The first was to characterise the enclosure conditions and bleed duct heat transfer within the sealed leading edge. This was performed firstly to gain an understanding of the heat transfer characteristics within the leading edge and secondly to use as a comparison for when ventilation was introduced. It was observed that the confining effect of the enclosure was only evident at the higher Gr range investigated, with the initial heat transfer similar to that of an unconfined cylinder. The interior temperature distribution was also dependent upon Gr, with the plume mixing effect creating a more homogeneous air temperature as Gr increased.

- The effect of enclosure ventilation was also considered with a view to helping reduce a thermally aggressive environment which can be present in a hot ambient environment. The placement of the ventilation openings was limited due to the geometric constraints of the leading edge, yet a 77% increase in the bleed duct heat transfer was observed for a dual vent opening configuration.

- Geometrical effects within the leading edge were investigated in order to appreciate their influence on heat transfer and ventilation. Partitioning of the leading edge (due to a front sub-spar) produced a blockage between the inlet and outlet vent which reduced the bleed duct heat transfer. Interior temperature and flowfield were also observed to be dependent upon the bleed duct position, as its buoyant plume was the main driver of the flow structure within. The effect of a larger, constant temperature cylinder was also considered with its position relative to the bleed duct important for heat transfer as it was effectively eliminated when located below the bleed duct with a higher surface temperature.

The findings of this thesis allow for a detailed understanding of the thermal environment and flow structure in a non-standard enclosure shape subjected to exterior heating and internal heat source. This is of practical relevance for the design of aircraft enclosures to help eliminate the possibility of thermal failure of components and affect aircraft operations.
Acknowledgements

I firstly want to offer a special thanks and gratitude to my supervisors, Dr. David Newport and Dr. Vanessa Egan for their support, guidance and patience through the course of this research. Their experience has been invaluable to me.

I would like to acknowledge the support of the Irish Research Council for Science and Engineering Technology through the Embark Initiative for funding this research. Thanks also to Airbus for facilitating this research and particularly to Dr. Vesna Lacarac.

To the staff and postgraduate students at the Stokes Institute I would like to offer my warmest and sincerest thanks for the brilliant time I have spent there. To my fellow PhD students who travelled down this road with me I want to say thank you for all the good times together, this experience wouldn’t have been the same without you. To Tom, for support and companionship whilst testing in The Cave, to bounce ideas off and good craic on trips abroad, this truly has been a shared experience. To those I’ve had the pleasure to share an office with: Jason, Niall, Ollie, Paul, Fiachra, Stefano, Angie, Dave, Mark, Alessio, Cillian & Colin. To TJ, Brendan and to Emma, who carried the heavy weight of being the only girl in Stokes with pride and managed to survive all of us! To Paddy O’Regan and David Jones for their help and advice in all things experimental. Special thanks also goes to Fionnuala O’Connell who is the glue that keeps Stokes Institute held together.

To all those I met in Limerick, thorough UL or otherwise, I thank you. To those who I’ve lived with and have had to put up with me round the clock, Paddy, Julia, Soraya, Claire, Áine and Maria. To Maurice, from early mornings in the gym to mountain biking, kiting, going for a kick about or just killing time with, thank you for keeping me suitably distracted when I needed to be!

I would like to thank my family and friends for all their support, advice and encouragement over the years. You have made this journey possible and most importantly enjoyable.

Finally to Laura, a confidant to share the tough times with and a true friend to enjoy the good times with, I thank you with all my heart.

Daithí
This thesis is dedicated to my parents Paul and Anne, whose love and support has been invaluable in both my academic and personal pursuits.

Thank you.
## Contents

Abstract ................................................................. ii  

Acknowledgements .................................................. iii  

List of Tables ......................................................... viii  

List of Figures ......................................................... xvi  

Nomenclature ........................................................ xx  

1 Introduction ............................................................ 1  

1.1 Background ......................................................... 2  

1.2 Research Objectives ............................................... 5  

1.3 Thesis Structure .................................................. 7  

2 Literature Review ............................................................. 9  

2.1 Enclosure Natural Convection ................................... 10  

2.1.1 Geometrical Effects ........................................... 14  

2.1.2 Localised Heat Source ........................................ 21  

2.1.3 Enclosure Ventilation ......................................... 24  

2.2 Summary ............................................................. 35  

3 Experimental Methodology ..................................................... 37  

3.1 Leading Edge Test Section ..................................... 37  

3.1.1 Outer Enclosure .............................................. 40  

3.1.2 Internal Configuration ....................................... 41  

3.1.3 Ventilation .................................................... 44  

3.1.4 Instrumentation .............................................. 47  

3.2 Steady State Temperature Measurements ....................... 49  

3.2.1 Introduction .................................................. 49  

3.2.2 Constant Temperature ....................................... 50  

3.2.3 Constant Heat Flux .......................................... 53
List of Tables

3.1 Outline of the constant wall heat flux bleed duct configurations. Locations are illustrated in Fig. 3.4 and Fig. 3.6. ............................................. 45
3.2 Outline of the constant wall temperature bleed duct configurations. Locations are illustrated in Fig. 3.4 and Fig. 3.6. ............................................. 46
3.3 Measurement uncertainties and calculated uncertainties ......................... 64
5.1 Outline of geometrical configurations for Case A - E ............................ 118
5.2 Mass flow of air removed and heat exhausted from the leading edge for Cases A - E .......................................................... 142
C.1 Data points for the leading edge geometry supplied by Airbus. Dimensions in mm.............................................................. C1
List of Figures

1.1 Predicted passenger growth to 2030 released by Airbus in their Global Market Forecast 2012 – 2030 [1]. Similar rises are also predicted by Boeing in the same time frame [2]. ................................................................. 2

1.2 (a) Heat sources affecting the wing leading edge during turnaround. (b) Resultant convective flow structure within the leading edge for the given boundary conditions. The flow is primarily a result of the plume from the cylinder and the boundary layer flow along the interior wall surfaces. . . . . . 4

2.1 Typical flow structures seen in a differentially heated square cavity from the side (a), from below (b) and above (c). ................................. 11

2.2 Increase in heat transfer ($Nu$) for a changing flow regime from an increase in $Gr$ [3] ................................................................. 12

2.3 Contours of isotherms (a) and streamfunction (b) as a result of the change of enclosure inclination. From Bairi [4] ................................. 18

2.4 Flow patterns observed in a square enclosure as a result of varying the internal cylinder positioning. From Ekundayo et al. [5] ..................... 23

2.5 Mixing ventilation examples for a heated object within a square enclosure. Depending upon the strategy required, ventilation can be either be in the form of single inlet/outlet opening (a), single inlet, single outlet (b), multiple inlet/outlet openings (c). ................................................................. 25

2.6 Displacement ventilation examples for a heated object within a square enclosure. Presented here are multiple inlet/outlet openings (a) and single inlet/outlet opening (b). ................................................................. 26

2.7 Change in observed flow path resulting from the varying vent openings investigated by Yu and Yoshi [6] ............................................. 27

2.8 Reduction in enclosure $\Delta T$ between the average interior and exterior temperatures alternating between a pure mixing ventilation configuration ($R^* = 0$) and displacement ventilation ($R^* = 1$). From Haslavsky et al. [7] ............... 28
2.9 Interaction between the inlet and ventilation flow for varying outlet positions (a) and ventilation efficiency for increasing Re for the four cases (b) (Xaman et al. [8]). ................................................... 32

2.10 The change in heat transfer and contaminant concentration with Gr and Re investigated by Deng et al. [9]. ................................................... 33

3.1 Location of the heater mats upon the upper surface (a) and the lower surface (b). The actual leading edge section with the hollow Ø50mm cylinder mounted horizontally is shown in (c). Channel for laser access into leading edge is highlighted at point (1). ................................................... 38

3.2 Image of the outer enclosure constructed to allow for adequate seeding in the ventilated case studies. Also seen here are the Litron Nd-YAG laser head at the left of the picture and the CCD camera in the foreground. ...... 40

3.3 Location of measurement thermocouples on constant temperature bleed duct (a) and constant heat flux bleed duct (b). ................................. 41

3.4 Cylinder locations for the single cylinder test cases: bottom left (A), top left (B) and bottom right (C) ................................. 43

3.5 Mounting method for hollow Aluminium cylinder (a) and solid Aluminium cylinder (b) ................................................... 43

3.6 Cylinder locations for the two cylinder investigation cases. The Ø75mm cylinder’s position remains constant whilst the Ø50mm cylinder is placed above and below it adjacent to the vertical wall. The Ø50mm cylinder positions are shown in Fig. 3.4. ................................................... 44

3.7 (a) Vented leading edge enclosure with openings on the upper and lower surfaces, (b) Vented leading edge enclosure with both openings on the lower surfaces, (c) Top vent opening with half of the mesh grid removed to illustrate difference in open area between a fully open vent and one with mesh attached. (1) Bleed Duct. (2) Glass Pane. (3) Upper Vent. (4) Lower Rear Vent. (5) Lower Front Vent. ................................. 45

3.8 Variation in $R^*$ used in testing as a result of changing the open area of the front and rear vents for Case 4 ................................. 46
3.9 Schematic of the heating and measuring equipment used throughout testing of the leading edge. (1) Personal computer for recording data, (2) Stanford Research Systems SR630 dataloggers, (3) HP Agilent 34970a modular datalogger, (4) Outer enclosure, (5) Leading edge test section, (6) Eurotherm 2216e PID controllers, (7) 240V adjustable autotransformer, (8) Digital Multimeter ................................................................. 48

3.10 Thermocouple locations along the three horizontal and vertical planes ($X^*, Y^* = 0.25, 0.5, 0.75$) and on the vertical wall surface. Measurements recorded at mid-plane ($D = 150mm$) of leading edge enclosure. ............................... 48

3.11 Notation used for calculation of airflow velocity passing normal to the exit vent boundary. ................................................................. 53

3.12 Heat flux acting upon the horizontal cylinder when supplied with a constant heat input from the power supply ................................................. 54

3.13 Resultant resistance network for two surfaces in radiative balance which exchange heat only between each other. ................................. 55

3.14 Schematic of method used to calculate the radiative exchange between individual sections of the cylinder and the vertical wall. This is then totaled over the entire cylinder surface to obtain the total radiative heat transfer. ................................. 56

3.15 PIV equipment setup. (1) Data acquisition and processing personal computer, (2) TSI synchroniser (610035), (3) Litron Nano LPIV Nd-YAG laser, (4) & (5) LPU1000 Laser power supplies, (6) Leading edge enclosure, (7) TSI 2MP Plus CCD camera ................................................................. 58

3.16 Change in the mean velocity recorded for the number of image pairs acquired. ................................. 60

3.17 Overview of the PIV processing workflow. (a) Raw PIV image taken above the partition. Note the shadow present due to the chamfer in the partition. (b) Image after substitution of the average background light intensity. (c) Calculated vector map. (d) Vector map after smoothing and conditioning to fill holes. Here the gap in data due to the presence of the partition shadow has been successfully resolved. ................................. 61
4.1 Temperature rise \((T - T_C)\) for the cylinder (○), interior air (○) and rear wall
(□) for increasing \(q''_{BD}\) ................................. 67

4.2 Case 1 horizontal and vertical temperature profiles \((X^*, Y^* = 0.25(□), 0.5
(○) & 0.75 (△))\) within the enclosure for \(Gr_{BD} = 1.4 \times 10^5\) (a,b), \(2.5 \times 10^5\n(c,d), 3.3 \times 10^5\) (e,f) & \(3.9 \times 10^5\) (g,h). The rear wall temperatures (○) are
superimposed onto the vertical temperature profiles for comparison. .... 68

4.3 Case 1 rear wall temperature profiles for \(Gr_{BD} = 1.4 \times 10^5(○), 2.5 \times 10^5\ (□),
3.3 \times 10^5\ (○) & 3.9 \times 10^5\) (△) .... 69

4.4 Overview of flow structure with the leading edge as a result of the imposed
boundary conditions. .................................................. 70

4.5 Measured flow velocity fields near the Bleed Duct at \(Gr_{BD} = (a) 1.4 \times 10^5,
(b) 2.5 \times 10^5\) .................................................. 71

4.6 Measured flow velocity fields near the Bleed Duct at \(Gr_{BD} = (a) 3.3 \times 10^5
& (b) 3.9 \times 10^5\) .................................................. 72

4.7 Velocity profiles taken between the vertical wall \((x = 0mm)\) and Bleed Duct
surface \((x = 25mm)\) for \(Gr_{BD} = 1.4 \times 10^5\) (□), \(2.5 \times 10^5\) (○), \(3.3 \times 10^5\) (△),
& \(3.9 \times 10^5\) (○) .................................................. 75

4.8 Variation in temperature difference between the bleed duct and the various
reference temperatures chosen. \(T_{ref} = \) average enclosure temperature
(○), enclosure centrelines (□), temperatures local to bleed duct (△), \((T_h + T_c)/2\) (○) and adiabatic bleed duct temperature (×) .......................... 76

4.9 Variation in bleed duct convective Nusselt number. \(T_{ref} = \) average enclosure
temperature (○), enclosure centrelines (□), temperatures local to bleed
duct (△), \((T_h + T_c)/2\) (○) and adiabatic bleed duct temperature (×). .... 77

4.10 Heat flux ratio for increasing \(Gr_{BD}\): (○) convection , (□) radiation. .... 79

4.11 Isolated convective Nusselt number (○). Free cylinder correlation (dashed
line) from Churchill & Chu [3], enclosed cylinder correlation (solid line)
from Sparrow et al. [10]. Remaining correlations are from comparative
study by Fand & Brucker [11]: (□) Morgan [12], (○) Rice [13], (△) Kyte
[14], (×) van der Hegge Zijnen [15], (○) Fand & Brucker [11]. .... 80
4.12 Temperature difference between the bleed duct and $C_2$. Case 2 ($\diamond$), Case 3 ($\Box$) ................................. 82

4.13 Case 2 horizontal and vertical temperature profiles ($X^*$, $Y^*$= 0.25($\Box$), 0.5 ($\diamond$) & 0.75 ($\triangle$)) within the enclosure for $Gr_{BD} = 1.5 \times 10^5$ (a,b), $2.5 \times 10^5$ (c,d), $3.2 \times 10^5$ (e,f) & $3.7 \times 10^5$ (g,h). The rear wall temperatures ($\diamond$) are superimposed onto the vertical temperature profiles for comparison. ...................... 83

4.14 Case 3 horizontal and vertical temperature profiles ($X^*$, $Y^*$= 0.25($\Box$), 0.5 ($\diamond$) & 0.75 ($\triangle$)) within the enclosure for $Gr_{BD} = 1.5 \times 10^5$ (a,b), $2.5 \times 10^5$ (c,d), $3.2 \times 10^5$ (e,f) & $3.7 \times 10^5$ (g,h). The rear wall temperatures ($\diamond$) are superimposed onto the vertical temperature profiles for comparison. ...................... 85

4.15 Case 2 (a) & Case 3 (b) rear wall temperature profiles for $Gr = 1.5 \times 10^5$($\diamond$), $2.5 \times 10^5$ ($\diamond$), $3.2 \times 10^5$ ($\triangle$) & $3.7 \times 10^5$ (□) .................................................. 86

4.16 Average interior air $T^*$ for Case 1 ($\diamond$), Case 2 ($\diamond$) and Case 3 (□) ................................. 86

4.17 Average rear wall $T^*$ for Case 1 ($\diamond$), Case 2 ($\diamond$) and Case 3 (□) ................................. 87

4.18 Temperature rise between the bleed duct and interior air ($T_{BD} - T_{avg}$) for Case 1 ($\diamond$), Case 2 ($\diamond$) and Case 3 (□) .................................................. 88

4.19 Isolated convective Nusselt number for Case 1 ($\diamond$), Case 2 ($\diamond$) and Case 3 (□) ................................. 90

4.20 Change in bleed duct temperature ratio for Case 2 ($\diamond$) and Case 3 (□) ................................. 91

4.21 Comparison of normalised convective Nusselt number vrs. cylinder temperature ratio for Case 2 ($\diamond$) and Case 3 (□). Also present is the data from Sparrow and Niethammer [16] for unconfined cylinders at $s/D = 2$ ($\times$). .................. 92

4.22 Normalised convective Nusselt number ($Nu/Nu_0$) for Case 3 (□) compared to unconfined cylinder (solid line) from Sparrow and Niethammer [16]. .................. 93

4.23 Effective cylinder temperature ratio (□) for Case 3 compared to the measured value (solid line). ................................................................. 94

5.1 Temperature distributions within the leading edge along the (a) horizontal centreline, (b) vertical centreline & (c) rear wall comparing the ventilation configurations investigated: -$\diamond$- Unventilated enclosure, -$\diamond$- Single bottom vent, -$\Box$- Vents on upper and lower surfaces, -$\triangle$- Two vents on lower enclosure surface. These openings are illustrated in (d). ................................................................. 99
5.2 Difference in temperature rise \((T - T_c)\) for the bleed duct (○), average enclosure interior air (●) and rear wall (□) based upon bleed duct input power (a) and Grashof number (b). Closed and open symbols represent the sealed and ventilated cases respectively. .......................................................... 102

5.3 Interior air \(T^* \left[ \frac{(T - T_c)}{(T_h - T_c)} \right]\) distribution for the sealed (○) and ventilated (●) leading edge based on bleed duct input power (a) and Grashof number (b). .......................................................... 103

5.4 Rear Wall \(T^*\) distribution for the sealed (○) and ventilated (●) leading edge based on bleed duct input power (a) and Grashof number (b). .......................................................... 104

5.5 Interior temperature profiles taken along the horizontal planes \(Y^* = 0.25(□), 0.5 (○ & 0.75 (△)) \) (a, c, e, & g) and along the vertical planes \(X^* = 0.25(□), 0.5 (○ & 0.75 (△)) \) (b, d, f, h). \(Gr = 1.4 \times 10^5 \) (a,b), \(2.2 \times 10^5 \) (c,d), \(2.8 \times 10^5 \) (e,f) & \(3.3 \times 10^5 \) (g,h). The rear wall temperatures (○) are superimposed onto the vertical temperature profiles for comparison. .......................................................... 106

5.6 Ventilated leading edge rear wall temperature profiles for \(Gr_{BD} = 1.4 \times 10^5 \) (○), \(2.5 \times 10^5 \) (●), \(3.3 \times 10^5 \) (△) & \(3.9 \times 10^5 \) (□) .......................................................... 107

5.7 Temperature rise between cylinder and interior air \((T_{BD} - T_\infty)\) for sealed (○) and ventilated (●) leading edge. .......................................................... 108

5.8 Nusselt number for sealed (○) and ventilated (●) leading edge ................. 109

5.9 Difference in temperature rise \((T - T_c)\) for the bleed duct (○), average enclosure interior air (●) and rear wall (□) based upon bleed duct input power (a) and Grashof number (b). Closed and open symbols represent the sealed and vented & partitioned cases respectively. .......................................................... 110

5.10 Interior air \(T^*\) profile for the sealed (○), vented (●) and vented & partitioned (□) leading edge based on bleed duct input power (a) and Grashof number (b).......................................................... 111

5.11 Rear Wall \(T^*\) profile for the sealed (○), vented (●) and vented & partitioned (□) leading edge based on bleed duct input power (a) and Grashof number (b).......................................................... 112
5.12 Interior temperature profiles taken along the horizontal planes \((Y^* = 0.25(\square), 0.5 (\diamond) \& 0.75 (\triangle))\) (a, c, e, & g) and along the vertical planes \((X^* = 0.25(\square), 0.5 (\diamond) \& 0.75 (\triangle))\) (b, d, f, h). \(Gr = 1.8 \times 10^5\) (a,b), \(2.7 \times 10^5\) (c,d), \(3.3 \times 10^5\) (e,f) & \(3.7 \times 10^5\) (g,h). The rear wall temperatures (\(\circ\)) are superimposed onto the vertical temperature profiles for comparison.

5.13 Vented & partitioned leading edge rear wall temperature profiles for \(Gr = 1.8 \times 10^5\) (\(\circ\)), \(2.7 \times 10^5\) (\(\diamond\)), \(3.3 \times 10^5\) (\(\triangle\)) & \(3.7 \times 10^5\) (\(\square\))

5.14 Temperature rise between cylinder and interior air \((T_{cyl} - T_{avg})\) for sealed (\(\circ\)) and vented & partitioned (\(\diamond\)) leading edge.

5.15 Nusselt number for sealed (\(\circ\)), ventilated (\(\diamond\)) and vented & partitioned (\(\square\)) leading edge.

5.16 Horizontal centreline (a), vertical centreline (b) and rear wall (c) temperature profiles for the empty (\(\diamond\)), bleed duct only (\(\circ\)) and bleed duct and partitioned leading edge (\(\square\)). These correspond to Case A - C respectively.

5.17 Horizontal centreline (a), vertical centreline (b) and rear wall (c) temperature profiles for the various bleed duct locations: above rear vent opening (\(\diamond\)), adjacent to partition over lower surface (\(\circ\)) adjacent to rear wall in upper corner (\(\square\)). These correspond to Case C - E respectively. Bleed duct locations are illustrated in (d) (i) - (iii).

5.18 PIV flow structure over lower surface for unpopulated vented leading edge (Case A).

5.19 PIV flow structure for vented leading edge with bleed duct (Case B).

5.20 PIV flow structure for vented leading edge with bleed duct & partition (Case C).

5.21 PIV flow structure for vented & partitioned leading edge with bleed duct adjacent to partition (Case D).

5.22 PIV flow structure for vented & partitioned leading edge with bleed duct in upper rear corner (Case E).

5.23 Airflow crossing exit vent boundary for Case A (a), Case B (b), Case C (c), Case D (d) Case E (e).
Variation in interior air $T^*$ with $R^*$ for the Case A ($\phi$), Case B ($\circ$) and Case C ($\square$). ... 135

Variation in $T_A$ with $R^*$ for Case A ($\phi$), Case B ($\circ$) and Case C ($\square$). ... 137

Variation in $T_R$ with $R^*$ for Case A ($\phi$), Case B ($\circ$) and Case C ($\square$). ... 138

Variation in ventilation efficiency ($\varepsilon$) with $R^*$ for Case A ($\phi$), Case B ($\circ$) and Case C ($\square$). ... 139

Velocity profile of air crossing exit vent boundary for Case A ($\square$), Case B ($\phi$), Case C ($\circ$), Case D ($\triangle$) & Case E ($\times$). ... 141
## Nomenclature

### Roman

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Area</td>
<td>m²</td>
</tr>
<tr>
<td>c</td>
<td>Circumference</td>
<td>m</td>
</tr>
<tr>
<td>(C_p)</td>
<td>Pressure coefficient</td>
<td>-</td>
</tr>
<tr>
<td>D</td>
<td>Depth</td>
<td>m</td>
</tr>
<tr>
<td>(E_b)</td>
<td>Blackbody emissive power</td>
<td>(W/m²)</td>
</tr>
<tr>
<td>F</td>
<td>Radiation view factor</td>
<td>-</td>
</tr>
<tr>
<td>Gr</td>
<td>Grashof number</td>
<td>-</td>
</tr>
<tr>
<td>H</td>
<td>Height</td>
<td>m</td>
</tr>
<tr>
<td>h</td>
<td>Heat transfer coefficient</td>
<td>(W/m²K)</td>
</tr>
<tr>
<td>I</td>
<td>Current</td>
<td>A</td>
</tr>
<tr>
<td>J</td>
<td>Radiosity</td>
<td>(W/m²)</td>
</tr>
<tr>
<td>k</td>
<td>Thermal conductivity</td>
<td>(W/mK)</td>
</tr>
<tr>
<td>(\dot{m})</td>
<td>Mass flow rate</td>
<td>kg/m²</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt Number</td>
<td>-</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl number</td>
<td>-</td>
</tr>
<tr>
<td>(q^*)</td>
<td>nondimensional heat flux</td>
<td>-</td>
</tr>
<tr>
<td>(q'')</td>
<td>Heat flux</td>
<td>(W/m²)</td>
</tr>
<tr>
<td>R*</td>
<td>Vent ratio</td>
<td>-</td>
</tr>
<tr>
<td>Ra</td>
<td>Rayleigh number</td>
<td>-</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
<td>-</td>
</tr>
</tbody>
</table>
s  Cylinder separation distance  m
T  Local temperature  K
\( t \)  Time  s
\( T^* \)  nondimensional temperature
U  Velocity relative to u,v coordinates  \( m/s^2 \)
u  Horizontal velocity component relative to exit vent boundary  \( m/s^2 \)
V  Velocity relative to x,y coordinates  \( m/s^2 \)
V  Voltage  V
v  Vertical velocity component relative to exit vent boundary  \( m/s^2 \)
W  Width  m
\( x \)  Horizontal position  m
\( X^* \)  nondimensional horizontal position
\( y \)  Vertical position  m
\( Y^* \)  nondimensional vertical position

**Greek**

\( \beta \)  Thermal expansion coefficient  K\(^{-1}\)
\( \epsilon \)  Emissivity
\( \nu \)  Kinematic viscosity  \( m^2/s \)
\( \Omega \)  Ohmic dissipation
\( \rho \)  density  kg/m\(^3\)
\( \theta \)  Angle of inclination  °
\( \varepsilon \)  Efficiency

**Subscripts**

\( \infty \)  freestream conditions

0  Normalised w.r.t. the single bleed duct configuration

A  Temperature rise (exit/average)

avg  Average inside leading edge

BD  Bleed duct

c  Cold surface

cond  Conduction

conv  Convection

cyl  Cylinder surface

ex  Exhausted

exit  At exit vent

front  Front vent

h  Hot surface

I  Temperature rise (inlet/average)

inlet  At inlet vent

n  Normal to vent boundary

R  Temperature rise (exit/inlet)

rad  Radiation

rear  Rear vent
ref  Reference condition

total  Combined front and rear vent

wall  Enclosure wall

**Acronyms**

2-D  Two Dimensional

3-D  Three Dimensional

AC  Alternating Current

ACH  Air Changes per Hour

AR  Aspect Ratio

CCD  Charge Coupled Device

CWHF  Constant Wall Heat Flux

CWT  Constant Wall Temperature

FFT  Fast Fourier Transform

IO  Input/Output

PC  Personal Computer

PID  Proportional-Integral-Derivative

PIV  Particle Image Velocimetry

TR  Temperature Ratio
Chapter 1

Introduction

“Oh, I have slipped the surly bonds of earth and danced the skies on laughter silvered wings.
Sunward I’ve climbed and joined the tumbling mirth of sun-split clouds
And done a hundred things you have not dreamed of.”

— John Gillespie Magee, High Flight (An Airman’s Ecstasy)

Modern air travel has revolutionised the movement of people and goods across the globe and truly made the world a smaller place. In the first century of aviation, terrific advancements have seen man travel in ways and means unimaginable to generations previous: the first heavier than air flight, the birth of the jet age, supersonic flight and the first tentative steps beyond the atmosphere. It is a testament to the dedication and sacrifice of the pioneers of aviation that flight has become accessible to the many on a grand and affordable scale. All too often has this taken the aeroplane out of the realm of the dreamer and become a slave of schedules, costs and efficiency. These influences are of concern for the aviation engineer where the increased demand placed upon aircraft drives the need for more efficient and reliable designs in order to fill this demand and maintain airworthiness. With the worlds leading aircraft manufacturers predicting at least a doubling of passenger numbers by 2030 (highlighted in Fig. 1.1) [1, 2], this is going to further increase the pressure on aircraft operations, particularly in the high growth areas of Latin America and Asia-Pacific. These locations also bring their own challenges, as beyond the relatively temperate climates of Europe and North America, environmental conditions such
Air travel remains a growth market

Figure 1.1: Predicted passenger growth to 2030 released by Airbus in their Global Market Forecast 2012 – 2030 [1]. Similar rises are also predicted by Boeing in the same time frame [2].

as high ambient temperatures and high humidity can affect aircraft performance and reliability. The additional weight penalty of including some form of active thermal management system (eg. cooling fans) is also to be avoided where possible as this too can reduce aircraft performance and efficiency. In such scenarios, natural ventilation must be considered, but to do so requires extensive knowledge of the convective heat transfer in aircraft internal structures in order to make such a solution viable.

1.1 Background

Natural convection is present in a broad range of engineering applications and is often unavoidable under certain operating conditions. This is particularly of interest with regards to enclosures in industrial applications, especially as these enclosures tend to be quite different to those typically investigated in engineering literature. From an aircraft perspective this can provide quite a challenge, as not only are there a wide range of enclosure types and locations due to the design of an aircraft structure, the flight profile of a modern airliner means that is is subjected to a wide range of thermal environments. To this extent, the
aircraft is put through a rigorous flight test schedule which includes operation in extreme conditions across the globe to attain certification of the aircraft before introduction into service. This exposes the aircraft to the extreme conditions it may encounter during its lifetime and ensures it is able to operate reliably in such environments. In order to effectively design aircraft components, engineers must be able to understand the heat transfer phenomenon occurring within the aircraft during exposure to these environments and how they interact with the systems onboard.

**Leading Edge Heat Transfer**

The focus of this work is on the wing leading section of a large commercial aircraft. Here exposure of the aircraft skin to the environment imposes an exterior boundary condition upon the enclosure, which changes depending upon the ambient conditions. The conditions under investigation here are for high ambient surroundings (such as operation in a dry desert environment), with the aircraft exposed to significant solar loading. This raises the aircraft skin temperature considerably and provides the external heating boundary conditions for the leading edge enclosure. The sensitivity of enclosure natural convection to the thermal boundary conditions has been well documented, with highly complex flows present even for relatively simple geometries [17–19]. As such, any deviation from these geometries is expected to increase the complexity of the heat transfer analysis even further. Often complex geometries are approximated by simpler shapes such as triangular [20–22], trapezoidal [23–25], parabolic [26, 27] etc. to help simplify this analysis, but care must be taken to ensure that such simplifications can accurately represent the enclosure conditions as often-times this cannot be guaranteed unless an investigation into the more complex geometry is performed beforehand.

In a standard aircraft leading edge compartment there are also a number of components housed within. One such component is the engine bleed duct which transports high temperature air which is bled from the high-stage compressor of the turbine engine and used in numerous aircraft functions such as wing de-icing, powering hydraulic systems and air conditioning. The bleed duct also contributes to the heat transfer regime present in the leading edge through its elevated surface temperature. An outline of the heat transfer regime
Figure 1.2: (a) Heat sources affecting the wing leading edge during turnaround. (b) Resultant convective flow structure within the leading edge for the given boundary conditions. The flow is primarily a result of the plume from the cylinder and the boundary layer flow along the interior wall surfaces.

present in the enclosure is presented in Fig. 1.2. The confinement effect upon an isolated heat source has been seen to produce a marked change on the heat transfer and flow patterns compared to an unconfined one due to entrainment effects [5], recirculations [28] and the presence of and distance to confining enclosure walls [29]. Electrical wires also pass through the wing structure, while hydraulic pipes and mechanisms for operating leading edge high-lift devices, de-icing equipment, temperatures sensors and structural components such as sub-spar assemblies can also be present. When these components are considered, the conditions within the leading edge can be quite different to that of a simple empty enclosure when trying to understand its heat transfer characteristics.
CHAPTER 1. INTRODUCTION

Ventilation

As a result of the extreme environments which an aircraft can operate in, the thermal conditions inside the leading edge enclosure can be much more aggressive on internal components than intended. The thermal environment may be such that equipment within may be functioning beyond their specified thermal limits, opening them up to possible unreliable operation or at worst, thermal failure. In these circumstances, it is necessary to introduce some form of cooling into the enclosure to help alleviate this situation. The strict weight limits put on aircraft design means that a passive ventilation solution would be the most applicable as this does not add any significant amount of weight to the structure, it also reduces the complexity of the ventilation regime by not requiring any additional power from the aircraft or control requirements, be they from a systems standpoint or via direct input from the pilot or crew.

Similar to enclosure natural convection, producing an optimised ventilation strategy is dependant upon a number of parameters such as vent size and location [6], the type of ventilation (mixing or displacement) [7,30], the location of the heat source [31] and the flow path through the enclosure [8]. These are often unique for any given configuration. With this in mind, the effect of placing a number of ventilation openings in the leading edge upon the resulting thermal distribution and flowfield is required in order to assess the suitability and effectiveness of such a solution. Due to the highly curved nature of the leading edge, a further enclosure effect is also introduced. Optimisation of any ventilation therefore requires a deeper knowledge of the interior conditions as a result, which is intended to be gained from this thesis for the leading edge enclosure.

1.2 Research Objectives

This thesis presents an investigation of the thermal distribution and flowfield present in an aircraft wing leading edge compartment subjected to exterior solar loading in the presence of a hot bleed duct. The primary objectives were:

- To investigate the heat transfer characteristics of the bleed duct when it is confined
in the non-standard shape of the leading edge enclosure. This was performed to
determine if the leading edge shape had any particular impact upon the bleed duct
heat transfer not observed for other configurations.

- To determine the interaction between the bleed duct plume and the flow on the in-
terior enclosure walls due to its influence upon the internal temperature distribution
and bleed duct heat transfer.

- To detail the interaction between the bleed duct and a hot bundle of wires which
also passed through the leading edge. Bleed duct placement was also considered to
determine its effect upon the bleed duct heat transfer and the internal temperature
distribution in the multiple cylinder configuration.

- To evaluate the impact and effectiveness of enclosure ventilation with a view to re-
ducing the aggressive thermal environment present. Influence on both the bleed duct
heat transfer and internal temperature distribution was also considered.

- To observe the change in flowpath in the leading edge as a result of the changing
geometric conditions investigated. This was necessary to understand the influence
of the enclosure on the flowfield, but also with a view to creating a more optimised
flowpath, allowing for greater ventilation efficiency due to increased interaction with
the core bulk air.

- Vent location and size was considered with a view to determining if an optimum
configuration can be achieved, given the geometrical constraints of the leading edge.
Further geometrical effects such as enclosure partitioning and bleed duct placement
were also considered to investigate their impact upon the interior temperature distri-
bution and ventilation efficiency.

The conclusions obtained from these investigations can be used as guidance for the design
and optimisation of both the placement of heated objects and geometrical constraints when
considering the ventilation of a non-standard enclosure type such as the leading edge.
1.3 Thesis Structure

The following work in this thesis has been split into four main chapters. Chapter 2 presents a review of the published literature on enclosure heat transfer and ventilation. Chapter 3 details the experimental setup, the measurement techniques and analysis methods used. Chapter 4 discusses the heat transfer from the horizontal engine bleed duct in the leading edge, including the effect of a second larger bundle of high temperature wires placed above or below the bleed duct. Chapter 5 describes the impact of enclosure ventilation upon the bleed duct heat transfer, thermal distribution and flowpath in the leading edge. Bleed duct placement and the presence of a partition in the leading edge are also considered. Finally, Chapter 6 presents the conclusions and recommendations arising from the work presented in this thesis.
Chapter 2

Literature Review

Enclosure natural convection has received significant detail in engineering literature, mainly as it occurs across numerous engineering disciplines such as electronics cabinets, solar collectors, nuclear reactors & aircraft enclosures. Natural convection can consist of very complex thermal distributions and flow structures that are highly dependent upon the environment within the enclosure and the boundary conditions placed upon it. In free convection, the fluid beyond the thermal and velocity boundary layers is unaffected by the presence of the heated surface. In enclosure natural convection however, this is not the case as each boundary layer is surrounded by a core region which is in turn affected by the boundary layers on the other walls. As a result, a change in the boundary conditions can alter the conditions experienced in the core region and on other surfaces. This is perhaps one of the main reasons behind the complexity involved in enclosure natural convection, even when relatively simple enclosure and thermal boundary conditions are considered.

The purpose of this review is to determine the underlying factors affecting the natural convection mechanism in an enclosure, how the introduction of ventilation into an enclosure with internal heat sources affects the heat transfer and to examine the possibility of optimising such a system for a given geometry or boundary conditions. The study of natural convection and ventilation is highly specific [32] and as a result, each solution obtained for a given geometric and temperature arrangement is highly unique and is not easily applied to other configurations. The following review is presented to detail the current knowledge in enclosure heat transfer with a view to understanding the conditions present within the
CHAPTER 2. LITERATURE REVIEW

leading edge enclosure under operating conditions.

2.1 Enclosure Natural Convection

Fundamental investigations into enclosure natural convection typically involve a differentially heated square or rectangular cavity. These generally consist of heat flowing through the cavity from the sides or from below. The density difference imposed between the heated (or cooled) air and that of its surroundings sets up the natural convection flow with its associated velocity and thermal boundary layers. An extensive investigation into enclosure natural convection has been performed over the years which has helped to increase the understanding of buoyancy induced flows in enclosures, but at the same time has also exposed their complexity. In a review of natural convection in enclosures, Ostrach [17] outlines the main mechanisms for natural convection in enclosed spaces. He notes the two main methods for initiating buoyancy driven flow. The first occurs when the density gradient is normal to the direction of gravity. In this case flow is initiated instantly and recirculates throughout the cavity. The second is when the density gradient is in the same plane as the influence of gravity, but in an opposing direction. Initially this remains in a state of “unsteady equilibrium” until a critical density is reached whereby the denser (cooler) fluid can no longer remain suspended above the more buoyant warm fluid. A steady cellular recirculation exists in the cavity as a result. When the density gradient acts in the same direction as gravity, the fluid becomes stably stratified. These are the three basic flow patterns presented in Fig. 2.1. Ostrach also notes how it is rarely as simple as one or the other of these types occurring on their own, but an interaction between two (or indeed three) of these modes combined.

When heated from the sides, simultaneous opposing boundary layers form on the hot and cold walls and as a result of the recirculation, distinct thermal layers also form over the adiabatic surfaces. This loop facilitates the flow of heat between the hot and cold wall surfaces which can be calculated by

$$\frac{q}{A} = h(T_h - T_c)$$ \hspace{1cm} (2.1)

The heat transfer, characterised by the Nusselt number
\[ Nu = \frac{hx}{k} \]  

describes the exchange of heat across a boundary (which can be at a local point or taken globally across a surface) for a given length scale \( x \), compared to a pure conduction case with no convection present and can be loosely correlated with the change in flow regime in the enclosure. When the Grashof number

\[ Gr = \frac{g \beta (T_h - T_c) x^3}{\nu^2} \]  

of the system changes (the ratio of buoyancy to viscous forces in the flow) so too does the type of flow structure in the cavity, which will in turn affect the amount of heat transferred by any given surface. This relationship between \( Nu \) and \( Gr \) (from [3]) is presented in Fig. 2.2. Initially, at low \( Gr \) (below \( 10^3 \)), motion of the fluid in the cavity is minimal such that its effect is negligible compared to the conduction across the cavity through the fluid, with a linear increase in temperature across the cavity. As \( Gr \) increases, the development of the thermal and velocity boundary layers on both the hot and cold wall surfaces occurs. Initially, these are not distinct enough to avoid meeting in the centre of the cavity and an asymptotic flow is present (Region II). This recirculation is sufficient however to promote convective heat transfer between the hot and cold surfaces, leading to a rise in \( Nu \). Laminar boundary flow is then established within (Region III), with the flow (and subsequent
transfer of heat) confined to the near wall regions of the enclosure, with relatively stagnant and stratified fluid in the core of the cavity. This leads to a reduction in the increase in $Nu$ with $Gr$. Further increases to $Gr$ see the development of turbulence in the cavity in both the transition period (Region IV) and turbulent boundary layer flow (Region V). Here the increase in $Nu$ with $Gr$ increases once more as heat transfer is aided by enhanced viscous dissipation of heat throughout the turbulent boundary layer.

The condition in Fig. 2.1 (a) holds providing that $\frac{H}{L} < \frac{GrPr^{1/4}}{4}$ is observed. If the aspect ratio ($H/L$) of the enclosure increases beyond this “tall enclosure limit” [33], then the gap between the two vertical walls is not sufficient for two distinct boundary layers to form, which causes conduction to be the main form of heat transfer between the hot and cold sides. Similarly, if the aspect ratio decreases below the “shallow enclosure limit” then the recirculations along the upper and lower non-participating (from a heat transfer perspective) surfaces come into contact and form a barrier to the flow path and transfer of heat between the hot and cold walls.

The review of early literature [17] has also illustrated how important accurate assumptions to the flow conditions are upon the results and conclusions obtained. In the absence of more detailed information, several authors initially made assumptions about the flow which were deemed to be accurate at the time, yet as further investigations were performed these
were found to be not representative of the flow at all. Examples of this are the isothermal rotating core assumed to be present in both rectangular enclosures and horizontal cylinders when exterior thermal boundary conditions were applied. These were shown to be incorrect when thermally stratified stagnant fluid was observed under the same conditions when advances in experimentation allowed. As a result researchers were misled (however unintentionally) of the details of the conditions present and led Ostrach to remark how the belief in this assumption “certainly delayed proper description of the problem” for too many years until the confirming experiments were performed.

The complexity of such natural convection systems has led to a desire for more accurate and robust prediction tools which has in turn led to a growth in the use of numerical modelling for investigations into enclosure natural convection. The extent of research has been such that benchmark solutions have been available for natural convection in a square enclosure for a long time (De Vahl Davis [34]), albeit not an exhaustive one as it was limited for Rayleigh numbers of $10^3 - 10^6$, but provided a solid background into the types of conditions most commonly experienced.

Building on this benchmark to include the Rayleigh number range of $10^3 - 10^8$, Wan et al [35] noted that even with the large advances in computing power and computational algorithms to that point, the quest for “efficient and high accuracy methods for problems of this type” detailed by De Vahl Davis was and still is pertinent for the thermal cavity problem and by extension, a variety of other thermal problems as well. Drawing a parallel with the assumptions made in the early days of enclosure natural convection investigations, the accuracy of these models are often assumed to be correct when they can reproduce (to within an acceptable deviation) the results of previous published studies. Most typically global properties are reported (often $\overline{N_u}$) and comparisons drawn. They note that whilst initially this may seem like a reasonable assumption to the accuracy of the numerical investigation, it can be observed that two predictions of notably different local Nusselt number distributions on a surface have the ability to produce almost identical global values over a surface. The same can be said for velocity and temperature distributions along with the location and magnitude of local maximum and minimum $u$, $v$ & $\Psi$. The accuracy of a solution can be called into question when these values cannot be replicated correctly even
CHAPTER 2. LITERATURE REVIEW

if global measurements agree, given the insensitivity of these to local conditions. This is
a reiteration of an almost identical argument by Ostrach 13 years previously where he too
noted the need for proper and careful comparison with benchmark data for both global and
local measurements in order to produce high quality numerical data.

This highlights that although it is easy to become complacent with the efficiency of
numerical modelling (particularly given the dramatic rise in easily accessible computing
power in the last few decades) to produce a required solution or insight into a given scen-
ario, a lot of care and attention must still be put into modelling to ensure the data obtained
is physically realistic. This would be seen to give credence to the notion of careful com-
plimentary experimental testing, especially when little information about the flow field and
enclosure conditions is known beforehand. This is important for new enclosure shapes not
found in literature or experienced before, such as those found routinely in industrial ap-
lications. Indeed a properly benchmarked and “accurate” numerical model may initially
agree with other investigations (and reality), but once the geometry is altered to reflect the
actual configuration, significant deviations in the predictions may occur, particularly given
how sensitive the flow field is to minor changes in the enclosure conditions. This is com-
pounded further if the model cannot predict these local values accurately in the first place
but based upon global values only. This too must come down to the engineer to decide
whether a detailed solution of the flow field is required or a somewhat crude approximation
is suitable for any given application when investigating enclosure natural convection. In
order to be able to do so however, the effect of various enclosure heating and geometrical
configurations must be understood in order for such approximations to be valid.

2.1.1 Geometrical Effects

The coupling between velocity and thermal fields in natural convection means that the sys-
tem is highly susceptible to any change in the enclosure conditions. When natural convec-
tion is weak (low $Gr$) the possibility of a complex flow regime still exists, even when simple
enclosure geometries are considered. An extensive amount of literature is available for
simple square, cuboid and rectangular enclosures [36–50] as a result of the need to under-
stand the fundamental mechanisms of heat and fluid transport across a wide range of uses.
Occasionally more complex enclosure shapes are considered, but these are often highly application specific and deviate significantly from standard enclosure shapes [51–54].

**Varying Boundary Conditions**

Due to the vast range of operating conditions present in industrial scenarios, a number of different heating configurations are encountered which deviate from those presented in Fig. 2.1. This increases the complexities in the flow regime. This is true also for the wall properties where the presence of a vertical temperature gradient can initiate convective motion in an enclosure heated from above which was found to be stagnant under idealised conditions (Fig. 2.1 [c]). The presence of radiative exchange between participating surfaces and fluids in an enclosure will also have an impact upon natural convection heat transfer.

Using a non-standard heating configuration Wu and Ching [55] showed how, in the presence of a heated top wall in a differentially heated cavity, flow separation occurred along the upper surface when its temperature was lower than the heated vertical wall. This separation caused an interaction between the boundary layer flow on the upper surface and the separated flow, which enhanced local heat transfer. When the top wall temperature was significantly higher than the vertical wall [56], this separation moved to the top of the vertical wall with a region of stratified fluid below the upper surface which inhibited convective heat transfer. Corcione [43] reports that an enhancement in heat transfer from the lower surface of a bottom cooled enclosure is increased by the addition of a cooled vertical wall and decreased by a heated vertical wall. The opposite effect is noted on the upper surface. The effect of a conductive vertical wall has been shown by Nouanegue et al. [57], Said et al. [58] & Yedder and Bilgen [59] to affect the flow structure and resultant heat transfer from the surface. The Rayleigh number and wall conductivity ratio was found to have more of an influence than the enclosure aspect ratio.

Aydin et al. [60] show that in a rectangular enclosure heated from the side and cooled from above, Rayleigh number effects plays a dominant role on heat transfer in the shallow enclosure for $10^3 < Ra < 10^7$. As $Ra$ increases, the centre of the recirculation flow formed moves away from the heated wall surface and with further increase in $Ra$ forms a secondary cell in the opposite corner to the heated and cooled walls. This asymmetry is quite different
to the typical flow pattern for a differentially heated enclosure. For a tall enclosure, Nusselt number was found to exhibit a strong dependence on $Ra$ at $Ra > 10^5$. For square and shallow enclosures, the average Nusselt number increases monotonically with $Ra$. Heat transfer is also found to be enhanced compared to a standard differentially heated enclosure. Discrete heating also affects the flow and heat transfer compared to heating along the length of wall, due to an increase in the complexity of the flow regime. Cheikh et al. [61] Chen and Chen [62] Corvaro et al. [63], Koca et al. [22], Sezai and Mohamad [64], Sharma et al. [65] & Varol et al. [66] outline this interaction and heat transfer across various enclosure shapes and heating configurations.

Akiyama and Chong [36] show how the effect of the presence of grey surfaces and radiation alter the temperature distribution in the cavity. Radiation is found to decrease convective heat transfer, particularly at high surface emissivity. Similar findings have been made by Diaz and Winston [26] and Lei and Patterson [21]. Vivek et al. [67] shows the competition between natural convection and surface radiation in a square and shallow enclosure. The hot wall $Nu$ is found to increase when the aspect ratio of the enclosure decreases due to an increased contribution from radiation. Sieres et al. [68] reports a similar dependency comparing the aperture angles of right angled triangles in the presence of radiation. Bianco et al. [69] also showed how the influence of radiation in convergent channels is more prominent at larger convergence angles.

**Enclosure Aspect Ratio and Orientation**

Overviews into the enclosure effect on natural convection in sealed enclosures have been performed [17] which have been updated and expanded by numerous authors recently including Khalifa [48], Ganguli et al. [18] & Turan et al. [70]. Turan et al. [70] presented the effect of a varying aspect ratio on the temperature distribution in a cavity subjected to both constant wall temperatures (CWT) and constant wall heat fluxes (CWHF). For the CWT case the average wall Nusselt number increases with enclosure aspect ratio ($AR$) until a maximum aspect ratio is reached, beyond which $\overline{Nu}$ decreases. For the CWHF scenario, this is found to not be the case and $\overline{Nu}$ increases monotonically with $AR$. They state that
this occurs as a result of the trade-off between advective thermal transport and thermal dif-
fusive transport in the cavity. The point where advection dominates over diffusion is shown
to be $Ra > Pr^{-3}AR^{-7}$ for the CWT and $Ra > Pr^{-4}AR^{-9}$.

Ganguli et al. [18] showed how multiple secondary cells can be formed in tall cavities
in the convection regime for low $Ra$ and $5 \leq AR \leq 40$. These are stable and remain for
infinite time. As $Ra$ increases and for $5 \leq AR \leq 110$, these cells emerge and disappear and
move due to self generated pressure gradients, with small tertiary cells also forming in the
regions between secondary cells which aid in the cell merging process, creating a complex
flow structure. When multicellular flow patterns form, these fluctuations in the flow increase
the local HTC, leading to an increase in overall HTC. They also noted a change in the
relationship between the gap width and HTC for different temperature gradient across the
cavity. Increasing $\Delta T$ increases the likelihood of the formation of multi cellular flow which
stabilises the decrease in local HTC with gap width.

Bairi [4] presented a comprehensive overview of the effect of enclosure inclination
($0 – 360^\circ$) between $Ra = 10^3 – 10^{10}$. Contours of isotherms and streamfunction are presen-
ted in Fig. 2.3. They observed that wall Nusselt number increases with inclination angle to
a critical value at $45^\circ$ for $Ra < 10^5$ and $60^\circ$ for $Ra > 10^6$. $Nu$ then decreases to a minimum
at $270^\circ$, where the hot wall is above and stratified fluid is present. Bahlaoui et al. [71]
showed that for an inclined slender enclosure at $45^\circ$, a number of multicellular configura-
tions are possible (as well as a unicellular condition) and that the presence of these were
dependent upon the Rayleigh number and the surface emissivity of the internal walls. For
the vertical cavity they also showed how the presence of radiation decreases the convective
heat transfer contribution, but also increases the overall combined heat transfer.

This shows how the simple act of rotating a simple enclosure geometry creates complex
heat transfer and flow patterns. This is relevant if components are used beyond their de-
signed operating conditions and configuration, then the performance of the enclosure and
components within could be severely compromised.
CHAPTER 2. LITERATURE REVIEW

Figure 2.3: Contours of isotherms (a) and streamfunction (b) as a result of the change of enclosure inclination. From Bairi [4]

**Enclosure Shape**

As natural convection arises due to the temperature gradient imposed on a fluid from an adjacent heated (or cooled) wall, the resultant flow is highly dependent upon the enclosure configuration. Shown previously has been the significant impact that changing either the aspect ratio or the angle of inclination has on the resultant flow, even for a simple enclosure geometry. It is thus expected that creating increasingly complex geometries (typical of industrial applications) where natural convection can initiate, has the possibility to create even more complex flows and heat transfer as a result. Typical enclosure shapes investigated are those of triangular [20–22, 66, 68, 72], trapezoidal [23–25] and circular [73, 74] as oftentimes more complex geometries can be approximated by these simplified configurations, when such simplifications are valid.

As examples of the additional complexity encountered, Ridouane and Campo [73] presented the variation in heat transfer between a square, arc and circular enclosure subjected to identical boundary conditions when heated and cooled from the sides. In the conduction regime \(Ra < 10^3\), the circular cavity produces an enhancement in heat transfer of up to 22%, but this diminishes considerably when convection dominates, and is within 2%
CHAPTER 2. LITERATURE REVIEW

of the square cavity at $Ra = 10^6$. The benefit of the circular cavity however, is that it allows for a reduction in cross-sectional area of approximately 20%, and proves to be a better configuration from a thermo-geometric standpoint. Hasan et al. [75] showed how the presence of a wavy structure on the upper surface of an enclosure subjected to a constant heat flux representing a corrugated sheet on top of a building subjected to solar loading affected the heat transfer. The corrugation allowed for an increase in the surface area for heat transfer but also the formation of convective eddies in the sinusoidal region of the upper surface which allowed for enhanced heat transfer. This effect was more prevalent for the increase in corrugation frequency compared to increase in $Ra$, which only increased $Nu$ slightly under the same conditions. Varol and Oztop [76] studied the opposite configuration with a wavy heated surface on the bottom surface and also found an increase in heat transfer for an increase in the corrugation frequency but also for an increasing enclosure aspect ratio. Natarajan et al. [23] showed how for a trapezoidal enclosure with heating from below and cool vertical walls, that uniform heating of the lower surface produced a greater overall Nusselt number from the surface compared to non-uniform heating. Symmetrical flow structures were observed in both heating configurations.

Chen and Cheng [77] presented the natural convection within an arc-shaped enclosure heated on the curved face and showed that natural convection effects only begin to become appreciable when $Gr > 10^5$, with little change in $Nu$ with inclination angle below this due to the poor convective structures present. Above this, $Nu$ becomes dependent upon the angle of inclination, decreasing as the enclosure changed from heated from below to heated from above with inclination angle, until conduction again becomes the dominant form of heat transfer. The flow structure can also change from a unicellular to multicellular depending on inclination angle. A wide range of further literature available into various enclosure shapes and configurations highlights the need to investigate these geometrical effects on flow structure and heat transfer as they typically cannot be fully understood using the knowledge gained from simpler geometric shapes. This can make predicting natural convection challenging in non-standard enclosure shapes and configurations.
CHAPTER 2. LITERATURE REVIEW

Partitioning

Internal structures which interact with the flow regime can have serious consequences on the advection of heat in an enclosure. They have the potential to restrict the movement of either the hot or cold air in an enclosure to a smaller region, limiting the heat transfer possibilities. Khalifa and Abdullah [78] showed how the heat transfer in an enclosure is reduced due to the presence of a partitioning structure but also how the location of the partition is important. In a simple rectangular room, a partition with an opening in the centre reduces heat transfer the least (8%) whilst an opening close to the side wall reduces it the greatest (17.5%) due to poor convective recirculation between the hot and cold zones. Increasing the size of the opening is also found to mitigate this effect. A similar study by Khalifa and Khudheyer [79] revealed a similar result, with the heat transfer reducing significantly inside the enclosure for the partitioned case. The effect of the aperture height in the partition was of main concern and was found to have a huge effect on the Nusselt number. For an aperture height of 2/3 of the enclosure height, the decrease in Nusselt number is 7%, whilst an aperture height of 1/3 the enclosure height led to a reduction of the Nusselt number by 41%. The effect of location of the partition inside the enclosure was found to be minimal compared to the aperture size and position.

A partially partitioned enclosure with differential heating was studied by Bilgen [80]. The positions of the partitions on the upper and lower surfaces were varied along with the Rayleigh number. Without partitioning a strong circulation exists, with the flow ascending up the hot wall, descending down the cool wall and travelling along the top and bottom surfaces as expected. Stable stratification exists within the centre of the enclosure. The effect of the partitions is to choke the circulation within the enclosure, reducing its velocity along the top and bottom walls. Removing the partition from the bottom of the enclosure reduced this choking effect and increased the heat transfer compared to the two partition case by approximately 8%, but still saw an overall decrease in the Nusselt number. Increasing the distance of the partition from the hot wall also decreased the Nusselt number by 4.5% due to a weakening of the circulation flow in the enclosure and again removing the bottom partition increased the heat transfer. This study is extended by Mahaputra et al. [81] to include the varying position and height of the partitions inside an enclosure for a natural
convection and mixed convection regime. One partition is located on the top wall, the other on the bottom. Increasing the partition heights from 20 – 40% of the enclosure height has a marked effect on the forced convection heat transfer, reducing it by up to 90%, compared to a reduction of 75% for natural convection dominated flows. Offsetting the partitions inside the enclosure leads to an enhancement in the heat transfer as the obstruction to the flow in the enclosure is reduced. Even for partition heights of 55% of the enclosure, mixing between the two sections of the enclosure exists when the offset is sufficient, although severely limited by the narrow channel dividing the sections. The presence of a partition in the path of a jet entering an enclosure also has the possibility to increase the mixing effect of the jet, due to an increase in the swirl of the flow and enhanced horizontal and vertical spreading beyond the partition Srinivansan et al. [82]. This would aid heat transfer in the region beyond the partition, and would be of benefit if some thermally sensitive equipment were to be located in this region.

2.1.2 Localised Heat Source

In practical applications enclosures rarely are empty cavities, but are used as housing for various components. Whilst some may be passive (from a thermal perspective) and merely prove to be an obstacle for the flow path within, oftentimes the component will bring its own heating effects to the enclosure, be it electrical wires passing through which heat up due to the joule effect, piping carrying hot or cold fluid through the cavity or even an electrical component dissipating heat internally. On their own these can produce very complex flow conditions due to recirculation and entrainment effects, the interaction with the walls of the enclosure, component shape, placement etc. which is further complicated when the walls of the enclosure are heated or cooled (or a non-standard enclosure shape etc.). This creates highly complex flow regimes which need to be understood thoroughly during design stages. This is especially true when the shape and configuration are different to those previously studied due to the complexities of the coupling of the velocity and thermal fields. It is prudent however, to try and identify the main sensitivities of natural convection to the placement of heated objects to allow for a greater understanding of their interactions in the enclosure.
CHAPTER 2. LITERATURE REVIEW

Enclosure Effects

Isolated heat sources have been comprehensively documented in the heat transfer literature and well-established correlations exist for a variety of surface and boundary conditions be they flat horizontal or inclined plates, spheres and cylinders etc. Cylinders in arrays also show an interdependency between each other as the heat transfer and flow field passing over a particular cylinder in the array is as a result of the conditions imposed on it by the cylinders downstream [83–85]. This will change depending on whether the array is in natural convection, buoyancy aided or opposing forced flow or cross flow. Typically the only cylinders which escape this effect are the ones which experience freestream conditions (i.e. bottom for natural convection, side for crossflows) as the presence of other cylinders does not usually have any marked influence on them.

This is also reflected in the enclosure effect on discrete heat sources. Koizumi and Hosokawa [29] investigated the presence of a confining conductive vertical wall on the natural convection regime associated with an isothermally heated cylinder. The velocity of the flow is severely reduced by the presence of the wall. Separation distance determined the interaction between the plume and the confining wall, changing from steady, to unsteady 2-D, to unsteady 3-D as $Ra$ increased. The effect of the distance of the cylinder from vertical confining walls was studied by Sadeghipour and Razi [86] for low Rayleigh numbers (650–100). An optimum position of approximately $W/D = 3$ was observed, where $W$ is the gap between the confining walls. A similar finding was observed by Ekundayo et al. [5] who found that the heat transfer was found to increase as the distance from the cylinder to the wall decreased with a maximum found at $x/W = 0.06$. This is found to be as a result of the increased flow between the cylinder and the confining wall and it is the resulting chimney effect which increases the convection heat transfer from the cylinder.

They also observed how the maximum heat transfer for a centrally located cylinder was at different vertical positions for different $Ra$, at $y/H = 0.25$ at $Ra = 7 \times 10^4$ and $y/H = 0.5$ at $Ra = 1 \times 10^5$. This is due to a dominance of the recirculation of fluid in the upper region of the enclosure for lower Rayleigh numbers and a reduced entrainment from the lower half of the enclosure as the cylinder height increases. The observed flow patterns for the varying cylinder position are presented in Fig. 2.4.
CHAPTER 2. LITERATURE REVIEW

Cesini et al. [28] found that the enclosure aspect ratio had a large influence on the resulting flow field present in the enclosure. For lower Rayleigh numbers, the average Nusselt number was at a maximum at the largest aspect ratio, whilst for higher Rayleigh numbers, the maximum average Nusselt number was found at the lowest aspect ratio. This was attributed to the increase in heat transfer due to convection in the lower part of the enclosure at high Rayleigh numbers and low aspect ratios. De and Dalal [87] also investigated the effect of the aspect ratio of a heated enclosure on a square, horizontal cylinder. It was found that for high aspect ratio enclosures, a multi-cellular flow existed. For the lower aspect ratio enclosure, more thermal stratification was noticed. The effect on the interior flowfield by the cylinder is much less pronounced and the enclosure behaves more like a differentially heated one. The influence of the cylinder position was found to be minimal on the local and average wall Nusselt number, the position only strongly influencing the flow structure within the enclosure.

From this review it is evident that there seems to exist an optimal positioning of a cylinder inside an enclosure for best heat transfer, with both distance from the wall and

Figure 2.4: Flow patterns observed in a square enclosure as a result of varying the internal cylinder positioning. From Ekundayo et al. [5]
height within the enclosure being important. Little work has been carried out with both a heated enclosure surface and a heat source inside the enclosure; it is usually the generalised case of differential heating of a single heat source alone, whereas a mixture of the two would represent a more physically realistic model for a real-world situation.

### 2.1.3 Enclosure Ventilation

The requirement for more durable and efficient electronics requires careful consideration of the thermal environment in which they operate. If the thermal environment is too aggressive for safe operation of the component, then some form of enclosure ventilation must be incorporated. Similar to the thermal and geometric configuration and the location of heat sources, ventilation is highly enclosure specific and must be carefully analysed in order to achieve the maximum amount of cooling possible. As such, it is necessary to understand the main types of ventilation along with how they perform under any given configuration. Ventilation of rooms and building spaces are considered here along with the cooling of heated and electronic components in enclosures as the findings and principles of optimisation are common between applications. The choice of ventilation configuration for a given enclosure type will dictate its effectiveness in cooling the interior of the enclosure and/or increasing the heat transfer from a surface or component. This too will be governed by the enclosure shape, heating configuration, the position of the heat source and ventilation opening(s) etc. It is useful to understand what is the optimum configuration for any given situation along with its limitations and sensitivities.

### Mixing and Displacement Ventilation

Typically, enclosure ventilation consists of one of two types: mixing or displacement. A combination of both modes is also possible if conditions allow. In mixing ventilation, the ventilation flow enters and exits the enclosure from one or more openings that are at the same height within the enclosure. This allows the ventilation flow (either forced or entrained) to enter and interact with the core fluid within, reduce its overall temperature before being removed again from the enclosure. Examples of mixing & displacement ventilation regimes are presented in Fig. 2.5 & Fig. 2.6.
Figure 2.5: Mixing ventilation examples for a heated object within a square enclosure. Depending upon the strategy required, ventilation can be either be in the form of single inlet/outlet opening (a), single inlet, single outlet (b), multiple inlet/outlet openings (c).

In displacement ventilation, there must be at least two openings in the enclosure at separate heights. This effectively produces a condition whereby the cooler air entering the enclosure at the bottom displaces the warm air vertically upwards before it is removed at the second vent. The inlet and outlet must be at significantly different heights, as it is quite difficult to initiate a pure displacement regime with two vents located only at an elevated height in the enclosure, except in the case where high temperature/contaminated air is already isolated to the upper region before ventilation. Displacement ventilation has the possibility of producing quite large thermal gradients, especially at lower levels of the ventilated space. This can lead to uncomfortable conditions in a buildings and office-space environments, where occupants can be subjected to cold conditions at feet level and warm conditions at head level. In equipment and electronics cooling however, this can be seen as quite advantageous if the components are placed in the lower region where these gradients are present, allowing for possible enhancements in convective heat transfer. In reality, there can typically exist a mixture of both forms of ventilation, with one more dominant than the other. Due to the general layout and orientation of multiple vent configurations, it is usually a displacement regime with some mixing element included. Effective displacement ventilation occurs when both the inlet and outlet openings see one flow direction across their boundaries only i.e. there is no back flow present. If this occurs then the efficiency of the exit for removing hot air from the enclosure decreases and effectively reduces the size of the vent area open for heat exhaustion.
Yu and Joshi [6] investigated the effect of ventilation on an enclosure with a heated object recessed into one wall of the enclosure. For a single vent located at the vertical wall opposite the heated component, an entrainment of exterior air into the enclosure is produced. As this passes over the component surface, a 30% increase in the average Nusselt number is noted, showing significant component cooling and enhanced heat transfer, shown in Fig. 2.7 (a). For the vent place at the top of the enclosure (Fig. 2.7 [b]), a cool jet of air enters the enclosure and effectively splits the enclosure into two distinct regions, one adjacent the component surface and one which recirculates in the opposite half of the enclosure. As such not all of the fluid entering the enclosure is used to cool the component, so some inefficiencies are present for this ventilation method. A vent located on the bottom surface of the enclosure did not alter the flow and thermal distribution within the enclosure to any significant extent. It was noted to be almost identical to the sealed enclosure scenario with just a small variation across the vent boundary itself due to some weak inflow and outflow present. As a result, the heat transfer from the component is unaltered.

Moving to a multiple vent configuration compared to a single opening provides this benefit of allowing for a direct path for the cool exterior air to enter the enclosure, interact with the heated surface or component before being exhausted at a separate vent as shown in Fig. 2.7 (c). This produced the greatest increase in component heat transfer. Tanny et al. [30] investigated changing the mode of ventilation from mixing to displacement by varying
the ratio of the open area of the lower to upper vents. For a vent ratio of 0 (lower vent closed), mixing ventilation dominates. As the ratio is increased towards unity, a mixture of the two modes is present until pure displacement ventilation occurs at a ratio of 1. It was found that increasing the vent ratio resulted in a decrease in the difference between the outside ambient air temperature and the average temperature within the enclosure. This increased the thermal efficiency of the ventilation based on the temperature distribution effectiveness (ventilation efficiency):

\[
\bar{\varepsilon}_j = \frac{T_{\text{out}} - T_{\text{in}}}{T_{\text{average}} - T_{\text{in}}} \tag{2.4}
\]

This efficiency increased from 0.86 at a vent ratio of 0, to 1.26 at a vent ratio of 1. The value of displacement efficiency is typically expected to be greater than 1 at a vent ratio of 1 due to the influence of displacement ventilation and the internal stratification present in the enclosure. Haslavsky et al. [7] also showed how the temperature difference between the inside and the outside of the enclosure is seen to be lower for the displacement method compared to the mixed method. This occurs for when the input power to the internal heated object is increased from 100W to 500W as shown in Fig. 2.8. The benefit of displacement ventilation over mixing was observed in both the transient response of the system from an initially unventilated enclosure as well as the steady state vented condition. Bolster et al. [88] commented on how displacement natural convection is a self regulating system, for when the heat load from an internal source increases, so too will the flow rate through the space. This results in no change in the interface between the cool inflow air and the
Figure 2.8: Reduction in enclosure $\Delta T$ between the average interior and exterior temperatures alternating between a pure mixing ventilation configuration ($R^* = 0$) and displacement ventilation ($R^* = 1$). From Haslavsky et al. [7]

warmer enclosure air. The response of the system is also observed to be in proportion to the amount of change in heat load, with a faster response for a large increase in the heat load and vice versa. Transient effects on the mixing and displacement of an enclosure with a point source are also noted by Fitzgerald and Woods [89].

A similar study carried out by Wang and Zhen [90] also showed the effectiveness of the displacement mode compared to the mixing mode. In a 3-D numerical study of a simulated office environment with numerous heat sources simulating computers, lamps, human bodies etc. distributed throughout the enclosure, the efficiency of the mixing mode was found to be 0.97, with the displacement ventilation giving an efficiency of 1.49, a 50% increase.

The displacement mode was also deemed to be more advantageous in an office situation as the air quality is enhanced by stratification, allowing the separation of the warm and contaminated air from the cool clean air at desk level and also had the possibility to reduce the energy consumption required by the ventilation system by up to 30%. A drawback of the efficiency of the displacement ventilation is the relatively steep temperature gradients in the region close to the desk level which could lead to an uncomfortable working environment for occupants. The problem of room air comfort is a major concern for building thermal engineers, with reports indicating that 24% of occupants have experienced some form of discomfort in rooms ventilated by the displacement method [91]. The introduction of a cooled ceiling will increase the temperature gradients higher up and reduce them at
desk height (to a maximum of 15%), allowing for a more comfortable working environment. Bouzinaoui et al. [92] showed how this stratification can be altered by changing the ventilation configuration of the room if it is carefully considered.

This also highlights the difficulty in predicting room ventilation when internal factors seen in a normal work environment are considered: the presence of people [31], desk partitions [93] etc., even the simple act of opening a door can disrupt the room ventilation [94]. The benefit of having people or heat producing devices such as personal computers in a displacement ventilation regime is that they act as “plume convectors” [31] whereby the cooler air from the lower region of the room can be entrained up via these heat sources to the warm air section without any mechanical excitation of the air. Mangier [95] also notes that since the supplied air temperature is only just below that of the average room air temperature in displacement ventilation and in a room environment the area of concern is in the occupied zone, the cooling load of occupied zone is lower than the whole space cooling load, resulting in the airflow needed to be introduced into the room to be smaller. This in turn leads to lower loads on the air conditioning unit and lower energy use overall. Awad et al. [96] showed how the temperature of the inlet flow helps to reduce the thermal gradient in the occupied zone compared to a cold inlet which is beneficial from an occupant comfort perspective.

Fitzgerald and Woods [89] showed how through careful selection of the ventilation configuration, an optimum configuration can be obtained that allows for effective cooling during summer months and ventilation in the winter that reduces the need to preheat the air due to mixing invoked in the lower region of the enclosure. It has also been shown to reduce the heat load to between 20–40% of that required for displacement ventilation in a winter office environment [97]. Flynn demonstrated how with effective flow control, a summertime ventilation system can be used as an efficient room heater in winter [98]. Similar results are also reported by Krajcika et al. [99]. The benefits of using a solar chimney to improve natural ventilation is also documented by Khanal and Lei [100]. Calay et al. [101] also showed how with a detailed knowledge of the contaminant distribution in an area, selective ventilation can be utilised to remove the contaminants at a specific height which is easier and more efficient to deploy than full scale room ventilation.
CHAPTER 2. LITERATURE REVIEW

Recent advances have involved using a chilled ceiling to aid in ventilation performance [102–104], but careful consideration must be made over flow control in order to eliminate any negative influences of natural convection at the ceiling. Novoselac and Srebric [105] showed how the cooling ceiling has advantages from greater thermal comfort and increase cooling capacity, with a greater cooling load from the ceiling reducing the vertical temperature gradient in the room. Control of the vertical airflow distribution is critical though as if incorrectly designed, contaminants can be suppressed in the breathing zone in the room. Taki et al. [104] showed how this may be achieved by using a honeycomb structure near the upper surface to help mitigate this effect and restore the pure displacement ventilation mechanism.

Inlet and Vent Considerations

In order to produce an optimal configuration for enclosure ventilation, care must be taken in locating the vent openings as these will directly affect the flow and temperature distribution throughout the space. The effect of changing the size of a single vent in the wall opposite a heated component was investigated by Yu and Yoshi [6]. It was found that a vent with a small size compared to the enclosure wall height (20% open area) contributed little to the change in flow within the enclosure due to poor entrainment into the enclosure and placement of the inlet vent. Increasing the open area of the wall to 70% resulted in an approximate reduction in the surface temperature of the component by 10%. This is due to larger entrainment through the bigger opening allowing for more cooler air to enter the enclosure and at a higher velocity. This effect increased slightly for an increase in Rayleigh number, with a 9.1% decrease in component temperature at a Rayleigh number of $10^5$ and a 9.6% decrease at $10^6$ compared to the sealed enclosure. This is not a particularly meaningful optimisation as opening 70% of a wall surface in practical terms may not be possible, primarily due to structural requirements of the enclosure. Bilgen and Yamange [106] also showed how the impact of the size of the vent opening was dependant on the Rayleigh number, with minimal impact as a result of increasing the inlet vent at low $Ra$, due to weak convection, which increased as $Ra$ increases. A 20% increase in Nusselt number was possible for a large inlet vent due to enhanced entrainment and an increase in
the average flow velocity in the enclosure. Conversely, a decrease in Nusselt number of 30% results when the outlet vent is increased to be much larger than the inlet vent when the heated object is close to the exhaust vent. This occurs due to a reduction in the flow velocity at the exit vent (up to 43%) which in turn decreased the temperature gradient over the heated surface, reducing heat transfer as a result. Gao et al. [107] reported that simultaneously increasing the size of each inlet and outlet vent by the same amount will lead to a constant increase in airflow across their boundaries. It is noted however, that once the size of one vent is larger than the other, increasing the size of the larger vent by any discernible amount will contribute little to an increase in airflow in the enclosure. Thus careful consideration and design must be carried out when determining the size (individual and relative) of the vents. Enclosure aspect ratio effects were also observed by Bilgen and Yamange [106] with both a tall and wide cavity reporting reduced heat transfer (50% lower) and flow velocities resulting from choking of the flow within the enclosure which stifled ventilation compared to the optimal enclosure aspect ratio of 1.

Dubovsky et al. [32] demonstrated how ventilation openings on the top of an enclosure with its upper surface heated limited the amount of exterior air entrained and exhausted leading to poor ventilation and little reduction in interior air temperatures compared to vent opening at different heights (inlet lower than exit). They also found that having a lower thermal resistance on the upper surface increased the heat loss through the wall to the detriment of the throughput of air through the enclosure. Matching the heat transfer coefficients of all the walls increased the throughput of air by 6 – 8%, leading to more effective ventilation of the enclosure via convection.

Mangier [108] investigated the inlet vent plume dynamics inside a ventilated room and showed how the flow typically dispersed upon entry. An understanding of this effect is required in order to determine the interaction of the ventilation flow with the room/enclosure and to evaluate the local air distribution and comfort for occupants, especially in the lower region people occupy and could also be important if the heat transfer of components in this region was a priority. Neilsen [109] showed how this is also dependant upon the flow rate to the room, the temperature difference and the Achimedes number, with a notable difference between different types of diffusers. Lee et al. [110] showed the difference between a
Figure 2.9: Interaction between the inlet and ventilation flow for varying outlet positions (a) and ventilation efficiency for increasing \( Re \) for the four cases (b) (Xaman et al. [8]).

Wall jet and ceiling diffuser on the ventilation and condition in a room with a contaminant source. Markedly different flow structures were observed. When the contaminant source was located at the centre of the room, then a ceiling diffuser was found to be optimal as high contaminate levels in the occupancy level \( H < 1.2m \) was only observed directly above the source. The contaminant was also well dispersed around the room. The ceiling diffuser was observed to be more efficient than a wall jet at reducing contaminate concentration in the room. Location of wall jet also was a determining factor on the spread of contaminants in the room, whereas the ceiling diffuser dispersed the contaminant evenly throughout the room. Lin et al. [111] also presented the effect of the inlet position on performance of displacement ventilation, with a more central located inlet preferential for the high heat loads encountered in a large room served by a single ventilation unit. Awad et al. [96] demonstrated how altering the vent locations can also increase (or decrease) the degree of isothermality in the lower region, leading to greater comfort for occupants.
In scenarios where natural ventilation is not sufficient at enclosure cooling, the room designer is forced to try and introduce some form of active component. This may be either from a forced inlet (jet) flow or mechanical excitation of the air (via fans etc.). *Xaman et al.* [8] studied the effect of forced convective ventilation on the ventilation efficiency of an enclosure subjected to varying exhaust positions. They found that an optimal position for the exit vent exists based upon the ventilation efficiency (Eqn. 2.4) and the Reynolds number of the flow. A maximum efficiency was observed when the interaction with the cool inlet jet and the interior was maximised combined with the ease of removal of the high temperature air. This change in flow structure and efficiency with outlet vent position and *Re* is presented in Fig. 2.9. *Noh et al.* [112] describe how increasing the inlet airflow velocity can produce a “piston-like effect” to help remove contaminants in the occupied zone subjected to mixing ventilation, while also helping to reduce the mean air age. *Deng et al.* [9] showed how the location of the flowpath close to the heat source is of importance in being able to remove heat from the enclosure, and that this is also sensitive to the flow Reynolds number. For buoyancy dominated flows, increasing *Re* can decrease heat transfer due to a reduction in the buoyancy contribution without any enhancement from the forced component. This decreases until it reaches a minimum where both natural and forced convection contribute equally to the heat transfer. Then forced flow begins to dominate and heat transfer increases.
As shown in Fig. 2.10, beyond \( Gr = 4 \times 10^3 \), forced convection decreases heat transfer compared to natural convection for the range of \( Re \) studied. As a result, the interaction between \( Re \) and \( Gr \) plays a vital role in the removal of heat from the enclosure and as noted by the authors, increasing the forced ventilation flow may not always prove beneficial to the heat transfer characteristics, as perhaps one would intuitively expect. A similar study by Beya and Lili [113] for oscillatory flows within a geometrically identical enclosure observed much the same trends. They discovered the onset of a periodic oscillatory regime for increasing \( Re/Gr \). This periodic structure altered the flow regime within the enclosure which increased the Nusselt number by up to approximately 21% compared to the steady flow structure.

Ventilation was analysed by Angirasa [114] for a number of different values of the ratio \( Gr/Re^2 \), ranging from 0.1, indicating forced convective flow, to 10 for buoyancy driven flow. The influence of the forced flow by aiding the buoyant flow and opposing it was also considered. For aiding flow, the local Nusselt number was found to be at a maximum at the inlet, and a minimum towards the top vent, due to sharper temperature gradients present at the inlet compared to the vent. As the Reynolds number of the forced flow increases, however, the local Nusselt number is seen to decrease at the inlet and increase towards the vent. The average Nusselt number for the system was found to remain the same for the varying Reynolds number. For the opposing flow, the wall temperature gradients are greatest at the inlet, decreasing towards the centre of the enclosure and then increasing again towards the outlet vent. For buoyancy dominated flow, the heat transfer is low due to the dominance of a recirculation zone present in the centre enclosure. The forced flow is deflected around this to the far region of the enclosure and as a result low temperature gradients exist at the wall surface, resulting in poor heat transfer. As the Reynolds number of the flow increases, this recirculation zone becomes reduced and confined to the near wall region, with the forced flow dominating the enclosure, leading to an increase in the heat transfer from the wall.

Raji and Hasnaoui [115] investigated the effect of varying the emissivities of the walls on ventilation of an enclosure. The increase in the emissivity of the walls leads to an increase in the temperature of the walls within the enclosure. This alters the flow regime
in the enclosure and leads to the formation of closed recirculations within. The change in the emissivity of the walls also leads to the formation of thermal gradients along the wall of the enclosure which are not present in the non-radiative case. These gradients alter the flow structure along the walls and have a tendency to reduce the convection within the enclosure. The radiative effect will produce a more uniform distribution of temperatures within the enclosure. This results in a reduction in the maximum temperature within the enclosure and an increase in the average temperature. This allows for a 40% increase in heat transfer at a Reynolds number of 50, and an increase of 23.4% at \( Re = 5000 \). Heat transfer by radiation increased as the wall emissivity increased and decreased with an increase in Reynolds number. At low \( Re (Re = 50) \) radiation will contribute to approximately 60% of the total heat transfer. Higher values of \( Re (Re = 5000) \) leads to a decrease in the effect of radiation but still remains significant at approximately 20% of the overall heat transfer for large wall emissivity. Thus, an optimum value of emissivity and \( Re \) is possible to be achieved for the given situation, but is not noted by the authors.

### 2.2 Summary

It is clear that while enclosure natural convection has received a significant amount of attention in order to understand the heat transfer and internal conditions present, gaps in the knowledge still remain as a result of its high sensitivity to enclosure conditions and the coupling of the velocity and thermal fields. From the review presented here, various parameters such as the thermal boundary conditions, enclosure aspect ratio, shape and geometric configuration all influence the natural convection system established in an enclosure.

The following gaps in the literature are observed when considering a wing leading edge enclosure:

- The influence of a non-standard enclosure geometry is not well documented. Whilst shapes such as square, rectangular, triangular trapezoidal, circular etc. have been extensively investigated, the highly curved leading edge shape presents a configuration which has not been presented before, nor is easily approximated by any other shape.

- The effect of heating from above is only considered when other effects are present.
such as a conductive vertical wall or vertical temperature gradient, as without them, no convection will occur and stratification is dominant. As such the heat transfer and flow conditions are unique to the configuration of the enclosure when heated from above. In the leading edge this occurs as a result of the curved upper heated surface and the vertical conductive wall, a configuration which has not been presented before.

- The interaction between a discrete heat source and its enclosure has been documented, with enhancements possible depending on location and distance to a confining wall. The influence of confinement within the curved leading edge subjected to exterior heating is unclear from the published literature however.

Previous investigations have also illustrated how heat transfer from an enclosure can be improved as a result of ventilation. Placement and number of ventilation openings, their relative size, the degree of mixing or displacement ventilation and internal configuration of the enclosure have been shown to be important upon the resultant ventilation efficiency. This is particularly true given the limiting geometrical conditions imposed by the construction of a typical aircraft wing leading edge. This thesis aims to address the deficit of knowledge concerning:

- The impact of ventilation upon the conditions within the curved leading edge enclosure, including the confinement of the ventilation openings to the lower region of the enclosure.

- The influence of bleed duct position relative to the ventilation path. It has been shown how the position of the heat source relative to the flow path is important for optimising heat transfer. This needs to be addressed to determine if an optimum configuration exists for the leading edge enclosure shape.

- Internal effects such as partitioning need to be addressed as these represent typical obstacles present in industrial applications and whilst they are known to reduce heat transfer due to the confinement of high temperature fluid to a smaller region of the enclosure, its effect upon the ventilation path in the leading edge is unknown.

To the best of the authors knowledge, no such investigation has been presented previously and this gap in literature is addressed in this thesis.
Chapter 3

Experimental Methodology

This chapter describes the experimental apparatus, methods and instrumentation used on a representative aircraft wing leading edge section for heat transfer and flow field analysis. An outline of experimental test facility components and measurement apparatus along with detailed description of the methodologies used are included.

3.1 Leading Edge Test Section

A leading edge test section was constructed in order to replicate the enclosure geometrical and thermal configurations seen in an actual aircraft wing during operation. The test section measured 550mm (W) x 600mm (D) x 300mm (H) and is presented in Fig. 3.1. The aircraft skin was constructed from a 3mm thick aluminium sheet into the shape of the leading edge. The vertical rear wall (representing the front spar of the wing) is made from 10mm thick aluminium. The reference geometry for the leading edge supplied by Airbus based upon a generic aerofoil shape is included in Appendix C.

- Two separate rubber backed ELEMEX 500W heater mats were attached to the upper and lower surfaces of the aluminium (as shown in Fig. 3.1 (a) & (b)). These allow the upper and lower surfaces to be independently heated to a defined setpoint temperature via a Eurotherm 2216e PID (Proportional-Integral-Derivative) controller. Steady state temperatures are $100^\circ C (T_h)$ and $70^\circ C (T_c)$ respectively for the upper and lower surfaces.
The surfaces are joined at the top and bottom of the vertical wall via a series of 5mm hex head bolts through a flange on the 3mm aluminium. Conduction between the upper and lower heated surfaces to the vertical wall was inhibited through the use of a 2mm rubber insulating strip ($k = 0.275 \text{W/mK}$) at the attachment point between surfaces. This mitigated any direct heating of the vertical wall via conduction from the applied boundary conditions on the upper and lower surfaces. In a typical aircraft leading edge configuration, conduction can occur between these surfaces as a result of riveting and/or integrally machined panels. This is not accounted for in the present study due to the additional complexity introduced to the heat transfer mechanism in the leading edge as a result. The configuration presented in this section was chosen.
CHAPTER 3. EXPERIMENTAL METHODOLOGY

to study the effects on the isolated rear wall only, meaning that heat transfer to the vertical wall was predominately via radiative and convective means from the rest of the enclosure. As such, any change noted in the temperature distribution along the wall surface between enclosure configuration should be as a result of a change in the internal environment and not influenced by any interaction with the upper and lower heated surfaces.

• The interior surfaces of the test section were coated with a matt black paint to replicate the high emissivity of the interior of an actual wing leading edge as a result of the application of an anti-corrosion treatment. The emissivity was measured via a Fluke FLK-Ti25 thermal imaging camera to be 0.96 (±0.1). This was obtained by using a calibrated thermocouple to measure the surface temperature (40°C, 60°C, 80°C & 100°C). Then by acquiring test images of the region adjacent to where the thermocouple was located, the emissivity value on the FLK-Ti25 was varied until the correct surface temperature was obtained. The high emissivity and temperatures of the surfaces in the enclosure meant that convective and radiative contributions are coupled and play an important role in the steady state enclosure conditions. As such, the influence of radiation must be included in the analysis of the heat transfer regime present. The method used to account for this is detailed in Section 3.2.

• A 30mm (D) × 280mm (H) channel was cut into the rear wall at mid-depth and sealed with glass to allow for laser access into the leading edge during PIV testing. This is highlighted in Fig. 3.1 (b).

To minimise any unwanted heat loss to the surroundings, the enclosure was covered with approximately 50mm of fibreglass insulation. An opening was made in the insulation on the vertical wall to allow for entry of the laser lightsheet into the enclosure when PIV (Particle-Image-Velocimetry) was carried out. Where the end of the enclosure was open for optical access to the flow-field, double glazed glass with a 3mm air gap was used to minimise heat loss from the enclosure. The mounting points of the cylinder were also insulated using fibreglass insulation and made of wood to inhibit any conductive losses from the cylinder.
to the enclosure.

### 3.1.1 Outer Enclosure

In order to adequately seed the enclosure with particles necessary for local air velocity measurements, an exterior chamber was constructed into which the particles could be introduced without loss to the environment. This allowed the particles to achieve a neutral buoyancy with the ambient air within the leading edge. This ensured that any motion of the particles was due to the movement of the airflow and not any inherent buoyancy of the particles themselves. The leading edge was supported upon four standing metal legs, which were insulated at the connection points by a 2mm rubber strip, inhibiting conduction losses. The outer enclosure measured 1200mm (H) x 1000mm (W) x 600mm (D). The horizontal centreline of the leading edge was placed at a height of 500mm from the bottom of the enclosure. The outer enclosure containing the leading edge is presented in Fig. 3.2.

During the ventilation testing, the air within the outer enclosure was found to increase to approximately 30°C, which then remained constant during the testing run. This is an increase in the inlet temperature with respect to the ambient air temperature in the laboratory,
which was typically between 18 – 20°C. However, such an elevated inlet air temperature into the enclosure was beneficial as they provided for inlet temperatures more appropriate to those seen in the environments the experimentation is intended to replicate (i.e. high ambient temperature locations). The temperature at the inlet vent was recorded at steady state for each test and was used as the input temperature to the enclosure during the analysis.

### 3.1.2 Internal Configuration

A number of different geometric configurations within the leading edge were investigated and their influence on the temperature distribution, heat transfer and flowfield was determined. The presence and position of a constant temperature and constant heat flux cylinder (simulating a bleed duct passing through the leading edge), the presence of a partition (representing a sub-spar structure) and ventilation openings on the upper and lower surfaces.

To replicate the bleed duct, a Ø50mm × 570mm(D) horizontal cylinder was used.

- For the constant temperature case, a hollow aluminium cylinder (wall thickness 2mm) with an Elemex 250W rubber backed heater mat attached to the outer surface. The inner cavity of the cylinder was filled with fibreglass insulation and was mounted to the outer surfaces of the enclosure via insulated wooden blocks which inserted into
the cylinder cavity. A schematic of this is provided in Fig. 3.5 (a). Air leaving the high stage compressor in a typical gas turbine engine which is transported through the bleed duct can be up to 250°C, which means that upon insulating the bleed duct sufficiently, the surface temperature is found to be approximately 135°C, as measured by AIRBUS. This is used as the setpoint temperature for the constant temperature condition. Three calibrated k-type thermocouples were placed (Fig. 3.10), one at \( D = 285\, mm \) on the upper surface on the vertical centreline which was used as input to the PID controller and one on either side the horizontal centreline at \( D = 135 \) & 435\,mm to ensure isothermal conditions.

- During testing, the position of the bleed duct varies from bottom left (50mm from lower and vertical enclosure walls), top left (50mm from upper and vertical enclosure walls) and bottom right (50mm from lower enclosure and partition walls). These are presented in Fig. 3.4.

- For the constant heat flux case, a solid aluminium cylinder with a Ø20\,mm channel in the centre into which a 1000W Elemex 240V AC cartridge was inserted. The cylinder was held in position with mounting points made from a right angle steel bracket placed upon an insulated wooden base. This is detailed in Fig. 3.5 (b). Eight calibrated k-type thermocouples were placed on the cylinder surface, located as per Fig. 3.10 to capture the temperature distribution on the cylinder.

The second heat generating component investigated is the route of electrical wires passing through the structure. This collection of wires will have an elevated surface temperature due to a power supply passing through them and this is represented by a second hollow aluminium cylinder (Ø75\,mm) which is instrumented identically to the Ø50\,mm one. The Ø75\,mm cylinder (\( C_2 \)) was placed in the enclosure adjacent to the vertical wall such that the gap between itself and the vertical wall was the same as for the smaller cylinder (25\,mm). Its centre is placed at the enclosure mid-height. For these tests, the location of the smaller cylinder was varied between the lower and upper left positions, as shown by points A and B of Fig. 3.6. The Ø75\,mm cylinder remained in the same position for all tests where it was present.
CHAPTER 3. EXPERIMENTAL METHODOLOGY

Figure 3.4: Cylinder locations for the single cylinder test cases: bottom left (A), top left (B) and bottom right (C)

Figure 3.5: Mounting method for hollow Aluminium cylinder (a) and solid Aluminium cylinder (b)

The location of the partitioning subspar within the enclosure is also illustrated in Fig. 3.4. To permit the movement of air between the front and rear partitioned spaces, a gap
CHAPTER 3. EXPERIMENTAL METHODOLOGY

Figure 3.6: Cylinder locations for the two cylinder investigation cases. The Ø75mm cylinder’s position remains constant whilst the Ø50mm cylinder is placed above and below it adjacent to the vertical wall. The Ø50mm cylinder positions are shown in Fig. 3.4.

was left at the top of the partition which was the same size as the openings made to the enclosure surfaces (50mm). In order to allow lightsheet entry into the front section of the leading edge, this partition was chosen to be made from 8mm clear glass. The partition was insulated from the lower surface via a 2mm rubber strip, identical to that used at the rear wall junctions. Similar to the rear wall, conduction from the lower surface to the subspar is possible in a typical aircraft leading edge, however this effect is not considered here due to the additional complexity to the heat transfer introduced. The influence of the subspar is limited to blocking the convective flowpath as intended.

3.1.3 Ventilation

The leading edge enclosure was constructed in a manner which allowed for a number of different ventilation configurations to be investigated. Three openings can be made in the leading edge, two on the lower surface (one at the rear directly below the bleed duct and one at the front of the enclosure) and one on the upper surface, directly above the bleed duct. An outline of the ventilation openings for the constant bleed duct heat flux cases is shown in Table 3.1 and for the constant temperature bleed duct cases in Table 3.2.

The openings were made the entire depth of the enclosure to ensure a 2-dimensional flow was present. The openings were also covered with a mesh grid which reduced the open area to 60% of the vent size. Fig. 3.7 (c) shows the difference between a fully open
Figure 3.7: (a) Vented leading edge enclosure with openings on the upper and lower surfaces, (b) Vented leading edge enclosure with both openings on the lower surfaces, (c) Top vent opening with half of the mesh grid removed to illustrate difference in open area between a fully open vent and one with mesh attached. (1) Bleed Duct. (2) Glass Pane. (3) Upper Vent. (4) Lower Rear Vent. (5) Lower Front Vent.

and meshed opening on the upper surface for comparison. The openings were 50mm in width and 580 mm in depth.

Table 3.1: Outline of the constant wall heat flux bleed duct configurations. Locations are illustrated in Fig. 3.4 and Fig. 3.6.

<table>
<thead>
<tr>
<th>Case</th>
<th>Bleed Duct Location</th>
<th>$C_2$</th>
<th>Rear Bottom Vent</th>
<th>Front Bottom Vent</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Bottom Left</td>
<td>-</td>
<td>Closed</td>
<td>Closed</td>
</tr>
<tr>
<td>2</td>
<td>Bottom Left</td>
<td>Present</td>
<td>Closed</td>
<td>Closed</td>
</tr>
<tr>
<td>3</td>
<td>Upper Left</td>
<td>Present</td>
<td>Closed</td>
<td>Closed</td>
</tr>
<tr>
<td>4</td>
<td>Bottom Left</td>
<td>-</td>
<td>Open</td>
<td>Open</td>
</tr>
<tr>
<td>5</td>
<td>Upper Left</td>
<td>-</td>
<td>Open</td>
<td>Open</td>
</tr>
</tbody>
</table>
Figure 3.8: Variation in $R^*$ used in testing as a result of changing the open area of the front and rear vents for Case 4

Table 3.2: Outline of the constant wall temperature bleed duct configurations. Locations are illustrated in Fig. 3.4 and Fig. 3.6.

<table>
<thead>
<tr>
<th>Case</th>
<th>Bleed Duct position</th>
<th>Rear Bottom Vent</th>
<th>Front Bottom Vent</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>-</td>
<td>Open</td>
<td>Open</td>
</tr>
<tr>
<td>B</td>
<td>Bottom Left</td>
<td>Open</td>
<td>Open</td>
</tr>
<tr>
<td>C</td>
<td>Bottom Left</td>
<td>Open</td>
<td>Open</td>
</tr>
<tr>
<td>D</td>
<td>Adjacent to Partition</td>
<td>Open</td>
<td>Open</td>
</tr>
<tr>
<td>E</td>
<td>Upper Left</td>
<td>Open</td>
<td>Open</td>
</tr>
</tbody>
</table>

The relative ratio of vent open areas in a multiple vent configuration has been seen to have a significant effect upon the ventilation [7, 30, 90], particularity when it involves transferring from one mode of ventilation (mixing or displacement) as a result. In order to investigate such a scenario, the openings on the lower surface are altered based upon the vent open area ratio ($R^*$)

$$ R^* = \frac{A_{\text{front}} - A_{\text{rear}}}{A_{\text{total}}} $$  \hspace{1cm} (3.1)

This alters the ventilation type from a rear-biased configuration ($R^* < 0$) to a forward biased one ($R^* > 0$). An illustration of this is presented in Fig. 3.8. This was achieved by blocking the openings using a combination of long wooden strips (580mm × 10mm) over the openings which were held in place using silicone sealant.
3.1.4 Instrumentation

Thermocouple Placement

The leading edge was instrumented in order to measure and record the steady-state temperatures within the enclosure. Surface temperature measurements were also measured and recorded. A view of the measuring equipment is shown in Figure 3.9. Temperature measurements were made using k-type thermocouples which were calibrated to 0.1°C with a Laude RM6 thermal bath.

- For the surface temperature measurements, the thermocouple bead was attached to the surface with an epoxy resin to secure the thermocouple in position and prevent any convective effects on the thermocouple readings. Surface temperature measurements were taken on the vertical wall, the cylinder surface and the upper and lower heated surfaces. Seven thermocouples were placed along the height of the vertical wall, three thermocouples were placed upon the heated upper and lower surfaces; one connected as feedback to the PID controller and the other two to ensure isothermal conditions along the surface. The positions of the thermocouples on the bleed duct surface was detailed in Section 3.1.2.

- To measure the interior air temperatures, up to thirty one thermocouples were distributed throughout the enclosure along three horizontal and three vertical planes ($X^* = 0.25, 0.5, 0.75$). Five thermocouples were also placed along the inlet and outlet vents as well as four in the outer enclosure to measure the ambient air temperature. This allowed for a detailed description of the thermal distribution within the enclosure to be obtained during testing. The locations of the thermocouples are illustrated in Fig. 3.10.

- The thermocouples were connected to two SR630 thermocouple dataloggers which could monitor up to 16 individual channels per device and an Agilent 34970a. The Agilent system was interfaced to a desktop PC via a RS-232 serial cable and the Aglient IO Libraries Suite 14.2. The datalogger was controlled via the Benchlink Datalogger 3 software and datapoints were recorded at 10 second intervals.
Figure 3.9: Schematic of the heating and measuring equipment used throughout testing of the leading edge. (1) Personal computer for recording data, (2) Stanford Research Systems SR630 dataloggers, (3) HP Agilent 34970a modular datalogger, (4) Outer enclosure, (5) Leading edge test section, (6) Eurotherm 2216e PID controllers, (7) 240V adjustable autotransformer, (8) Digital Multimeter

Figure 3.10: Thermocouple locations along the three horizontal and vertical planes ($X^*, Y^* = 0.25, 0.5, 0.75$) and on the vertical wall surface. Measurements recorded at mid-plane ($D = 150mm$) of leading edge enclosure.
CHAPTER 3. EXPERIMENTAL METHODOLOGY

Surface Heating Control

- The Ø50mm and Ø75mm hollow cylinders, upper and lower surface heating mats are each controlled by individual PID controllers. These supplied a constant surface temperature boundary condition via a Eurotherm 2216e controller set to a controllable setpoint.

- For the constant heat flux bleed duct, a Ø20mm cartridge heater supplied the required power to the Ø50mm aluminium cylinder via ohmic dissipation from an AC power supply. This power supply produced a variable power input as required via an adjustable autotransformer (Variac) from a single phase 240V supply. The voltage supplied and current in the cartridge heater lead wire was measured using a calibrated digital multimeter and recorded the power supplied to the heater.

3.2 Steady State Temperature Measurements

3.2.1 Introduction

The horizontal cylinder (Ø50mm) in the leading edge is subjected to two different heating mechanisms. The first is a constant surface temperature condition. This was achieved using the heating mat attached to the cylinder surface as previously described in Section 3.1 and was heated to a predefined setpoint. This kept the bleed duct surface temperature constant across the various enclosure configurations tested. This method was employed to replicate the actual conditions encountered within the leading edge, where the high thermal mass and insulation of the bleed duct means that its surface temperature remains constant across a wide range of operating conditions. A constant temperature surface boundary condition most accurately replicates this condition.

The second method employs a constant heat flux from the bleed duct. This is provided by a constant input from a power source to the cylinder. This configuration allows for characterisation of the heat transfer ($Nu_{BD}$) from the cylinder to the leading edge for a range of power inputs, a scenario which has not been extensively investigated for the current enclosure shape.
Throughout the analysis, the non-dimensional temperature within the enclosure is defined as

\[ T^\star = \frac{T - T_c}{T_h - T_c} \]  

(3.2)

where the subscript \( c \) denotes the coldest temperature seen in the enclosure and \( h \) denoted the hottest surface temperature. For the empty enclosure this is the upper surface temperature, for all other cases it is the bleed duct surface temperature. \( T_c \) is the boundary condition imposed on the lower enclosure surface (70°C) for all cases.

The material and fluid thermal properties are taken at the film temperature, defined as

\[ \frac{T_{BD} - T_{ref}}{2} \]  

(3.3)

### 3.2.2 Constant Temperature

The bleed duct surface temperature was held at a constant temperature as specified by Airbus (135°C). The system was allowed to reach steady state, at which point the temperature distribution of both the interior air and wall surfaces were recorded. Steady state conditions were defined as no more than a 0.5K change in temperature per hour.

For the ventilated scenario, a number of parameters are employed to assess the impact and performance of enclosure ventilation on the thermal distribution within the enclosure. These temperature ratios are based upon the average interior air temperature and the temperature at the inlet and outlet of the enclosure. \( T_R \) is a measure of the temperature rise of the air exiting the enclosure compared to that entering and is defined as follows:

\[ T_R = \frac{T^\star_{exit}}{T^\star_{inlet}} \]  

(3.4)

This determines the influence of the ventilation based upon the air entering and exiting the enclosure only. A \( T_R > 1 \) value is expected in any effective ventilation strategy and designers should look to maximise this value as it indicated a greater amount of high temperature air removed form the enclosure as \( T_R \) increases. It does not give any clear insight into the interaction with the core bulk fluid of the enclosure as this value is not included
in this parameter. This is important as a high exit to inlet ratio alone does not necessarily mean that ventilation is optimised, merely that the exit is higher than the inlet, which as noted before is only to be expected in any ventilation scenario. To overcome this $T_A$ and $T_I$ are also defined which both include references to the inter air temperature.

$T_A$ presents the temperature ratio of the air exiting the leading edge with respect to the average temperature within the enclosure

\[ T_A = \frac{T_{\text{exit}}}{T_{\text{average}}} \]  

(3.5)

In this case, a change in $T_A$ will indicate the amount of change in the interior air as a result of the ventilation flow into the enclosure and can give an insight as to the degree of interaction of the inlet air with the interior. If the exit temp is low compared to the average, even though $T_R$ might seem favourable, then it is clear that the ventilation strategy is not optimal as the air is exiting the enclosure without properly mixing with or displacing the warmer air within. A $T_A$ value below 1 would indicate such a case, whereas $T_A \geq 1$ indicates that the air temperature exiting the enclosure is at or greater than that which occupies the bulk of the enclosure space. This would indicate a more optimised ventilation strategy as is suggests that there is no confinement of any high temperature air in the enclosure, leading to an increase in the average temperature compared to that at the exit.

In natural convection, vent placement will play an important role upon both $T_R$ and $T_A$, as the height of the exit vent relative to the overall height of the enclosure determines the ability to remove the air and unless the flow-path is accurately controlled by geometric effects, it can be quite challenging to remove the more buoyant high temperature air if the exit vent is positioned low in the enclosure.

To overcome the limitations of using the individual temperature ratios when assessing ventilation performance, the temperature efficiency of ventilation as described by [30] and [116] is used. This combines the effect of the inlet, outlet and average enclosure air temperatures and produces a global ventilation efficiency for the enclosure based upon these parameters. It is defined as follows
This allows for the entire ventilation regime to be included in one single variable as it accounts for both the temperature rise between the inlet and outlet flow and the impact ventilation has upon the core bulk temperature of the enclosure. Ultimately it is this variable which is the most important factor in determining the effectiveness of a particular ventilation strategy. However, in comparison of individual ventilation cases, an understanding of each of the temperature ratios $T_R$ and $T_A$ is necessary in order to decide upon the most effective method to improve ventilation efficiency. It is worth noting that the most efficient ventilation strategy comes when both $T_R$ and $T_A$ are at a maximum.

In order to compare the heat exhausted from the enclosure for individual ventilation / geometrical configurations, a heat flux based upon an enthalpy change between the inlet and outlet flow is calculated. This allows for comparison between each of the individual cases in order to assess the impact of cylinder position, partitioning etc. upon the amount of heat exhausted from the enclosure. It is defined as

$$Q_{ex} = \dot{m}C_p(T_{exit} - T_{inlet})$$

(3.7)

where $\dot{m}$ is the mass flow of air crossing the exit vent boundary, and $C_p$ is the specific heat capacity of the air taken at the film temperature.

The mass flow is calculated using the airflow velocity normal to the vent boundary (Eqn. 3.8) as shown in Fig. 3.11

$$U_n = V_x \cos \theta + V_y \sin \theta$$

(3.8)

and obtained using Eqn. 3.9. $A$ is the cross sectional area of the vent open along the depth of the enclosure and the air density ($\rho$) is taken at the film temperature (Eqn. 3.3).

$$\dot{m} = \rho A \int U_n$$

(3.9)
CHAPTER 3. EXPERIMENTAL METHODOLOGY

3.2.3 Constant Heat Flux

For the constant heat flux condition, the horizontal cylinder was changed to include a Ø20mm cartridge heater placed inside the aluminium Ø50mm body. This supplied the required power to the cylinder via ohmic dissipation from an AC power supply. This power supply produced a variable power input to the cylinder as required via an adjustable auto-transformer (Variac) from a single phase 240V supply.

Taking an energy balance upon the cylinder (as illustrated in Fig. 3.12) produces

\[ q''_\Omega = q''_{\text{cond}} + q''_{\text{conv}} + q''_{\text{rad}} \] (3.10)

The end faces of the cylinder were insulated with fibreglass insulation between it and the front and back walls of the enclosure where it is mounted onto. A 2mm thick rubber pad was also placed on the enclosure wall where the cylinder insulation meets it. This lack of an air gap at the end of the cylinder inhibits any convective losses from these faces and the insulation inhibits conductive losses from the cylinder to the enclosure. Neglecting conductive losses through the lead wires also simplifies this expression to

\[ q''_{\text{conv}} = \frac{VI}{A} - q''_{\text{rad}} \] (3.11)
Figure 3.12: Heat flux acting upon the horizontal cylinder when supplied with a constant heat input from the power supply

where $V$ and $I$ are the power supply input voltage and current respectively and $A$ being the total surface area of the cylinder. This assumes that all the heat flux supplied to the aluminium cylinder via the cartridge heater is evenly conducted radially through the material to the cylinder surface where it is available for heat transfer via radiation and convection.

In order to isolate the convective heat loss from the cylinder, it is first necessary to determine the heat flux that is lost from the cylinder surface via radiation. Typical radiation analysis is based upon the standard equation for heat loss from a non-blackbody to its surroundings

$$q = \varepsilon \sigma A \left( T_h^4 - T_c^4 \right)$$

(3.12)

which assumes that both the hot and cold surfaces are of constant temperature and are of sufficient size and distance to ensure that the thermal radiation is evenly distributed to the surroundings from the heated body. This may be useful for a first approximation of the radiation heat exchange or where the surroundings are of sufficient size to neglect any difference in thermal conditions present along its surfaces. Its accuracy is questionable where the position of the heat source is positioned eccentrically, where large thermal gradients are present between surfaces of the surroundings or in a non-standard enclosure shape as it does not account for any of these conditions when using this method.
Figure 3.13: Resultant resistance network for two surfaces in radiative balance which exchange heat only between each other.

To this effect, an analysis procedure is used which builds upon the radiation network analysis method outlined by Oppenhiem [117]. It has been shown that the net energy leaving the surface of a non-blackbody of emissivity $\epsilon$ is

$$q = \frac{E_b - J}{(1 - \epsilon)/\epsilon A}$$  \hspace{1cm} (3.13)

where $E_b$ is the blackbody emissive power and $J$ is the radiosity of the surface. For two surfaces in radiative balance, the net exchange of energy between them is

$$q_{1 \rightarrow 2} = \frac{J_1 - J_2}{1/A_1 F_{12}}$$  \hspace{1cm} (3.14)

Thus, for a two surface radiative exchange situation, a network element can be produced which combines the surface resistance of the bodies (Eqn. 3.13) and the space resistance (Eqn. 3.14). The resultant network is shown in Fig. 3.13.

Therefore, the net heat transfer between the surfaces becomes the potential difference over the total resistance across the network

$$q_{net} = \frac{E_{b1} - E_{b2}}{(1 - \epsilon_1)/\epsilon_1 A_1 + (1/A_1 F_{12}) + (1 - \epsilon_2)/\epsilon_2 A_2}$$  \hspace{1cm} (3.15)

To apply this method to the bleed duct, the enclosure surfaces are split into distinct regions where their surface temperature and position are known. On the vertical wall, the positions of thermocouples and their temperature reading define the sections into which the surface is divided. Along the remainder of the enclosure, the surfaces are split into regions where the temperature is known (on the upper and lower heated surfaces), where the ventilation openings are positioned and where the unheated section exists. This produced 17 individual segments on the enclosure walls. The surface of the cylinder is then divided into
sections which correspond to the divisions on the exterior surface. Fig. 3.14 outlines the interaction between each of the individual segments on the cylinder and enclosure surfaces.

The net heat transfer between the individual section and its corresponding wall section thereby becomes

\[ q_{i_{rad}} = \frac{\sigma (T_{cyl}^4 - T_{i_{wall}}^4)}{\left(1 - \epsilon_{cyl} \right) \epsilon_{cyl} A_{icyl} + \frac{1}{A_{icyl} F_{icyl \rightarrow wall}} + \frac{1 - \epsilon_{wall}}{\epsilon_{wall} A_{iwall}}} \]  

(3.16)

At each section, the heat flux is transmitted radially outwards towards the corresponding region on the enclosure surface. All the energy leaving the cylinder is absorbed by its corresponding section and as such, the shape factor between each of the individual sections \((F_{icyl \rightarrow wall})\) is 1. When this is totalled across the circumference of the cylinder surface \((c)\), the total radiative heat flux from the cylinder becomes

\[ q''_{rad} = \frac{1}{A_{cyl}} \sum_{c=0}^{c=\pi d} \sigma (T_{cyl}^4 - T_{i_{wall}}^4) \left(1 - \epsilon_{cyl} \right) \epsilon_{cyl} A_{icyl} + \frac{1}{A_{icyl} F_{icyl \rightarrow wall}} + \frac{1 - \epsilon_{wall}}{\epsilon_{wall} A_{iwall}} \]  

(3.17)

Where openings exist in the enclosure as for the ventilated cases, the radiative properties of the exterior are taken to be that of an infinite surface of \(\epsilon = 1\) at a temperature equal to the wall temperature.
CHAPTER 3. EXPERIMENTAL METHODOLOGY

to that of the ambient surroundings. This allows for an accurate account of the radiative loss to the environment to be included in the analysis, something which would be difficult, if not impossible to obtain with Eqn 3.12 alone.

The convective and radiative heat flux are normalised with respect to the overall input power in order to compare their contribution to the overall cylinder heat transfer

\[
q^*_{\text{conv}} = \frac{q''_{\text{conv}}}{q_{\Omega}}; \quad q^*_{\text{rad}} = \frac{q''_{\text{rad}}}{q_{\Omega}}
\]  (3.18)

The Nusselt number is based on the convective heat flux from the cylinder. The length scale is chosen to be the cylinder diameter, the temperature difference is based upon the cylinder surface temperature and a chosen reference temperature. The thermal conductivity is taken at the film temperature (Eqn. 3.3).

\[
Nu_{\text{conv}} = \frac{\left(\frac{q''_{\text{conv}}}{q_{\text{ref}}}\right) d}{k(T_{\text{cyl}} - T_{\text{ref}})}
\]  (3.19)

3.3 Velocity Measurements and Flow Visualisation

In order to understand the contribution of convection to the overall heat transfer regime and how the change in enclosure configuration can affect this, it was necessary to both qualitatively and quantitatively analyse the flow structures set up inside the leading edge structure. Fig. 3.15 shows the experimental setup for flowfield measurements within the leading edge. A double glazed window is placed at the front of the enclosure to view the illuminated flowfield. An opening was placed on the vertical wall of the leading edge where a glass pane [30mm (W) x 10mm (D) x 570 (H)] was inserted to allow for the laser lightsheet to enter into the enclosure. Particle Image Velocimetry was then used to visualise the flowfield, obtain the local velocity data for the convective regime and to study the effect of altering the geometric configuration on the flow structure.
3.3.1 Particle Image Velocimetry (PIV)

PIV is a non-invasive method of capturing and analysing an instantaneous region of interest within a flowfield. This instantaneous data can be combined over a test period to produce a time-averaged description of the flow. A detailed description of the fundamentals of this measurement technique can be found in Raffel et al. [118]. In order to perform PIV, the flow under investigation must first be seeded with tracer particles in order for it to be illuminated by a laser and captured by an acquisition camera. A requirement of the seeding method used is to be small enough to accurately follow the flow it is representing without any lag, yet large enough to scatter sufficient light to be detectable by the CCD (Charge-Coupled-Device) sensor of the camera. The particles are illuminated by a laser lightsheet formed when a coherent light beam is passed through a cylindrical lens. The lightsheet pulse is synchronised with the camera shutter and ensures that illumination of the articles occurs only once per frame. This is necessary to ”freeze” the flowfield in the image and remove any streaking in the image (due to incorrect timing / exposure settings) which will hinder accurate measurements. Measurement occurs at the focal point of the light sheet, where the beam is at its narrowest and 2-D analysis of the flowfield is possible. Care must be taken so that the out-of-plane movement of the particles is at a minimum to avoid lost particle pairs between images. The movement of the suspended particles is measured over a specific time interval between two images. This produces a 2-D displacement map of the region, divided
into individual interrogation regions, where a displacement value (\(\Delta x\)) is known in each region. A velocity vector for each of the interrogation regions is obtained by calculating a \(\Delta x/\Delta t\) value for region. This time interval (\(\Delta t\)) between image pairs is a determinate of the flow under investigation and the laser/optical setup. It is chosen such that the maximum displacement of a particle is no greater than 0.25 of the interrogation region length and that displacements of less than two particle distances were avoided. Due to the difference in velocity between the bleed duct plume and the stagnant surroundings, a \(\Delta t\) of 500\(ms\) was used in the near plume region, with \(\Delta t\) of 1000\(ms\) in the core region. The final flowfield is produced by combining the value for each of the individual interrogation regions together into one velocity map. The number of image pairs recorded must be sufficient to describe the time-averaged flow conditions with enough temporal resolution. To check for this, the region above the cylinder adjacent to the rear wall was chosen where the fast moving bleed duct plume, the opposing recirculation down the vertical wall and a stagnation region was present all in the same region of interest. Fig. 3.16 presents the percentage change in the mean velocity recorded across this image for the number of image pairs recorded. This assumed that 450 image pairs was sufficient to describe the flow accurately. It can be seen that once 100 – 150 image pairs are recorded then a no greater than 0.5% change is noted going to 450 image pairs and as a result 150 image pairs was deemed to be sufficiently accurate for the purpose of this analysis. This provided the best trade-off between accuracy of the resultant vector map and the acquisition time, storage requirements and processing time of the number of image pairs.

### 3.3.2 Experimental Apparatus

The measurements were made using the apparatus as described below:

- **Seeding particles:** The flow was seeded using smoke obtained from burning incense sticks. This method was chosen as it was readily available and the particle size produced when burnt (typically between 0.1 - 0.5\(\mu\)m) are deemed to be of sufficient size to follow the flow correctly, whilst scattering enough light back to the sensor to be accurately detected [119, 120].

59
Figure 3.16: Change in the mean velocity recorded for the number of image pairs acquired.

- Laser system: The laser used was a Litron Nano LPIV double cavity laser manufactured by TSI Inc. This was powered by a dual power supplies (Model no. LPU1000). The system consists of two independent laser units which are fired separately to enable minimal delay between pulses. The laser is a Nd-YAG laser of monochromatic light (532nm wavelength), which emits 50mJ of energy per pulse. The laser has a maximum operating frequency of 90Hz. A series of lens attached to the laser head produces a 2-D light sheet which narrows to a minimum of 1.5mm at the focal length of the lenses. This was arranged to be at a distance of 500mm from the lens. Measurements are taken at this point and combined with the camera lens operating at the narrowest depth of field, ensure a 2-D measurement plane.

- CCD Camera: A TSI 2 Megapixel Powerview 2M Plus CCD camera was used. This was interfaced to a processing PC via a 64-bit frame grabber PCI-Express card. The TSI 2MP Plus CCD camera has a maximum image acquisition rate of 30Hz at a resolution of 1600 x 1200 pixels. An AF Micro Nikkor 50mm f/1.2-16D lens was attached to the camera which operated at an aperture of f/1.2. The image pairs are acquired in a frame-straddling mode, where the laser fires at the end of the first exposure and at the beginning of the second exposure. This allows for the shortest time possible between pulses and can limit any blurring of the particles caused by
Figure 3.17: Overview of the PIV processing workflow. (a) Raw PIV image taken above the partition. Note the shadow present due to the chamfer in the partition. (b) Image after substitution of the average background light intensity. (c) Calculated vector map. (d) Vector map after smoothing and conditioning to fill holes. Here the gap in data due to the presence of the partition shadow has been successfully resolved.

movement from frame to frame which aids proper peak detection in the processing algorithm.

- Hardware Synchroniser: A TSI Synchroniser (Model no. 610035) was used to control the timing parameters of the laser and camera shutter to ensure the required pulse separation and lightsheet intensity. This can be either set manually or via the Insight 3G software supplied by TSI.
3.3.3 Experimental Procedure

During testing, the following procedure was adopted:

1. The test equipment was setup as outlined in Section 3.1 and the PIV system was setup as in Fig. 3.15. The laser was orientated towards the region to be measured, the camera was aligned to be perpendicular to the light sheet and was focused on it.

2. The prescribed surfaces within the leading edge were heated and the system was left until steady state conditions were obtained.

3. At this point, seeding was introduced into the enclosure. Depending upon whether the leading edge was vented or not, different methods to seed the flow were employed. Care must also be taken that any inherent buoyancy in the smoke dissipates and it reaches thermal equilibrium with the flow it’s measuring before image acquisition commences.

   (a) For the sealed enclosure tests, the incense was introduced via an opening in the back wall of the enclosure. The incense is burnt for a time (typically 1 – 2 minutes) to allow sufficient seeding to occur before being removed and the enclosure sealed again. The particles are then left to attain thermal equilibrium for (10 minutes approx.) with the flow before testing commenced.

   (b) For the ventilated tests, a number of sticks were introduced into the bottom of the outer enclosure and were burnt until sufficient seeding had entered both the outer and inner enclosure. Again, this was left to achieve thermal equilibrium with the airflow before testing, typically 30 minutes.

4. 150 image pairs were recorded for the given region of interest.

5. The camera (and laser lightsheet if necessary) were traversed horizontally and vertically to obtain a larger field of view of the flow structures in the leading edge.

6. The image pairs obtained were processed in the Insight 3G software package to produce the corresponding velocity vector maps of the flow field.
7. The steps above were repeated for the various enclosure configurations as required.

3.3.4 PIV Processing

Processing of the raw image data to obtain the resultant vector field maps is carried out using the commercial software Insight 3G by TSI. The acquired images are input into the software and where necessary, image masking is used to remove any areas of the image which do not need to be processed. The image pairs are then subdivided into interrogation regions which typically measure 32 x 32 pixels, with a 50% overlap between regions, as per the Nyquist grid engine. A cross correlation is performed on each section of the grid using a Fast Fourier Transform (FFT) to produce a correlation map. The distance between the centre of the correlation map and the peak represents the displacement of the interrogation region between subsequent frames. This is performed by a Gaussian peak engine, whereby only those points with a peak threshold greater than 1.5 are taken to be valid vector regions. Since the time between laser pulses is known from testing, the average velocity of the fluid in that interrogation region can be calculated. This is repeated across the entire image to produce the resultant flow-field.

3.3.5 Postprocessing

Postprocessing was performed on the resultant vector field maps produced. The purpose of this step was to both validate the vectors by comparing to the local median of the neighbouring vectors in a 5x5 surrounding region and to condition the vector field by filling holes due to the lack of a valid vector present in a given interrogation region. This is performed by interpolation from the local mean of the surrounding vectors (3x3) to recursively fill in the gaps in the data field. An example of this is presented in Fig. 3.17 (c) & (d). In order to produce an image of the global flowfield within the leading edge, individual 120mm x 120mm region vector maps were assembled in Matlab™ with an approximate 25% overlap to produce the final 550mm (H) x 300mm (W) vectorfield.
3.4 Experimental Uncertainty

An uncertainty analysis is performed based upon the methods outlined by Moffat [121] where the uncertainty in a calculated result as a function of several independent measured variables is given as

$$\delta R = \left( \left( \frac{\delta R}{\delta x_1} \delta x_1 \right)^2 + \left( \frac{\delta R}{\delta x_2} \delta x_2 \right)^2 + \left( \frac{\delta R}{\delta x_3} \delta x_3 \right)^2 + \cdots + \left( \frac{\delta R}{\delta x_N} \delta x_N \right)^2 \right)$$  \hspace{1cm} (3.20)

A list of measurement uncertainties and their resultant uncertainty upon the calculated values are presented in Table 3.3.

Table 3.3: Measurement uncertainties and calculated uncertainties

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Attributed Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length</td>
<td>± 2mm</td>
</tr>
<tr>
<td>Diameter</td>
<td>± 0.02mm</td>
</tr>
<tr>
<td>Voltage</td>
<td>± 0.2V</td>
</tr>
<tr>
<td>Current</td>
<td>± 0.02A</td>
</tr>
<tr>
<td>Emissivity</td>
<td>± 0.2</td>
</tr>
<tr>
<td>Temperature</td>
<td>± 0.2K</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Calculated Value</th>
<th>Maximum Uncertainty at $Gr = 1.3 \times 10^5$</th>
<th>Maximum Uncertainty at $Gr = 3.9 \times 10^5$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$q''_{total}$</td>
<td>2.2%</td>
<td>2.1%</td>
</tr>
<tr>
<td>$q'_{rad}$</td>
<td>2.4%</td>
<td>1%</td>
</tr>
<tr>
<td>$Nu_{rad,conv}$</td>
<td>6.7%</td>
<td>5.8%</td>
</tr>
<tr>
<td>$Nu_c$</td>
<td>11.6%</td>
<td>8.7%</td>
</tr>
</tbody>
</table>

3.5 Summary

In this chapter, the experimental test rig for the thermal analysis of an aircraft leading edge was presented. The enclosure and ventilation configurations were presented along with the placement and positions of geometrical objects within. The methods for analysing the heat transfer properties were outlined. The equipment and experimental setup was described along with an uncertainty measurement of the input and calculated variables.
Chapter 4

Sealed Leading Edge Enclosure

4.1 Introduction

This chapter presents the analysis of the non-ventilated leading edge enclosure as described in Section 3.1. Flow characteristics and heat transfer measurements from the bleed duct are presented for the leading edge subjected to the boundary conditions presented in Section 3.1. The heat transfer measurements were performed using the setup outlined in Section 3.2 and the flowfield was obtained using the PIV system detailed in Section 3.3.

Heat generating objects exhibit different heat transfer characteristics when confined in an enclosure. The presence of a bounding horizontal and/or vertical walls can alter the conditions experienced by the heated object compared to those present in a free stream or infinite medium [87]. This can be due to the presence of recirculation zones, fluid confinement, inhibited or augmented entrainment etc. If the surrounding walls are heated this adds a further level of complexity, as thermal boundary layers on the interior surfaces interact with the core bulk region of the enclosure, heating the air and further altering the conditions an object experiences compared to an unheated enclosure. In an aircraft, the interaction between the bleed duct and the heated leading edge results in a highly complex thermal environment. It is necessary to be able to understand this interaction in order to ensure the safe operation of thermally sensitive components within the leading edge. The objective of this chapter is to characterise the bleed duct heat transfer within the leading edge when operating in a hot day environment. The interaction between the bleed duct
and larger bundle of wires passing through the leading edge is also considered for different bleed duct locations.

4.2 Bleed Duct within Sealed Leading Edge (Case 1)

In the following section, the heat transfer from the bleed duct and the environmental conditions in the leading edge will be presented in terms of the temperature distribution within the enclosure, the influence of the flow structure surrounding the bleed duct and the resultant Nusselt number from the cylinder to the enclosure.

The temperature rise for the bleed duct, rear wall and interior air with $q_{BD}''$ is presented in Fig. 4.1. It is noted that the large rise in the bleed duct surface temperature is not reflected in the change in the surrounding air or rear wall temperatures. The interior air and rear wall surface temperatures increase with $q_{BD}''$, but not to the same extent as the bleed duct surface temperature. The average air temperature is initially greater than the rear wall, but the increase in the rear wall temperature with $q_{BD}''$ is such that the air temperature drops below that of the wall at the largest $q_{BD}''$ input. This would indicate a lack of effective convective cooling of the bleed duct, with the large rise in the bleed duct surface temperature not being reflected in the change in the surrounding air temperatures. It appears that the rear wall surface increases in temperature at a rate which is more in proportion with the increase in the bleed duct temperature.

4.2.1 Interior Temperature Profiles

The following section presents the change in the interior air distribution along three horizontal and vertical planes for a changing $Gr_{BD}$. These are presented in Fig. 4.2. Examining the horizontal temperature profiles in Fig. 4.2 (a, c, e & g), little change in $T^*$ is observed across the width of the enclosure at each of the $Gr_{BD}$ values presented. This suggests that the air within the centre region of the enclosure exhibits some degree of stratification, with the entrainment flow initiated by the bleed duct travelling horizontally throughout the enclosure. A slight increase in $T^*$ towards the rear of the enclosure, which is most evident along the lowest measurement location ($Y^* = 0.25$) can also be seen and is present for all
Gr\textsubscript{BD} values. The proximity of the bleed duct to the measurement location contributes to this local rise in temperature.

At Gr\textsubscript{BD} = 1.4 \times 10^5, the lower region of the enclosure (along the Y* = 0.25 plane in Fig. 4.2 [a]) is at a markedly lower T* value of 0.4 compared to Y* = 0.5 & Y* = 0.75 planes (0.55 & 0.6 respectively). An increase in T* is present as Y* increases from 0.5 to 0.75, however it is not as pronounced as seen in the lower region. This indicates that the mid and upper region of the enclosure consists of a relatively isothermal region with a reduction in T* below Y* = 0.4. This phenomenon reduces with an increase in Gr\textsubscript{BD}. At Gr = 3.9 \times 10^5, the increase in T* with enclosure height remains relatively constant between the three horizontal measurement planes in Fig. 4.2 (g). This can be attributed to the increased influence of the bleed duct plume and its mixing effect at the higher Gr\textsubscript{BD} values investigated.

The change in thermal gradient within the enclosure is also evident in the vertical temperature profiles in Fig. 4.2 (b, d, f & h). At the lowest Gr\textsubscript{BD} value investigated (1.4 \times 10^5), the enclosure is split into two distinct regions, each with different thermal gradients present. The separation points between each of these zones is shown by the green dashed line in each
Figure 4.2: Case 1 horizontal and vertical temperature profiles ($X^*, Y^* = 0.25(\Box), 0.5 (\diamond) & 0.75 (\triangle)$) within the enclosure for $Gr_{BD} = 1.4 \times 10^5$ (a,b), $2.5 \times 10^5$ (c,d), $3.3 \times 10^5$ (e,f) & $3.9 \times 10^5$ (g,h). The rear wall temperatures (○) are superimposed onto the vertical temperature profiles for comparison.
of the vertical temperature plots in Fig. 4.2. In the lower half of the enclosure ($Y^* < 0.5$), $T^*$ increases from 0.22 to 0.55, whereas the change in $T^*$ with height is not as significant in the upper portion of the compartment ($Y^* < 0.5$). $T^*$ only increases from 0.55 to 0.67 in this region. This indicates that the majority of the high temperature air is being confined to the upper region of the enclosure, creating a more isothermal zone here. Entrainment by the cylinder and the proximity of the cold surface helps establish this gradient in the lower region of the enclosure.

As $Gr_{BD}$ increases, this separation of the enclosure into two regions (the upper more isothermal region dominated by plume mixing effects and the lower region with a larger thermal gradient due to bleed duct entrainment) becomes less evident. At $Gr_{BD} = 3.9 \times 10^5$ in 4.2 (h), an almost constant increase in $T^*$ is seen throughout the enclosure height. This indicates that as the input power to the bleed duct increases, the cylinder and thermal plume effects begin to dominate the conditions within the enclosure. As a result, the enclosure bulk air becomes more isothermal as the high temperature air from the bleed duct is more evenly distributed throughout the enclosure. This leads to the lower increase in $T^*$ with height seen in Fig. 4.2 (h).

The temperatures recorded on the surface of the rear vertical wall are presented in Fig.
4.2 (b, d, f & h) and separately in Fig. 4.3. It is evident from these plots that $T^*$ remains approximately constant across the $Gr_{BD}$ range. The vertical wall is isothermal with $T^* = 0.28 - 0.3$ for $1.4 \times 10^5 < Gr_{BD} < 3.9 \times 10^5$. The constant $T^*$ on the rear vertical wall for increasing $Gr_{BD}$ and the drop in the interior air $T^*$ indicates that the rear wall temperatures increase in proportion to the bleed duct. It also suggests a change in the convective conditions experienced by the wall surface. The intersection point for the vertical wall and air temperature profiles in Fig. 4.2 represents a stagnation point between two opposing flows on the wall surface where both have the same $T^*$ value. At $Gr_{BD} = 1.4 \times 10^5$, this intersection is at a low point on the wall surface ($Y^* = 0.15$), indicating that the wall surface is dominated by the presence of downward, negatively buoyant flow. As the air $T^*$ drops and the wall value stays constant at 0.3, this intersection point rises to $Y^* = 0.2, 0.35 & 0.55$ as seen in Fig. 4.2. As a result the positively buoyant flow increases its influence on the lower region of the wall as $Gr_{BD}$ increases. The data presented here is useful as an approximation to this change in flow structure as it cannot account for the air temperature directly over the wall surface but is based on the vertical measurement plane at $X^* = 0.25$ which is expected to have a different temperature. However, this relationship will be shown to be reasonably accurate without the knowledge of the air temperature at the wall surface in the following section.
Figure 4.5: Measured flow velocity fields near the Bleed Duct at $Gr_{BD} = (a) \ 1.4 \times 10^5, \ (b) \ 2.5 \times 10^5$
Figure 4.6: Measured flow velocity fields near the Bleed Duct at $Gr_{BD} = (a) 3.3 \times 10^5$ & (b) $3.9 \times 10^5$
4.2.2 Flow Adjacent to Bleed Duct

Local air velocity measurements were obtained for the airflow adjacent to the bleed duct to understand the effect of confining the bleed duct to the lower corner of the leading edge. These are presented in Fig. 4.5 & 4.6. The general outline of the flowpath at the compartment level is presented in Fig. 4.4 as reference to these detailed measurements. The plume from the bleed duct establishes an anticlockwise recirculation adjacent to the upper section of the rear wall which becomes entrained by the bleed duct again after it travels down the surface of the vertical wall. Its influence is largely confined to the rear section of the enclosure near the vertical wall. The oscillations of the plume also help disperse the high temperature plume flow to the rest of the enclosure. A second anticlockwise recirculation is also encountered along the interior wall of the leading edge, driven by the curvature of the heated surface. The remainder of the compartment is largely stagnant, with slow moving flow in the region of 0.1 m/s, depending upon $Gr_{BD}$.

At $Gr_{BD} = 1.4 \times 10^5$ (shown in Fig. 4.5 [a]), the plume is observed to orient towards the centre of the compartment at an angle of approximately 15° to the vertical. An imbalance in the airflow over the two sides of the bleed duct occurs due to its confinement near the enclosure wall. Entrainment by the bleed duct in the region adjacent to the vertical wall occurs from two sources. The first is the plume recirculation travelling down the rear wall, where it feeds back into the top of the bleed duct and the second is from the airflow over the lower enclosure surface. At $Gr_{BD} = 1.4 \times 10^5$, the bleed duct plume is influenced by both of these contributions. The acceleration of the airflow between the bleed duct and the vertical wall (due to the energising of the airflow in the bleed duct boundary layer and the reduction in the gap between the bleed duct and wall) along with the addition of the upper recirculation feeding into the plume cause this airflow imbalance and leaning of the plume. There is upward flow evident on the lower section of the rear wall in Fig 4.5 (a).

As $Gr_{BD}$ increases, the change in the temperature of the rear wall relative to the air temperature begins to influence the flow structure adjacent to the bleed duct. The drop in the air $T^*$ relative to the wall establishes a secondary upward flow on the lower section of the rear wall. This is evident from the velocity profiles taken between the bleed duct and the rear wall (at the bleed duct mid-height) presented in Fig. 4.7. A clearly identifiably
boundary layer profile is evident at $Gr_{BD} = 2.5 \times 10^5$, which increases in magnitude with $Gr_{BD}$. A separation point between the upward and downward flows is also evident at $Y^* = 0.2$ in Fig. 4.5 (b). The upward flow in the lower section then detaches from the wall and is entrained by the bleed duct. The plume continues to lean towards the centre of the enclosure in this case. The maximum velocity recorded in the boundary layer on the wall (Fig. 4.7) increases from 20% of the maximum velocity recorded in the bleed duct boundary layer at $Gr_{BD} = 2.5 \times 10^5$, to 35% at $Gr_{BD} = 3.3 \times 10^5$ and 43% at $Gr_{BD} = 3.9 \times 10^5$. As this flow strengthens, it pushes the stagnation point between the opposing flows higher up the wall surface, increasing to $Y^* = 0.27$ and $Y^* = 0.35$ for $Gr_{BD} = 3.3 \times 10^5$ and $Gr_{BD} = 3.9 \times 10^5$ respectively. When this occurs, the portion of the air travelling below the bleed entrained by this secondary flow on the vertical wall bypasses the bleed duct surface altogether. This in turn causes the airflow imbalance on either side of the bleed duct to decrease, resulting in a plume that oscillated less and travelled almost vertically into the enclosure.

Although a change in flow structure over the wall surface in the region adjacent to the bleed duct is seen in Fig. 4.5 & 4.6 with $Gr_{BD}$, there is no significant change in the temperature distribution along its height as seen in Fig. 4.2 (a, c, e & g). It remains approximately isothermal despite this change. This would suggest that the rear wall temperature distribution is insensitive to a change in the convective regime on its surface, instead being influenced primarily by radiative exchange between the bleed duct and the wall, which maintains this approximately isothermal condition. The influence of this change in flow structure and the previously noted change in temperature distribution on the bleed duct heat transfer will be investigated in the following section.

### 4.2.3 Bleed Duct Heat Transfer

This section outlines the resultant heat transfer from the bleed duct in the heated leading edge for the prescribed environmental conditions. The influence of choice of a suitable convective reference temperature is also discussed for suitability and accuracy in the analysis. The analysis is performed as outlined in Section 3.2.3. The objective to characterise the heat transfer from the bleed duct as a result of the internal temperature distribution and flowfield adjacent to the bleed duct presented in Section 4.2.1 & 4.2.2.
4.2.3.1 Choice of Convective Reference Temperature

When performing the analysis of heat transfer from an isolated heat source in an enclosure, one consideration that must be made is the choice of a suitable convective reference temperature. For free convection, this can easily be taken to be the ambient or “freestream” conditions which are easily determinable in an experimental or numerical investigation. Simple enclosure natural convection also typically uses an easily definable temperature, typically based on a wall condition such as the cold wall \( T_C \). For an internal heat source within an enclosure, this is complicated by the fact that there is neither pure free convection nor simple enclosure convection present, but a mixture of both. The core bulk fluid within is surrounded by the heated enclosure surfaces which have their own thermal boundary layers and influence on the interior outside of the convection from the heat source alone. This alters the conditions in the leading edge beyond that of a typical enclosure configuration. The non-standard enclosure shape complicates this further.

The objective of this is to investigate the sensitivity of the Nusselt number calculations to the choice of the convective reference temperature and to determine a suitable choice for the analysis. Several surface and air based reference temperatures are investigated; the average enclosure wall temperature \( (T_h + T_c)/2 \), the adiabatic bleed duct temperature, the
total average air temperature throughout the enclosure, the average temperature along the horizontal and vertical enclosure centreline and the temperature taken local to the bleed duct only (as shown in Fig. 3.10).

Fig. 4.8 shows the variation in the temperature difference between the bleed duct surface and the various reference temperatures chosen. At low $q''_{BD}$, this temperature difference is quite similar for all $T_{ref}$ values, with the surface based measurements slightly higher than the air based ones. As $q''_{BD}$ increases, $\Delta T$ continues to increase and becomes much larger for the surface based temperatures than air based ones. Here, the interior air temperature of the enclosure increases with $q''_{BD}$ which reduces the $\Delta T$ compared to that of the surface based measurements which are independent of the power dissipated by the bleed duct and remain constant. The $\Delta T$ values for the three air based measurements begin to converge on each other as the average air within the enclosure begins to homogenise as shown in Fig. 4.2.

The resultant convective heat transfer ($Nu_{BD}$) is presented in Fig. 4.9. Here an appreciable difference in $Nu_{BD}$ is noted depending upon the choice of $T_{ref}$. Due to the lower $\Delta T$ produced when using an air based $T_{ref}$, the resultant $Nu_{BD}$ is higher compared to the
Figure 4.9: Variation in bleed duct convective Nusselt number. \( T_{ref} = \) average enclosure temperature (○), enclosure centrelines (□), temperatures local to bleed duct (△), \( (T_h + T_c)/2 \) (○) and adiabatic bleed duct temperature (×).

A variation in \( Nu_{BD} \) is noted for the air based \( T_{ref} \), particularly at the lower \( Gr_{BD} \) range. This can be explained by the thermal gradients and thermal confinement present in the enclosure as noted in the previous section (Fig. 4.2). When \( T_{ref} \) is based upon the entire enclosure thermal distribution, the heating effect of the upper enclosure surface on the air and its confinement to the upper region of the enclosure will produce a \( T_{ref} \) value that is larger than that of the air surrounding the bleed duct. This produces a lower perceived \( \Delta T \) for the bleed duct, which in turn produces a higher \( Nu_{BD} \) value. Due to the bleed ducts location in the lower region of the enclosure adjacent to the rear wall and the large drop in air temperature close to the cold enclosure wall (Fig. 4.2 [a]), the bleed duct will entrain air which is cooler that the average enclosure and is closer to \( T_C \). This means that the \( \Delta T \) will be higher in this case and \( Nu_{BD} \) lower. This gives a more realistic description of the
regime as it is with this air that the bleed duct convects heat to, not the confined hot air in the upper region of the enclosure.

When the temperature of the enclosure becomes more uniform as $Gr_{BD}$ increases, this effect diminishes and the $Nu_{BD}$ values for the three air based temperatures converge towards a single value. The $Nu_{BD}$ becomes insensitive to the choice of air based reference temperatures above $Gr_{BD} = 2.5 \times 10^5$. Below this point, taking the temperatures local to the cylinder will produce a more accurate representation of the heat transfer regime local to the cylinder. Care must be taken if this method is to be used as previous authors such as Liu et al. [122] have shown how sensitive this parameter is to the exact position of the thermocouples near the surface, especially in the presence of large thermal gradients. Also this requires a detailed knowledge of the thermal distribution to be known beforehand in order to determine which measurement locations to choose, something which may not be available in all investigations.

In summary, the choice of a suitable reference temperature is appropriate to ensure an adequate representation of the bleed duct heat transfer. It can often be determined by the level of detail of the thermal field known, where only limited air temperature measurements or surface measurements may be known. In the leading edge, a maximum deviation of approximately 21% is seen between the reported values of $Nu_{BD}$ between $T_{ref}$ values. In the current configuration, the use of a specific air based reference point (local or global within the enclosure) becomes less important as the internal temperature field becomes more uniform with increasing $Gr_{BD}$. Below the threshold value of $Gr_{BD} = 2.5 \times 10^5$, it may make more sense to use the temperatures local to the cylinder if this data is available, but the far-field data can be used with confidence if its limitation is acknowledged. The surface based $T_{ref}$ values underpredict the cylinder heat transfer due to the larger $\Delta T$ in this case, which increases the perceived temperature gradient on the cylinder surface, leading to a lower $Nu_{BD}$. For the analysis presented here and in the following sections, the reference temperature was chosen to be the entire enclosure average temperature due to its accurate representation of the overall enclosure conditions, the ease of obtaining this data in future investigations and comparability with the freestream conditions typically used in the analysis of convection heat transfer.
4.2.3.2 Bleed Duct Nusselt Number

The convective and radiative heat fluxes from the cylinder, calculated per Eq. 3.11, 3.17 & 3.18 are presented in Fig. 4.10. Overall the convective heat transfer from the cylinder is seen to be relatively small, accounting for between 21-28% of the overall heat removed. The convective contribution also decreases as $Gr_{BD}$ increases. The high surface emissivity and the large temperature differences in the compartment between surfaces accounts for this, suggesting a negative influence on the convective regime. Radiation heat transfer from the cylinder to the enclosure dominates over the convective losses.

Bleed duct positioning also plays a role here as for a unconfined cylinder or a cylinder centrally located in an enclosure, it is able to entrain cooler surrounding air easily from its surroundings. For the case presented here, the confinement effect means that the cylinder can only entrain air from its right hand side and what travels through the gap between the lower enclosure surface and the wall, as shown in Fig. 4.6. The increased influence of the secondary flow structure on the lower section of the wall as $Gr_{BD}$ increases also affects heat transfer as it entrains air over its surface which was previously available to the cylinder for convective cooling. This is also seen to straighten and stabilise the cylinder plume as previously noted, but also reduces the convective heat transfer.

Comparing the results to free cylinder (Churchill & Chu [3]) and enclosed cylindrical
objects in a circular enclosure (Sparrow et al. [10]) correlations in Fig. 4.11 show that there are discrepancies between them and the results obtained here. Initially the bleed duct behaves similarly to that of an unconfined free cylinder, with no apparent influence of the enclosure confinement or eccentric positioning. As $Gr_{BD}$ increases, $Nu_{BD}$ does not increase in accordance with the correlation, but remains at $Nu_{BD} = 9$, with a slight decrease visible. Here the classical dependence on $Nu$ with $Gr^{0.25}$ is not evident from the results presented. As $Gr_{BD}$ increases further, the heat transfer from the bleed duct behaves similar to that of a confined cylinder in an enclosure as predicted by Sparrow et al. [10] and remains within this range at the upper $Gr_{BD}$ range. This suggests that the influence of the enclosure and positioning on the cylinder heat transfer is more prevalent in this region. A crossover point exists between the bleed duct behaving as a cylinder in free convection and behaving as a confined cylinder within an enclosure at approximately $Gr_{BD} = 2.5 \times 10^5$. Fig. 4.11 also compares the current data to a range of correlations available for free cylinders over the testing range. Similar conclusions can be drawn from these, with agreement within the limit of experimental uncertainty at low $Gr_{BD}$, but a drop below the predicted range as $Gr_{BD}$ approaches $3.9 \times 10^5$ due to the enclosure effect.
4.3 Multiple Cylinder Interactions (Case 2 & 3)

In this section, the interaction of the bleed duct with a second larger cylinder at constant temperature is presented. In an aircraft leading edge the bleed duct is also co-located with a bundle of electrical wires which passes through the structure. These are of a larger diameter than the bleed duct and heat up due to ohmic dissipation. This is simulated by a Ø75mm hollow aluminium cylinder with a heating mat attached to its outer surface. In order to investigate the effect of this cylinder on the heat transfer from the bleed duct, experiments were performed for the bleed duct located directly below and above the cylinder as detailed in Section 3.1.2. These represent Case 2 and Case 3 presented in this section.

Multiple cylinder interactions can be quite complex. It has been observed that for a cylinder located at the bottom of an isothermal array, its heat transfer performance is not affected by the presence of heated objects above and it behaves similarly to a single cylinder under identical conditions [83–85]. The upper cylinders however, are influenced by parameters such as the vertical displacement between cylinders to their diameter ($s/D$ as per Fig. 3.6), the position within the array and also the cylinder Rayleigh number. Spacing plays an important role as it can either improve or impede the heat transfer from the cylinder, depending upon the distance between them in the array. The purpose of this section is to investigate the multiple cylinder interaction within the confining leading edge for a range of cylinder input powers and determine its effect on the resultant heat transfer from the cylinders.

4.3.1 Interior Temperature Distribution

This section presents the temperature distribution in the enclosure for where the bleed duct was placed below (Case 2) and above (Case 3) the larger constant surface temperature cylinder ($C_2$). Experiments allowed for the conditions where the bleed duct was both cooler and hotter than $C_2$ for both Case 2 & 3. This allowed for an investigation into the influence of a temperature imbalance between the cylinders to be performed. This has been shown by Sparrow and Niethammer [16] to have a large influence on the resulting cylinder heat transfer, and is predominately a negative influence at smaller cylinder separation distances.
(s/D < 5). This will be shown to be the case in the present setup of s/D = 2. The temperature difference between the bleed duct and $C_2$ ($T_{BD} - T_{C_2}$) is shown in Fig. 4.12. Here the temperature imbalance ranges from $-32$ to $+64$ for Case 2 and from $-18$ to $+40$ for Case 3.

The temperature profiles for Case 2 at $Gr_{BD} = 1.5 \times 10^5$, $2.5 \times 10^5$, $3.2 \times 10^5$ & $3.7 \times 10^5$ are presented in Fig. 4.13. Similar to those presented for Case 1 in Fig. 4.2, the temperatures across the enclosure stay relatively constant with the main thermal gradient present through the height of the enclosure in Fig. 4.13 (a, c, e & g). A slight increase in $T^*$ is noted towards the rear of the enclosure at $X^* = 0.25$ in Fig. 4.13 (a) & (c). Here heating from the bleed duct and cylinder increases $T^*$ locally. The increase in $T^*$ with height is relatively constant as shown in Fig. 4.13 (b, d, f & h). This suggests enhanced mixing of the interior air due to the additional cylinder compared to Case 1 in Fig. 4.2, where a noticeable change in $T^*$ with enclosure height was observed, particularly in the low section of the enclosure at low $Gr_{BD}$. Similar to Case 1, a drop in air $T^*$ is noted relative to the vertical wall in Fig. 4.13 (b, d, f & h). At $Gr_{BD} = 2.5 \times 10^5$, the isothermal point between the wall and air is at mid-enclosure height. As $Gr_{BD}$ increases, the drop in air $T^*$ is enough to produce the scenario at $Gr_{BD} = 3.7 \times 10^5$ where almost 90% of the air (based on height)
Figure 4.13: Case 2 horizontal and vertical temperature profiles ($X^*, Y^* = 0.25(□), 0.5 (∗) & 0.75 (△)$) within the enclosure for $Gr_{BD} = 1.5 \times 10^5$ (a,b), $2.5 \times 10^5$ (c,d), $3.2 \times 10^5$ (e,f) & $3.7 \times 10^5$ (g,h). The rear wall temperatures (∗) are superimposed onto the vertical temperature profiles for comparison.
is below the wall temperature. This will impart a significant positive buoyancy force on the air over the wall surface, which has the possibility of enhancing convective heat transfer on the rear wall.

A very different environment is present within the enclosure for Case 3 as presented in Fig. 4.14. Due to the location of the bleed duct and $C_2$ and the confinement of the air to the upper region, thermal gradients are present both along the height and through the width of the enclosure. $T^*$ rises sharply towards the rear of the enclosure ($X^* = 0$) along the horizontal profiles in Fig. 4.14 (a, c, e & g). A slight deviation is noted on the lowest measurement plane ($Y^* = 0.25$) near the front of the enclosure, where $T^*$ levels out at $X^* > 0.6$. The vertical temperature profiles in Fig. 4.14 (b, d, f & h) show that $T^*$ increases linearly with height above $Y^* = 0.2$ throughout the entire $Gr_{BD}$ range and for all the three vertical temperature profiles. This suggests the higher temperature air is being confined to upper and rear section of the enclosure adjacent to the vertical wall. This is confirmed when observing the temperature of the air relative to the vertical wall. At $Gr_{BD} = 2.5 \times 10^5$, it is only the air closest to the wall ($X^* = 0.25$) which is at a higher temperature due to the temperature gradient across the width of the enclosure. The rest of the interior air is at a lower $T^*$ level. As $Gr_{BD}$ increases, the air $T^*$ drops sufficiently such that the air $T^*$ is lower than the wall at all locations. If just the air closest to the wall is considered, this is quite similar to the observations made for this air-wall interaction in Case 2.

The rear wall temperatures for Case 2 & 3 are presented in Fig. 4.15. Unlike Case 1 shown previously, the rear wall $T^*$ decreases with $Gr_{BD}$. This highlights that the wall surface temperature does not increase in proportion with the bleed duct as $Gr_{BD}$ increases. This can be attributed to the additional heating effect of $C_2$ on the vertical wall. At the low $Gr_{BD}$ range investigated, the rear wall is heated mainly by $C_2$, with little influence from the bleed duct. Only as $Gr_{BD}$ increases and $T_{BD} > T_{C_2}$ does the reduction in $T^*$ decrease. Here the bleed duct begins to influence the temperature of the enclosure and rear wall, increasing both. This results in a small change in $T^*$ between $Gr_{BD} = 3.2 \times 10^5$ and $Gr_{BD} = 3.7 \times 10^5$ compared to lower $Gr_{BD}$ values. For Case 2, $T^*$ is approximately constant for the wall height. For Case 3 an increase in $T^*$ is observed with wall height and can be attributed to thermal confinement of the hot air to the upper section of the leading edge.
Figure 4.14: Case 3 horizontal and vertical temperature profiles ($X^*, Y^* = 0.25(□), 0.5 (♦) & 0.75 (△)$) within the enclosure for $Gr_{BD} = 1.5 \times 10^5$ (a,b), $2.5 \times 10^5$ (c,d), $3.2 \times 10^5$ (e,f) & $3.7 \times 10^5$ (g,h). The rear wall temperatures (♦) are superimposed onto the vertical temperature profiles for comparison.
CHAPTER 4. SEALED LEADING EDGE ENCLOSURE

Figure 4.15: Case 2 (a) & Case 3 (b) rear wall temperature profiles for $Gr = 1.5 \times 10^5 (\circ)$, $2.5 \times 10^5 (\diamondsuit)$, $3.2 \times 10^5 (\triangle)$ & $3.7 \times 10^5 (\square)$

Figure 4.16: Average interior air $T^*$ for Case 1 (○), Case 2 (●) and Case 3 (□)
The $T^*$ values for the interior air for Cases 1 – 3 are presented in relation to the input power ($q''_{BD}$) in Fig. 4.16 (a) and to $Gr_{BD}$ in Fig. 4.16 (b). For Case 2, the presence of the second heated cylinder increases the enclosure air temperature greater than that of the bleed duct alone, leading to an increase in $T^*$. For Case 3 the effect of the bleed duct confinement (resulting in a lack of convective mixing) and heating from the second cylinder below increases the bleed duct surface temperature as opposed to the air within the leading edge. This decreases $T^*$ compared to the bleed duct only configuration in Fig. 4.16 (a). As $q''_{BD}$ increases and the bleed duct becomes the dominant influence on the enclosure, the heating effect of the air from $C_2$ diminishes such that the air $T^*$ is determined by the bleed duct and Case 1 & 2 begin to converge upon each other. For Case 3, the drop in $T^*$ is somewhat proportionate to that of Case 2, suggesting that the rise in interior air relative to the bleed duct is similar for both Case 2 & 3. This indicates that for higher bleed duct input powers, heat transfer from the bleed duct becomes more important upon the interior conditions than heating from $C_2$.

The rear wall temperature profiles for increasing $q''_{BD}$ and $Gr_{BD}$ are presented in Fig. 4.17 (a) & 4.17 (b). For the bleed duct only case, the rear wall $T^*$ remained constant at 0.3 across the testing range. This highlighted how the rear wall temperature increased in proportion to the bleed duct temperature, as discussed in Section 4.3.1. With the presence
CHAPTER 4. SEALED LEADING EDGE ENCLOSURE

Figure 4.18: Temperature rise between the bleed duct and interior air \((T_{BD} - T_{avg})\) for Case 1 (○), Case 2 (⋄) and Case 3 (□)

of the second cylinder, the wall \(T^*\) is larger in both cases due to its heating influence. As the bleed duct input power increases, \(T^*\) reduces, indicating that the rise in bleed duct temperature is greater than the rise in the rear wall temperature. This demonstrated that the rear wall temperature is initially dependant upon heat transfer from the larger cylinder (which has a greater surface temperature), with heat transfer from the bleed duct becoming prominent as its dissipation level increases. This reduces the decrease in \(T^*\), becoming more like the bleed duct only configuration (i.e. smaller change in \(T^*\)) as the input power increases.

Fig. 4.18 presents the temperature difference between the cylinder and the surrounding averaged interior air \((T_{BD} - T_{avg})\). Here it is evident that there is little difference between the \(\Delta T\) in Case 1 and Case 3, even though there is much lower power being input into the cylinder in Case 3. This highlights the influence that the confinement of the cylinder and the heating from \(C_2\) has upon the cylinder properties, raising its temperature significantly. The \(\Delta T\) for Case 2 is largest across the entire \(Gr_{BD}\) range.
4.3.2 Bleed Duct Heat Transfer

The bleed duct Nusselt numbers, due to the presence of the second cylinder are presented in Fig. 4.19. The values of $Nu_{BD}$ for Case 1 & Case 2 indicate that locating of the second cylinder above the bleed duct has little influence on the resultant Nusselt number, particularly in the region where they are at approximately the same temperature ($2 \times 10^5 < Gr_{BD} < 3 \times 10^5$). A slight enhancement in $Nu_{BD}$ is observed for Case 2 beyond this point, with a maximum increase in $Nu_{BD}$ of 14% at $Gr_{BD} = 3.5 \times 10^5$.

For Case 3, the addition of the second heated cylinder below the bleed duct has a detrimental effect on $Nu_{BD}$. It has been reported previously that in a multiple array, upper cylinder heat transfer can be as low as 50% of that of a single unconfined cylinder [84]. At the separation distance seen here ($s/D = 2$), Sparrow and Niethammer [16] report a drop in $Nu_{BD}$ of almost 20%. They also discovered that this drop reduced slightly for an increase in Rayleigh number of $2 \times 10^4 - 2 \times 10^5$. This negative effect is further influenced by the confining effect of the leading edge when the cylinder is located in the upper section close to the wall surface.

In Case 3, the bleed duct is preheated by the lower constant temperature cylinder. This predominately occurs below $Gr_{BD} = 2.4 \times 10^5$ where $C_2$ is much hotter than the bleed duct. This has the effect of reducing the temperature gradient on the bleed duct surface, to the detriment of convective heat transfer. At values of $T_{BD} - T_{C_2} < 0$, this is enough to practically eliminate any convective heat transfer from the bleed duct and it participates in radiative heat transfer only. When the bleed duct & $C_2$ are at the same temperature at $Gr_{BD} = 2.4 \times 10^5$, $Nu_{BD}$ for Case 3 is only 15% of that of Case 1, much lower than expected from literature. Here the added influence of enclosure confinement plays a role in this decrease in of $Nu_{BD}$. A further increase in bleed duct input power raises its surface temperature sufficiently to produce a thermal gradient on its surface to allow for convective heat transfer to occur. This is indicated in the increase in $Nu_{BD}$ above $Gr_{BD} = 2.5 \times 10^5$. 
4.3.3 Effect of Cylinder Temperature Variation

The influence of the temperature variation between vertically spaced cylinders has been shown to be influential on the resultant cylinder heat transfer, particularly at the separation distance studied here (s/D = 2). At this distance the temperature imbalance reduces the convective heat transfer from the bleed duct. An increase in separation distance has been shown also to enhance the heat transfer as the effect of cylinder preheating diminishes and the forced convection from the lower plume begins to dominate, increasing the convective heat transfer from the cylinder [16].

In order to show this influence of the temperature variation, Sparrow and Niethammer [16] define a temperature ratio (TR)

\[
TR = \frac{(T_{cyl} - T_{\infty})_{\text{lower}}}{(T_{cyl} - T_{\infty})_{\text{upper}}} \tag{4.1}
\]

where the subscript \( \infty \) indicates the ambient conditions. This is taken to be the average enclosure air temperature within the leading edge. This temperature ratio for Case 2 and Case 3 is presented in Fig. 4.20. Here it can be seen that as \( Gr_{BD} \) increases, this ratio
increases for Case 2 (due to the increase in the bleed duct temperature located above) and decreases for Case 3 (due to the increase in the bleed duct temperature located below).

The change in the normalised Nusselt number (with respect to the bleed duct only case [Case 1]) with the change in $TR$ is presented in Fig. 4.21 for Case 2 & 3. Also present is the data from Sparrow and Niethammer [16] for the same cylinder separation distance ($s/D = 2$). This data is for the upper cylinder in a two cylinder configuration only. At this distance, an increase in the temperature imbalance is found to be to the detriment of the cylinder heat transfer. At a temperature ratio below 0.5 a slight enhancement is noticed where $Nu/Nu_0 = 1.05$, suggesting there is some benefit to having the upper cylinder significantly hotter than the lower one. No explanation is offered for this, but it could possibly be due to the enhanced entrainment and convection from the lower cylinder aiding the heat transfer from the upper cylinder. For the equal cylinder temperature scenario, the upper cylinder heat transfer is 0.85 times that of the single cylinder, compared to the lower cylinder which is found to be identical to the single cylinder case. As the temperature imbalance continues to where the lower cylinder is much hotter than the upper one, the effect of cylinder pre-heating and the reduction in the thermal gradient on the cylinder surface produces a large drop in $Nu/Nu_0$. 
The experimental data for Case 3 (where the bleed duct is now the upper cylinder) shows a large reduction in \(\frac{Nu}{Nu_0}\). \(\frac{Nu}{Nu_0}\) is at a maximum at \(TR = 0.5\), which is significantly lower than the value of Sparrow and Niethammer [16] of \(\sim 1\). In the current study, the confinement of the high temperature air to the upper section of the enclosure decreases the temperature between the bleed duct and the local surrounding air, which produces the drop in \(\frac{Nu}{Nu_0}\) seen here. As the temperature ratio exceeds 1, \(\frac{Nu}{Nu_0}\) levels out and remains constant for any further increase in \(TR\).

For the bleed duct placed below the heated cylinder (Case 2), the change in \(\frac{Nu}{Nu_0}\) with \(TR\) is small compared to Case 3. In the region \(0.75 < TR < 2\), \(\frac{Nu}{Nu_0}\) remains relatively constant just below 1 (0.95 – 0.98). As \(TR\) increases above 2, some slight enhancement in \(\frac{Nu}{Nu_0}\) is evident as noted previously. Overall, increasing \(TR\) increases \(\frac{Nu}{Nu_0}\), but this enhancement is quite limited. This tends to agree with the observations made previously that in a multiple cylinder array, the lowest cylinder does not generally show significant deviation from a single cylinder scenario. This also suggests that there is no added enclosure effect on the multiple cylinder heat transfer compared to the single
CHAPTER 4. SEALED LEADING EDGE ENCLOSURE

Figure 4.22: Normalised convective Nusselt number ($\frac{Nu}{Nu_0}$) for Case 3 (□) compared to unconfined cylinder (solid line) from Sparrow and Niethammer [16].

cylinder case for this configuration.

The actual $\frac{Nu}{Nu_0}$ value compared to that for an unconfined cylinder at the same $TR$ value is shown in Fig. 4.22 for varying $Gr_{BD}$. This is based on the assumption that the change in $\frac{Nu}{Nu_0}$ with $TR$ is independent of $Gr$ which is shown by Sparrow and Niethammer [16] to hold at $s/D = 2$. $\frac{Nu}{Nu_0}$ for Case 3 is much lower than expected for the same $TR$ value. At the maximum $Gr_{BD}$ value of $3.5 \times 10^5$ and at a $TR$ value of 0.57, the expected $\frac{Nu}{Nu_0}$ is just over 1, whereas a value of 0.5 is seen for Case 3. The main influence of the confinement within the leading edge for a multiple cylinder configuration is to drastically reduce the overall heat transfer when compared to that of an unconfined one.

In order to highlight this, the effective $TR$ value the cylinder is operating at due to the confinement is presented in Fig. 4.23. This is based upon the $TR$ value which would be expected for a given $\frac{Nu}{Nu_0}$ for an unconfined cylinder. Here it can be seen that when placed above the constant temperature cylinder in the leading edge, the bleed duct behaves as if it was operating in an environment where the temperature ratio between the cylinders and their environment is higher than it actually is. In this case, the bleed duct has effectively seen its surface temperature reduced relative to the lower one. As a result the heat transfer
Figure 4.23: Effective cylinder temperature ratio ($\square$) for Case 3 compared to the measured value (solid line).

from the bleed duct will be reduced due to the reduced thermal gradients on its surface, which produces the results seen in Fig. 4.19. From Fig. 4.23 it can be seen that the bleed duct is behaving as though its temperature ratio is $> 1$ throughout the entire testing range (i.e. the lower cylinder is constantly hotter than the bleed duct). In reality, it changes between the lower cylinder and bleed duct being the hotter surface (as shown by the dashed line in Fig. 4.23 and the temperature difference in 4.12), but this is not reflected in the results presented here.

4.4 Conclusions

The enclosure conditions and cylinder heat transfer were presented in this section for the sealed leading edge enclosure for single and multiple cylinder configurations. In the single cylinder scenario, the interior air distribution changed significantly with a change in $Gr_{BD}$. Initially at low cylinder power inputs, a confined region of high temperature air is present in the upper section of the enclosure, with the thermal environment split into two distinct regions of different thermal gradient. As $Gr_{BD}$ increases and the cylinder begins to dominate the convective regime within the enclosure, mixing from the plume is enhanced and
the enclosure becomes more isothermal. The interaction between the cylinder and the rear wall also causes a change in the convective structure in this region, which tends to equalise an airflow imbalance previously present over the cylinder, causing it to reduce its tendency to lean towards the centre of the enclosure at lower $Gr_{BD}$. Heat transfer from the cylinder is initially similar to an unconfined cylinder at low $Gr_{BD}$, but as this increases, the cylinder Nusselt number does not follow the $Gr^{0.25}$ dependence, but remains approximately constant across the testing range. This will serve as a basis for comparison with data obtained when the enclosure is vented to the environment in Chapter 5. This will give an indication of the impact of ventilation upon the cylinder heat transfer and enclosure conditions which will be presented in the following chapter.

For a multiple cylinder interaction in the enclosure, it has been seen that heat transfer is unaffected when a second constant temperature cylinder is placed above the bleed duct, similar to results found for unconfined cylinders. The enclosure effect does not appear to affect heat transfer in this configuration in any significant way. A slight increase in $Nu/Nu_0$ is observed for very large values of $TR$, but this is minimal at $1.05Nu$. A large drop in the cylinder heat transfer was observed when the constant temperature cylinder was placed below the bleed duct. When the lower cylinder is hotter, convective heat transfer from the upper cylinder is effectively eliminated, both as a result of the cylinder confinement and the preheating of the lower cylinder, reducing the thermal gradients on the upper cylinder surface.
Chapter 5

Leading Edge Ventilation

In this chapter, the influence of ventilation on the leading edge upon the bleed duct heat transfer and applicability of the ventilation method from a design and operational perspective is considered. Geometrical effects will also be taken into account including the introduction of a partitioning subspar, alternate bleed duct locations along with altering the relative open area between the inlet and exhaust vents. This is performed in order to determine whether an optimum ventilation configuration can be obtained under the geometrical constraints of the leading edge. The influence of ventilation on the leading edge is also considered for conditions replicating a hot day with the aircraft stationary on the tarmac under worst case thermal environmental load conditions.

5.1 Enclosure Ventilation

Typical enclosure ventilation consists of either mixing or displacement ventilation as detailed in Section 2.1.3. High temperature interior air can be removed by either mixing the cooler exterior air with the interior before being exhausted or displacing the warm buoyant air out of the enclosure via introducing cool air from below. The choice of which method is suitable for any given scenario depends on the effectiveness of the ventilation along with the feasibility of introducing the ventilation configuration and suitability of the ventilation type [30, 90]. Mixing ventilation is preferred where only single openings are available or openings are limited to being at a specific height within the enclosure. These systems also
benefit from being actively ventilated via a fan or other such device, as it is eases the introduction of cooler air into the upper region of a room/enclosure. Displacement ventilation benefits from multiple openings at different heights, with the exit higher than the inlet for cooling ventilation. Displacement ventilation is also beneficial for contaminant removal as it allows for the direct exhaustion of the interior air (along with the contaminants) without excessive dispersal throughout the area to be ventilated. It also allows for selective ventilation of contaminants or heat in a stratified environment if the heights of the openings are controlled [101], but an in-depth knowledge of the thermal or contaminant air profile is required for this to be effective. If the ventilation configuration is given sufficient consideration, no active ventilation component may be required and sufficient interior/component cooling can be achieved [123, 124].

5.1.1 Choice of Ventilation Configuration

For a specific industrial purpose such as in the leading edge case, the choice of ventilation configuration may be limited to the specific geometry of the enclosure and the availability of vent openings. The size of the ventilation openings required may also present a challenge, especially in situations where placing an opening on a surface can affect wall properties, such as structural integrity or lift/drag of the surface itself. This presents an additional challenge when ventilation is required \textit{a posteriori} due to some operational issue occurring (thermal degradation of a component or an aggressive thermal environment present) that was not anticipated during the early design phase. In such cases it may be quite difficult to achieve adequate enclosure ventilation due to the constraints imposed from an already mature design.

In order to determine the best configuration available for venting the leading edge enclosure, a number of ventilation configuration options were firstly considered. These openings allowed for a combination of mixing and displacement regimes to be present in the enclosure and their influence on the interior air temperature distribution were considered at the operation conditions of the leading edge.
Figure 5.1: Temperature distributions within the leading edge along the (a) horizontal centreline, (b) vertical centreline & (c) rear wall comparing the ventilation configurations investigated: -○- Unventilated enclosure, -◇- Single bottom vent, -□- Vents on upper and lower surfaces, -△- Two vents on lower enclosure surface. These openings are illustrated in (d).
5.1.1.1 Single Bottom Vent

In this configuration, the vent opening is placed below the bleed duct at the lowest point of the leading edge. This is illustrated in Fig. 5.1 (d) [i]. Effectively this produces a “worst case scenario” mixing ventilation regime. Similar to the findings of Yu and Joshi [6] for a single vented enclosure on the bottom with a heated element embedded in one of the vertical walls [34], placing an opening at this location on the bottom of an enclosure results in no appreciable entrainment of exterior air into the enclosure, with no reduction in the temperatures within. In effect, the enclosure behaves as an unventilated one. This is illustrated in the temperature profiles presented in Fig. 5.1.

This occurs primarily due to the fact that the single opening must allow for both an inlet and outlet flow path across the vent boundary. The mass flow of air entrained into the leading edge must be matched by an equal amount exhausted at the same opening. Any increase in the inlet airflow must also be counteracted by a corresponding rise in the air exhausted. As the opening is on the lowest point, it becomes extremely difficult to naturally vent the enclosure, due to the density difference between the hotter interior air and the surroundings (the coolest interior surface is $40^\circ C$ above ambient). The ventilation flow becomes choked at the opening which leads to ineffective venting of the leading edge, with little reduction in enclosure interior temperatures as seen here.

5.1.1.2 Upper and Lower Vent Openings

This configuration (Fig. 5.1 (d) [ii]) produces an optimal displacement ventilation configuration. Here the position of the vent openings allow for removal of the buoyant high temperature air from the upper region of the enclosure. This exhausted air is displaced by air entering the enclosure from the lower vent opening. The entrainment of air is also enhanced by the position of the bleed duct directly above the inlet vent opening, helping to draw more cooler exterior air into the enclosure.

A direct exit path for the bleed duct plume is available and also for the anticlockwise flow over the front surface of the leading edge. These combine to remove enough high temperature air from the enclosure to produce the drop in interior temperatures seen in Fig. 5.1. This is sufficient to reduce the interior air temperatures in the enclosure below
that of the coolest enclosure surface \( T_c \) as seen by the negative \( T^\star \) in Fig. 5.1 for this configuration.

### 5.1.1.3 Two Vent Openings on Lower Surface (with Height Difference)

Here the vent openings are limited to the lower enclosure surface as in Fig. 5.1 (d) [iii]. Due to the small height difference between the vent openings, minimal air displacement of the interior air out of the enclosure by the entrained air occurs, with the majority of this effect occurring in the region just above the lower enclosure surface. \( T^\star \) is seen to decrease further in the region below \( Y^\star = 0.3 \) as a result compared the upper region (Fig. 5.1) The remainder of the enclosure relies upon the mixing of the entrained exterior air and the core bulk air to reduce the interior air temperature. This is not as effective as the upper and lower vent configuration as a result, with the interior air temperatures in Fig. 5.1 higher for this configuration. Some reduction in \( T^\star \) is seen compared to the sealed and single bottom vent case, indicating that some removal of heat from the leading edge is possible.

### 5.1.2 Comparison of Ventilated and Unventilated Leading Edge

Section 5.1.1 shows the influence of vent position and configuration on the ventilation path and enclosure temperature profile. A marked difference in the interior air \( T^\star \) is noted depending upon the location of ventilation openings, with openings on the upper and lower surfaces allowing for the greatest reduction in air temperature. This is as a result of the ease of removing the hot buoyant air from the upper region whilst being displaced by the cooler entrained air from below. As discussed previously, the vent openings are limited to the lower surface (due to the presence of drainage holes on the actual aircraft wing structure). This limits the ventilation options available. As a single vent opening has been shown to be ineffective on the interior conditions, the option to place two openings on the lower surface is explored in this section as it provided the best possibility for ventilation of the leading edge under the current geometrical constraints. The following section will detail the enclosure conditions and the impact of ventilation upon the environment within the leading edge. Influence on bleed duct heat transfer is also considered.

The temperature rise for the bleed duct, interior air and the rear wall with \( q''_{BD} \) and \( Gr_{BD} \)
is presented in Fig. 5.2. At a similar input power, both surface and air temperatures in the leading edge are significantly lower due to the introduction of ventilation. Temperatures are up to $20^\circ C$ lower for the bleed duct and $10^\circ C$ & $8^\circ C$ lower for the rear wall and air temperatures respectively. This highlights the effectiveness of enclosure ventilation on reducing the component and air temperature within the leading edge, even when the openings are placed in an unfavourable (from a displacement ventilation perspective) lower position.

For $500\,\text{W/m}^2 < q''_{BD} < 2200\,\text{W/m}^2$, the interior air is at a higher temperature than the rear wall. Only as it approaches its maximum surface temperature above $q''_{BD} = 2200\,\text{W/m}^2$ does the rear wall temperature rise greater than that of the average interior air. This means that as the cylinder temperature increases, its influence on the rear wall is more pronounced than on the internal air, mainly due to the proximity of the bleed duct to the wall surface. The recirculation of the bleed duct plume in this region will also contribute to the increase in wall $T^\star$. This is similar to the unventilated scenario presented previously in Section 4.2. In the ventilated case however, the rear wall stays cooler than the interior air for a greater range of $q''_{BD}$, again highlighting the positive influence of ventilation on reducing temperatures within the leading edge.

Fig. 5.3 presents the change in interior air $T^\star$ with $q''_{BD}$ and $Gr_{BD}$ for the sealed and
CHAPTER 5. LEADING EDGE VENTILATION

Figure 5.3: Interior air $T^*$ $[(T - T_c)/(T_h - T_c)]$ distribution for the sealed (○) and ventilated (△) leading edge based on bleed duct input power (a) and Grashof number (b).

ventilated leading edges. Here a drop in $T^*$ from 0.4 to 0.25 is noted due to ventilation, but the trend in $T^*$ with $q''_{BD}$ and $Gr_{BD}$ remains similar between the unventilated and ventilated cases. As such, introducing ventilation does not alter the relationship between the change in bleed duct temperature and that of the surrounding air in the enclosure compared to the unventilated case. This decrease in $T^*$ highlights that $T_{BD}$ increases at a greater rate than the bulk air in the enclosure ($T_\infty$) which, similar to the unventilated leading edge in Section 4.2, indicates a reduction in convective heat transfer from the bleed duct for an increase in $q''_{BD}$.

Observing the rear wall temperatures in Fig. 5.4 shows that where $T^*$ stayed approximately constant for the rear wall in the sealed case, upon introducing ventilation $T^*$ increases as a result of the increase in power supplied to the bleed duct. This confirms the observation made in Fig. 5.2 where the temperature of the rear wall is observed to increase at a greater rate than that of the bleed duct, particularly at the highest $q''_{BD}$ values investigated. The temperature difference between the two is 10°C at $q''_{BD} = 500\,W/m^2$, whereas they are equal at 2200W/m$^2$. Initially at low bleed duct input powers, entrained air introduced from the ventilation helps cool rear wall. As $q''_{BD}$ increases the bleed duct temperature ($T_{BD}$) goes up, along with $q''_{rad}$ to the rear wall (amplified by the high emissivity of the enclosure.
surfaces). This increases $T_{RW}$, which cannot be counteracted by the entrainment of the cooler exterior air, leading to the rise in $T^*$ as a result. Care must be taken to ensure that an increasing bleed duct surface temperature does not increase the rear wall temperature enough to cause thermal or structural problems in the leading edge as a result of increased heat transfer to the wall. This is also of concern if the wall was constructed out of a composite material which has a much lower thermal conductivity than aluminium and would result in a different temperature and heat transfer to and from the vertical wall.

### 5.1.3 Interior Thermal Distribution

The interior air temperature profiles (taken along $Y^*$, $X^* = 0.25, 0.5 & 0.75$) for an increase in $Gr_{BD}$ are presented in Fig 5.5. Also presented in this figure are the corresponding rear wall temperatures on the vertical temperature profiles. At $Gr_{BD} = 1.4 \times 10^5$, $T^*$ increases with height in the region below $Y^* = 0.4$ from 0.1 to 0.35, above which the increase in $T^*$ with height is not as pronounced. This is seen in Fig. 4.2 (a) & (b). The reduction in $T^*$ with height above $Y^* = 0.4$ is a result of the high temperature air being trapped in the upper section of the leading edge due to its buoyancy. Cooler entrained air passing over the lower enclosure surface also reduces $T^*$ locally, but its influence is limited to the lower
half of the leading edge. Here $T^*$, measured at the lowest horizontal measurement plane ($Y^* = 0.25$), is less than half of that along the mid and three quarter height of the enclosure ($Y^* = 0.5, 0.75$). The influence of the ventilation opening on the rear wall temperature is also evident in the reduction in $T^*$ towards the bottom of the wall surface ($Y^* < 0.4$) seen in Fig. 5.5 (b). An approximately linear $T^*$ gradient is present on the surface, increasing from 0.13 to 0.25 with wall height. This was not seen in the unventilated configuration and can be attributed to the cooling effect of the entrained air on the lower section of the vertical wall.

Upon increasing $Gr_{BD}$ to $2.2 \times 10^5$, the temperature distribution within the leading edge does not change to any great extent. Overall $T^*$ drops from a maximum of 0.525 to 0.45 along the horizontal measurement planes (Fig. 5.5 [c]) due to the increase in bleed duct surface temperature and the increase in $T^*$ with height above $Y^* = 0.4$ decreases, indicating that the air in the upper section of the leading edge is becoming more isothermal compared to $Gr_{BD} = 1.4 \times 10^5$. Of interest to note is the drop in $T^*$ below 0 for the lowest measurement position on the vertical temperature profiles presented in Fig. 5.5 (d). This indicates that the air temperature at this location is below that of $T_c$ (the lowest enclosure surface temperature) and is a direct result of the cooler entrained air passing directly over the bottom surface. One possible reason for this is that the rise in $Gr_{BD}$ increases $T_{BD}$ sufficiently above the ambient exterior temperature to increase the mass-flow of air entrained into the leading edge, producing this enhanced cooling effect in the lower region. This increasing influence of $T_{BD}$ is also seen in the homogenization of $T^*$ above $Y^* = 0.25$, seen in the vertical temperature profiles in Fig. 5.5 (b, d, f & h), where plume mixing begins to dominate. This highlights an ideal location for the placement of temperature sensitive components as the large temperature gradient and resultant convective cooling would provide enhanced heat transfer when placed above the lower surface. This effect diminishes with enclosure height as the ventilation path is limited to the lower portion of the enclosure between the inlet and outlet vents.

Overall, as seen in the unventilated leading edge, $T^*$ continues to drop as $Gr_{BD}$ increases due to the increase in $T_{BD}$ over the interior air. $T^*$ in the upper region of the leading edge ($Y^* = 0.5, 0.75$) converges at 0.3 due to the confinement and mixing of the high
Figure 5.5: Interior temperature profiles taken along the horizontal planes ($Y^* = 0.25(□)$, $0.5 (○)$ & $0.75 (△)$) (a, c, e, & g) and along the vertical planes ($X^* = 0.25(□)$, $0.5 (○)$ & $0.75 (△)$) (b, d, f, h). $Gr = 1.4 \times 10^5$ (a,b), $2.2 \times 10^5$ (c,d), $2.8 \times 10^5$ (e,f) & $3.3 \times 10^5$ (g,h). The rear wall temperatures (○) are superimposed onto the vertical temperature profiles for comparison.
temperature plume. At $Y^* = 0.4$ in Fig. 5.5 (g), $T^*$ drops to 0.2 due to the cooler entrained air reducing local fluid temperatures. The drop in $T^*$ below 0 at the lowest measurement point on the vertical temperature profiles also remains, indicating that the cooler exterior air is still effectively being entrained into the leading edge as $Gr_{cyl}$ increases, providing a degree of cooling in this region.

Rear wall temperatures are presented in Fig. 5.6. At $Y^* < 0.6$, $T^*$ increases with Grashof number. However as $Gr$ increases, the wall becomes isothermal, resulting in a lower thermal gradient at $Gr_{BD} = 3.3 \times 10^5$. This demonstrates how ventilation helps to reduce the temperature of the rear wall, but only in the region below $Y^* = 0.6$, as the influence of ventilation is limited to the lower region of the leading edge. As $Gr_{BD}$ increases, the increase in $T_{BD}$ and the resultant increase in $q''_{rad}$ to the rear wall begins to outweigh the cooling effect of the entrained air, the wall becomes more isothermal with vertical conduction through the wall dominating, producing an almost constant $T_{RW}^*$ of 0.25 at $Gr_{BD} = 3.3 \times 10^5$. 

Figure 5.6: Ventilated leading edge rear wall temperature profiles for $Gr_{BD} = 1.4 \times 10^5 (\circ), 2.5 \times 10^5 (\Diamond), 3.3 \times 10^5 (\triangle) \& 3.9 \times 10^5 (\square)$
5.1.4 Effect of Ventilation on Bleed Duct Heat Transfer

The introduction of ventilation to the enclosure allows for a drop in component temperature, enclosure air temperature or ideally both. From Fig. 5.2 it is clear that both of these effects occur when the leading edge is vented. As a result of $T_{BD}$ decreasing to a greater extent than $T_{\infty}$, a drop in the temperature difference between the bleed duct and the surrounding air occurs. This results from a decrease in bleed duct surface temperature due to convection to the entrained air and a reduction in the interior air temperature due to the exhaustion of air from the enclosure through the exit vent. This temperature difference for the sealed and vented cases is presented in Fig. 5.7. On average the system is operating 10°C cooler ($T_{BD} - T_{\infty}$) compared to the sealed leading edge. This would require a drop in the power input to the bleed duct of between 275 – 300 W/m² to achieve without ventilation. This represented a reduction of between 19 – 35% depending upon the bleed duct input power.

The Nusselt number for the bleed duct for the sealed and vented leading edge cases is presented in Fig. 5.8. As a result of the decrease in $T_{BD}, T_{\infty}$ and the difference between the two (Fig. 5.7), $N_{UBD}$ is increased from approximately 9 to 16. This represents a significant increase in convective heat transfer from the bleed duct of 77%. As $N_{UBD}$ does not change significantly over the range of $Gr_{BD}$ tested (remains approximately 16), an average
heat transfer coefficient can be obtained for both a sealed and ventilated bleed duct. This may prove useful for reducing numerical modelling complexity as it can be used as an appropriate boundary condition for the bleed duct during the calculations. Based upon the data presented in Fig. 5.8, a bleed duct h value of 5.3 and 10.5 $W/m^2K$ is obtained for the sealed and vented enclosure respectively.

5.2 Partitioning of Leading Edge

The presence of a partitioning structure has the possibility of confining the high temperature fluid to a smaller region of an enclosure, thus reducing heat transfer from within [79,80,82]. This also will have serious consequences if it interferes with the ventilation path as it can reduce the amount of cool air passing through the enclosure. To investigate this, a $220(H) \times 580(D) \times 10(W)$ mm partition was introduced to the enclosure (as illustrated in Fig. 3.4). This is representative of a front sub-spar in the wing structure and will present a block in the airflow path between the rear and forward openings in the leading edge. The objective is to analyse the impact of this partitioning structure upon the leading edge ventilation and the heat transfer from the bleed duct.

The bleed duct, interior air and enclosure rear wall temperature rise (over $T_c$) is presented in Fig. 5.9. Shown here also is the data for the unventilated case. As seen in the unpartitioned case, the surface temperature of the bleed duct and the average enclosure air
is reduced compared to the sealed enclosure. Comparing the data presented in Fig. 5.2 for
the unpartitioned case shows that the drop in temperature rise is smaller as a result of the
partitioning. This was somewhat to be expected as the partition blocks the ventilation path
between the inlet and outlet vents which will reduce the ability of the ventilation solution
to reduce the interior temperatures.

Similar to the conditions observed for the unpartitioned configuration, the rear wall
temperature remains below that of the interior air for the majority of the bleed duct input
power. As $q''_{BD}$ increases above 1750 W/m² the wall and air temperatures become equal,
indicating that the rear wall increases in temperature greater than the cylinder for a similar
increase in $q''_{BD}$. This increase in $T_{RW}$ in relation to $T_{∞}$ as $q''_{BD}$ increases is less abrupt than
observed for the unpartitioned case in Fig. 5.2, due to the presence of the partition reducing
the ability of entrained ventilation airflow to keep the rear wall cool, leading to a more
gradual rinse in $T_{RW}$ in relation to $T_{∞}$. The difference in the wall and air temperatures is
constant for the unpartitioned case between $500\, \text{W/m}^2 < q''_{BD} < 1250\, \text{W/m}^2$, while this only
occurs for the partitioned case between $500\, \text{W/m}^2 < q''_{BD} < 800\, \text{W/m}^2$.

Fig. 5.10 presents the change in interior air $T^*$ with $Gr_{BD}$ and $q''_{BD}$ for the sealed
and partitioned ventilated leading edges. Again a drop in $T^*$ is noted due to ventilation, but as a result of the addition of the partition reducing the influence of the ventilation and increasing the bleed duct surface temperature, the interior air $T^*$ is lower than the unpartitioned configuration. The change in $T^*$ with $Gr_{BD}$ also remains similar between the unventilated and partitioned cases with only a slight increase in $T^*$ above $Gr_{BD} = 3 \times 10^5$ for the partitioned case compared to the unventilated one. This decrease in $T^*$ with $Gr_{BD}$ again highlights how $T_{BD}$ increases at a rate which is greater than that of the increase in the interior bulk air temperature.

The rear wall profiles in Fig. 5.11 show an increase in $T^*$ with $q''_{BD}$ similar to the data presented in Fig. 5.4 for the unpartitioned case. At low $q''_{BD}$, the rear wall is cooled as a result of ventilation, but as $q''_{BD}$ increases further this is outweighed by the increase in $T_{BD}$ and the resultant heat transfer to the wall (primarily through radiative exchange and the presence of the bleed duct plume). This is similar to the unpartitioned scenario. It is evident that ventilation does help reduce rear wall temperatures compared to the sealed case, particularly at lower bleed duct input powers, but as $T_{BD}$ increases so too does its influence on rear wall temperatures, which negate any ventilation effects. It also demonstrates how in the actual aircraft leading edge, the bleed duct is the main influence on the enclosure
5.2.1 Interior Thermal Distribution

The interior temperature profiles for the partitioned leading edge are presented in Fig. 5.12. Here the effect of placing the partition between the inlet and outlet vents of the leading edge is evident. For all $Gr_{BD}$ values investigated, $T^*$ rises significantly beyond the partition (located at $X^* = 0.72$). This is particularly noticeable along the lowest measurement plane, $Y^* = 0.25$, where $T^*$ rises from 0.175 to 0.4 at $Gr_{BD} = 1.8 \times 10^5$. Here the entrained exterior air has some influence in reducing the temperatures in the lower region of the leading edge before the partition. Due to the blocking effect of the partition, this air cannot travel directly towards the exhaust vent at the front of the leading edge, but is forced to interact with the core bulk air within the leading edge. In this scenario, the entrained air is used more effectively as it simply does not just exit the enclosure after passing over the lower enclosure surface only.

The vertical temperature profiles in the rear partitioned space are similar to those found in the unpartitioned case. Here a large decrease in $T^*$ is present below $Y^* = 0.4$ due to the presence of the entrained cooler air in this region, with $T^*$ increasing above this due to
Figure 5.12: Interior temperature profiles taken along the horizontal planes ($Y^* = 0.25 (\Box)$, $0.5 (\diamond)$ & $0.75 (\triangle)$) (a, c, e, & g) and along the vertical planes ($X^* = 0.25 (\Box)$, $0.5 (\diamond)$ & $0.75 (\triangle)$) (b, d, f, h). $Gr = 1.8 \times 10^5$ (a,b), $2.7 \times 10^5$ (c,d), $3.3 \times 10^5$ (e,f) & $3.7 \times 10^5$ (g,h). The rear wall temperatures (◦) are superimposed onto the vertical temperature profiles for comparison.
CHAPTER 5. LEADING EDGE VENTILATION

thermal confinement. In the measurement plane taken beyond the partition \((X^* = 0.72)\), \(T^*\) is significantly hotter and increases linearly with height. At \(Gr_{BD} = 1.8 \times 10^5\) this increases from 0.4 to 0.57 in this region. This is a direct result of only the more buoyant high temperature air passing over the top of the partition and into the front partitioned space.

The thermocouple located at \(X^* = 0.75\) on the \(Y^* = 0.75\) measurement plane in Fig. 5.12 (a) is located just beyond the top of the partition and as such give an indication of the temperature of the air passing into the front of the leading edge. At \(Gr_{BD} = 1.8 \times 10^5\) this measurement is higher than that of rest of the \(Y^* = 0.75\) measurement plane in the rear of the enclosure. This means that the air passing into the front of the leading edge is only the highest temperature air from the upper most section of the enclosure. This contributes to the rise in \(T^*\) beyond the partition seen in Fig. 5.12 (a, c, e & g). As \(Gr_{BD}\) increases and the plume mixing effect begins to dominate, this spike in \(T^*\) reduces until it is the same value as the rest of the \(Y^* = 0.75\) plane, indicating that the upper region of the leading edge has become more homogeneous in temperature as \(Gr_{BD}\) increases.

Due to the blocking effect of the partition also, there is no drop in \(T^*\) below 0 (which was noted in Fig. 5.5 for the unpartitioned case) at the lowest point in the vertical temperature profiles just above the bottom enclosure surface. Air entering the leading edge via the rear vent travels along the lower surface until it reaches the partition where it turns and flows to the rear of the leading edge. Here it is entrained by the bleed duct before travelling through the rest of the enclosure. In this scenario, practically all of the air entering the leading edge travels over the bleed duct surface to aid in convective cooling and subsequent heat transfer. However at similar bleed duct input powers and \(Gr_{BD}\), the bleed duct surface temperature is in fact higher than the unpartitioned case.

\(T^*\) measured on the rear wall surface also exhibits an increase with \(Gr_{BD}\). As a result of the ventilation cooling the wall initially at \(Gr_{BD} = 1.8 \times 10^5\), a gradient of \(T^* = 0.15 - 0.22\) is present through the wall height, in comparison to \(T^* = 0.13 - 0.24\) for the unpartitioned case. The ability of the entrained exterior air to cool the lower portion of the wall and produce a thermal gradient along its height is reduced as a result of the increase in both \(T_{BD}\) and \(T_\infty\). As \(Gr_{BD}\) increases, the influence of \(T_{BD}\) increases, with the wall becoming more isothermal.
It is evident that the enclosure conditions are dominated by the bleed duct, particularly as its surface temperature become significantly hotter than the other enclosure surfaces. The impact of the partition on the leading edge is that it reduced the thermal gradient produced on the rear wall surface along with creating two distinctive thermal regions in the leading edge: the cooler rear section where the bleed duct and entrained air dominate and the front partitioned space containing significantly hotter air. As the bleed duct power input increases however, this step change in temperature between the two zones diminishes as plume mixing begins to dominate in the rear space leading to a more isothermal section, particularly above the enclosure mid-height. The block in the ventilation path between the front and rear vents is also a factor, as local cooling over the lower surface is reduced, with $T^*$ not dropping below 0 here as was found for the unpartitioned case. The entrained air is used more effectively throughout the leading edge however as it does not simply pass over the lower surface before being exhausted, but is forced to interact with the core bulk air. However, this is still not sufficient to produce any meaningful drop in the interior air temperatures.
5.2.2 Effect of Partitioning on Bleed Duct Heat Transfer

Introducing ventilation to the leading edge was seen to produce an increase in \( Nu_{BD} \) of 77\%. As a result of the increase in the bleed duct surface temperature and the core interior air temperature due to the impact of partitioning the leading edge, the reduction in \( \Delta T_{BD} \) decreases from approximately 10\(^\circ\)C to 5\(^\circ\)C compared to the unpartitioned case, as presented in Fig. 5.14. This change would indicate that there is a drop in the convective heat transfer from the bleed duct to the leading edge and out to the surroundings as a result of ventilation.

Fig. 5.15 presents the change in \( Nu_{BD} \) as a result of the introduction of a partition to the enclosure. Here \( Nu_{BD} \) is reduced from 16 to a constant 13 across the \( Gr_{BD} \) range investigated. Bleed duct heat transfer is still enhanced compared to the unventilated case, so there is still an advantage to venting the enclosure in the presence of a partition. The heat transfer coefficient also drops from 10.5 to 7.6 \( W/m^2K \).
5.3 Case Study at Hot Day Conditions

This section outlines the conditions within the leading edge for a specified aircraft operating condition, namely on a hot standard day in a desert surrounding. The purpose of this is to investigate the thermal distribution within the leading edge in such a scenario and to observe the influence of enclosure ventilation and the impact of geometrical changes on the enclosure ventilation. The aim is to understand the effectiveness of ventilation on the leading edge under these worst case scenario conditions. For this section, the upper and lower leading edge surfaces are maintained at 70°C and 100°C respectively as before and the bleed duct is maintained at 135°C as detailed by AIRBUS from test data acquired in this environment. As this was based upon actual aircraft operating in observable conditions, the placement of the ventilation holes were also limited to the lower section of the enclosure in order to ensure it was similar to any realistic ventilation strategy for the leading edge.

5.3.1 Comparison with Empty and Partitioned Leading Edge

The following section describes the conditions present in the leading edge for an unpopulated enclosure (Case A), with the addition of the bleed duct (Case B) and also the presence of both the bleed duct and partition (Case C). The geometrical configurations are outlined
Table 5.1: Outline of geometrical configurations for Case A - E

<table>
<thead>
<tr>
<th>Case</th>
<th>Bleed Duct Position</th>
<th>Partition</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>-</td>
<td>No</td>
</tr>
<tr>
<td>B</td>
<td>Above rear vent</td>
<td>No</td>
</tr>
<tr>
<td>C</td>
<td>Above rear vent</td>
<td>Yes</td>
</tr>
<tr>
<td>D</td>
<td>Adjacent to partition</td>
<td>Yes</td>
</tr>
<tr>
<td>E</td>
<td>Upper corner of leading edge</td>
<td>Yes</td>
</tr>
</tbody>
</table>

in Table 5.1. Fig. 5.16 presents the interior air temperature distribution for the leading edge for the sealed, ventilated and ventilated enclosure without the bleed duct for comparison.

Without the presence of the bleed duct, the temperature \( T^* \) along the horizontal centreline in Fig. 5.16 (a) is the lowest of the configurations considered. In the region below \( X^* = 0.4 \), \( T^* \) falls towards 0 and is below 0 at \( X^* = 0.125 \). This indicates that in this region, the local air temperature drops below that of the coldest enclosure surface (from Eq. 3.2). This in turn signifies how the cooler exterior is being entrained into the leading with ease in the region close to the rear vent opening, reducing \( T^* \) locally. This effect does not influence \( T^* \) beyond \( X^* = 0.4 \) where it remains at approximately 0.05 for the remainder of the enclosure width.

A similar condition can be seen for the empty enclosure (Case A) along the vertical centreline in Fig. 5.16 (b). Here, almost half of the height of the air in the enclosure is below the temperature of the coldest enclosure surface \( (T^* < 0) \). \( T^* \) increases almost linearly with height, reaching a maximum of 0.57 at \( Y^* = 0.875 \). This \( T^* \) profile indicates that in the lower half of the leading edge the air is cooler due to the the ventilation path over the lower surface and the confinement of the high temperature air in the upper section of the leading edge as was previously noted.

This is confirmed when observing the \( T^* \) on the rear wall surface in Fig. 5.16 (c). Similar to the air temperatures, \( T^* \) increases with wall height. Above \( Y^* = 0.7 \) a large increase in \( T^* \) is noted as the confined hot air heat the upper section of the wall (the air \( T^* \) increases from 0.26 to 0.57 in the region above \( Y^* = 0.7 \)). At all points on the wall surface, \( T^* \) remains below 0, indicating that its temperature remains below the lowest enclosure surface temperature \( (T_c) \) due to the impact of ventilation and the lack of the bleed duct.
Figure 5.16: Horizontal centreline (a), vertical centreline (b) and rear wall (c) temperature profiles for the empty (♦), bleed duct only (○) and bleed duct and partitioned leading edge (□). These correspond to Case A - C respectively.

With the bleed duct present, a greater change in $T^*$ distribution in the leading edge is observed. For the bleed duct only scenario (Case B), $T^*$ is greatest towards the rear enclosure wall ($0 < X^* < 0.4$) before dropping to 0.25 towards the front section of the enclosure.
CHAPTER 5. LEADING EDGE VENTILATION

This occurs as a result of the influence of the thermal plume from the bleed duct on the conditions in the rear section of the enclosure, raising $T^*$ in this region. This local increase in $T^*$ is in contrast to the empty enclosure where the addition of ventilation produced a drop in $T^*$ here. Along the vertical centreline, the change in $T^*$ with height is lower compared to the unpopulated enclosure. This mainly occurs as a result of the additional influence of the bleed duct plume on the interior, as its mixing effect aids in the dispersing of heat throughout the leading edge. $T^*$ decreases below $Y^* = 0.4$ due to the presence of the ventilation flow over the lower enclosure surface in this region. $T^*$ becomes negative below $Y^* = 0.2$, again highlighting that the air temperature in this region is below that of $T_c$, indicating that the entrained ventilation flow is only limited to this region directly above the surface.

The temperature on the vertical wall indicates approximately isothermal conditions on the surface, with a constant $T^* = 0.08$ present, increasing slightly to 0.09 above $Y^* = 0.8$ due to the presence of confined hot air in this region. This lack of thermal gradient on the wall compared to the unpopulated condition suggests that the influence of the bleed duct is important upon the resultant wall thermal profile. The presence of the high temperature bleed duct increases the wall temperature enough to allow for an isothermal condition to be present.

With the addition of the partition to the leading edge (Case C) a block is formed between the inlet and outlet vents, as noted previously. The main effect of this is to recirculate the entrained airflow in the leading edge in order to more effectively distribute it throughout the leading edge. The change in bleed duct plume orientation changes the overall airflow structure inside and due to the coupling of the thermal and velocity fields results in the change in internal $T^*$ distribution within the leading edge. The primary effect of partitioning is to reduce the local influence of the bleed duct plume in the rear partitioned section of the enclosure as it now attaches to the rear wall at approximately mid-height, before travelling along the upper leading edge surface before entering the front partitioned space. As a result of this, mixing of the bleed duct plume with the rear partitioned space is lower than in the unpartitioned case, with a reduction in $T^*$ in this region visible in Fig. 5.16 (a), but this high temperature air is confined to the upper region of the enclosure. As this high
temperature plume flow travels into the front section, a large rise in $T^\star$ is seen. This arises as the only air entering this region is the high temperature plume which passes over the top of the partition where it mixes with the local air and increases its temperature. This high temperature air poses a problem for exhaustion through the opening on the lower surface as it will be inherently buoyant and difficult to remove from the enclosure.

The vertical temperature profile shows a linear increase in $T^\star$ with height, indicating that the interior air is thermally stratified with little influence from the bleed duct plume or ventilation flow upon it. If this profile is extrapolated down to the position at $Y^\star = 0$, then $T^\star$ becomes 0 at this point which indicates that the ventilation flow over the lower surface (present for the unpartitioned case) does not influence the temperatures in this region, with stratification present and vertical conduction through the air as opposed to convective mixing. This leads to an apparent drop in $T^\star$ in the rear partitioned space (as seen in Fig. 5.16 (a)), but the overall enclosure is not significantly cooler, due to the confinement of the high temperature air to the upper region and isolation in the front partitioned space.

$T^\star$ on the rear wall surface increases slightly from 0.08 to 0.1. Overall the temperature profile on the vertical wall for the partitioned case is similar to the unpartitioned case. Due to plume attachment on the vertical wall, a small increase in $T^\star$ is noted at the mid-point of the wall height in Fig. 5.16 (c) as a result. Also the increase in $T^\star$ above $Y^\star = 0.8$ is also as a result of the confinement of the high temperature air to the upper section of the enclosure. The impact of partitioning on the rear wall is minimal as a result.

### 5.3.2 Bleed Duct Placement

The following section details the leading edge interior when the bleed duct is placed at various positions within the leading edge. This was performed in order to investigate whether there is any advantage to having the bleed duct located in different positions. From an application perspective, it is worthwhile to investigate the conditions present in the leading edge for different bleed duct placements in order to determine if an optimum position exists with respect to the internal temperature distribution and ventilation efficiency. The influence of enclosure partitioning upon the bleed duct heat transfer has been described in Section 5.2. Here it was shown that the effect of the partition is to reduce heat transfer
Figure 5.17: Horizontal centreline (a), vertical centreline (b) and rear wall (c) temperature profiles for the various bleed duct locations: above rear vent opening (♦), adjacent to partition over lower surface (◦) adjacent to rear wall in upper corner (□). These correspond to Case C - E respectively. Bleed duct locations are illustrated in (d) (i) - (iii).
compared to the unpartitioned condition, primarily as a result of the increase in both the bleed duct and interior air temperature. The placement of the bleed duct within the enclosure and its influence upon the final enclosure conditions therefore may also be an important parameter to consider with regard to the design of such enclosures. The following section outlines the influence of placing the bleed duct in two alternative positions with the leading edge (as shown in Fig. 5.17 [d]).

The temperature distributions inside the leading edge for these alternate positions (Fig. 5.17) show a marked change within the leading edge depending upon bleed duct position. The overall temperature distribution within the leading edge when partitioned consists of a relatively low temperature region in the rear of the enclosure near the bleed duct, with an increase in $T^\star$ in the front partitioned space. This is seen for the original bleed duct position where $T^\star$ increases from $0.19 - 0.27$ along the horizontal centreline between the rear and front sections of the leading edge. This occurs as a result of the block in ventilation path and directing the bleed duct plume flow towards the front partitioned section. Thermal confinement also plays a role in keeping the high temperature air in the upper section of the enclosure.

Moving the bleed duct away from the rear vent opening along the lower enclosure surface (as for Case D) causes an increase in $T^\star$ with distance from the rear wall. $T^\star$ increases almost linearly from $0.16 - 0.25$ in the rear half of the enclosure ($0 < Y^\star < 0.5$), followed by a large increase to $0.39$ due to the presence of the bleed duct plume at the measurement point. Beyond the partition, $T^\star$ drops to $0.34$, but this is still higher than Case C in this location. As a result of the bleed duct position, the interaction of the plume with the core bulk air in the leading edge is altered, which changes the temperature distribution as shown. No longer does the plume merely travel along the upper surface of the leading edge before passing over the sub-spar and into the front partitioned space, but actively mixes with the surrounding air in the rear of the leading edge. As a result a smaller portion of the high temperature plume flow will pass into the front of the enclosure to be exhausted to the environment. This produces the increase in $T^\star$ with $X^\star$ seen along the horizontal centreline seen in Fig. 5.17 (a), as well as the higher $T^\star$ seen along the vertical centreline in Fig. 5.17 (b). The overall $T^\star$ is greater for Case D as well as a reduction in $\Delta T^\star/\Delta Y^\star$, 123
again indicating enhanced mixing of the plume occurring. The increase in $T^*$ also occurs as a result of the reduction in the entrainment of cooler exterior air into the leading edge via the rear vent opening from the increase in bleed duct to vent opening distance.

Increasing the distance between the bleed duct and the rear wall produced a reduction in the wall surface temperatures as it is not under the direct influence of the bleed duct and its plume as occurred for Case C. The temperature profile in Fig. 5.17 (c) also demonstrates an increase in $T^*$ with height, similar to the conditions found for the unpopulated enclosure in Fig. 5.16. The lower section of the wall will be kept cooler due to the entrained exterior air recirculating in this region as opposed to either being entrained by the bleed duct or travelling towards the front vent as occurred in the unpartitioned case. This confirms that the proximity of the bleed duct to the wall leads to a more isothermal surface (with the exception of discrete points where the plume attaches), with a thermal gradient present in an unpopulated or large bleed duct to wall separation configuration.

Placing the bleed duct in the upper section of the leading edge (as per Case E) both increases the vent to bleed duct separation distance and also the amount of thermal confinement in the leading edge. Observing the horizontal centreline temperature in 5.17 (a), a large reduction in $T^*$ is visible in the rear section ($X^* < 0.7$) with an increase in $T^*$ beyond the partition. Initially this would seem like an ideal scenario, with a lower $T^*$ in the area adjacent to the cylinder and a higher $T^*$ in the front partitioned space near the exit vent to be exhausted. The confinement of this high temperature air to the upper region is of no benefit from a ventilation perspective. The effect of confinement on the temperatures in the upper section of the enclosure is also evident from the vertical temperature profile in 5.17 (b) where below $Y^* = 0.3$, $T^*$ remains relatively low (0.06 – 0.07), but $T^*$ begins to increase significantly towards the bleed duct location. The largest rise is seen between $Y^* = 0.66 – 0.83$, where $T^*$ increases from 0.19 – 0.34. $T^*$ is expected to increase even greater with height beyond this, but this is not captured in the thermocouple positions available. This confinement of hot air is not ideal from a ventilation perspective, as it remains in the upper region of the enclosure and is not easily exhausted via the openings on the lower surface of the leading edge.

The bleed duct placed in the upper portion of the enclosure also has a significant impact
upon the temperature profile on the vertical wall (as shown in Fig. 5.17 [c]). Here a maximum \( T^* \) of 0.275 is observed at \( Y^* = 0.75 \) which is over double the maximum seen in either of the other two positions (\( T^* = 0.13 \) for Case C at \( Y^* = 0.5 \)). As a result of vertical conduction through the wall, \( T^* \) also remains higher than the other cases along this surface. In the absence of adequate ventilation and under the influence of confinement, the only method to distribute this heat effectively is via conduction in the wall, but this is neither beneficial nor welcome. A small reduction in wall \( T^* \) in the lower region below that of Case C may be attributable to the presence of the bleed duct only in the upper region, the entrained air recirculating in the lower region cooling this area locally, or a mixture of both. This effect is highly localised and does not provide any significant benefit other than to this immediate area.

5.3.3 Flow Structure

When it comes to effective enclosure ventilation, one of the key challenges is the ability to ensure that the flow path through the enclosure results in the entrained air interacting with the core air to the greatest extent (be it through mixing or displacement) to help reduce interior temperatures. The ease of which the air is exhausted is also of concern as this is a naturally ventilated environment, with no means of actively removing air from within so adequate consideration must be taken for this parameter. Doing so requires an extensive knowledge of the flow structure within the enclosure in order to design a suitable ventilation strategy which takes into account the thermal/velocity fields, the vent locations and geometric configuration of the leading edge.

Due to the strong coupling between velocity and thermal fields in natural convection, a change in air or surface temperature has the possibility to radically alter the flow path (and vice versa). Hence the degree to which ventilation alters the interior thermal distribution could be seen as an indicator as to how much the flow structure will change i.e. a large impact of ventilation may radically alter the flow structure while ineffective ventilation may not change it in any significant way. The interaction between the leading edge and the bleed duct also adds an additional complexity to the analysis. Such an interaction between an isolated heat source inside such a differentially heated cavity is not well documented in
the published literature, particularly when ventilation is included. The following section presents particle image velocimetry (PIV) images taken within the leading edge for the different enclosure configurations investigated. The purpose of this is to inspect the flow structure in the leading edge under the various geometrical constraints and how this impacts the ventilation of the enclosure.

Fig. 5.18 presents the airflow over the lower surface of the leading edge where no bleed duct is present (Case A). Here the influence of ventilation is limited solely to the region above this surface. Exterior air is entrained via the rear vent, where the majority of the air travels over the lower surface towards the front of the leading edge. A small portion travels over the lower section of the rear wall before detaching and rejoining the airflow over the lower surface, which produces the drop in $T^*$ at the bottom of the wall as seen in Fig. 5.16 (c). This also accounts for the drop in $T^*$ below 0 on the temperature seen along the vertical centreline in Fig. 5.16 (b), where the cooling effect of this entrained flow is seen in this region. The remainder of the enclosure is for the most part unaffected by the introduction of the vent openings, with a anti-clockwise recirculation (not shown in this figure) present in the upper section of the leading edge, which does not interact with the entrained air.

The presence of the bleed duct in the leading edge (as in Fig. 5.19) illustrates how it dominates the flow structure within the leading edge. In this configuration the ventilation path is still similar to the empty vented enclosure, where the exterior air flows over the lower surface only, but is also entrained by the bleed duct and sets up a strong anti-clockwise recirculation in the rear section of the leading edge, adjacent to the rear spar. Its influence is limited to this rear region, with minimal interaction with the rest of the leading edge, resulting in a relatively stagnant core section within the enclosure. Whilst the location of the bleed duct does allow for more entrainment of air into the leading edge via a greater
Figure 5.19: PIV flow structure for vented leading edge with bleed duct (Case B).
temperature differential at the vent boundary (leading to a greater airflow velocity at the exit vent), ineffective mixing within the leading edge along with the poor exit vent location means the majority of the enclosure is unaffected by the introduction of ventilation, with the only benefit to be found in the region directly above the lower enclosure surface.

The addition of the subspar partition in Fig. 5.20 alters the flow structure primarily as a result of the blockage between the inlet and outlet vent openings. This creates a barrier and prevents the entrained ventilation airflow from merely passing over the lower enclosure surface before being exhausted. Here the plume is also observed to be highly damped and orients towards the rear wall of the enclosure. A contributing factor to this is the change in aspect ratio of the enclosure in which the bleed duct is present. It attaches to the rear wall at approximately mid-height, setting up a small anti-clockwise recirculation in the upper region adjacent to the rear wall. It then travels along the upper surface of the enclosure into the front partitioned space, opposing the enclosure flow travelling in the opposite direction. Here the plume flow then attaches to the cooler partition surface, allowing it to travel down towards the front of the enclosure before exiting at the vent opening. This is beneficial because it generates a path for the buoyant plume to flow to the lower section of the enclosure and avoids its confinement in the upper region as was found for the unpartitioned case. This effect is dependant upon the temperature of the partition and any large increase in this surface temperature may negate this benefit.

Since the direct path between the rear and front vent openings is blocked by the presence of the partition, the entrained air which initially travels over the lower surface is forced back towards the rear of the enclosure where it is entrained by the bleed duct. This means that in addition to providing local cooling above the lower surface, it is further mixed with the remainder of the enclosure air, enhancing the utilisation of the ventilation flow within the leading edge. Whilst this has been shown to not have any positive influence on either the enclosure temperature distribution or bleed duct heat transfer, it is still noteworthy nonetheless as it could possibly lead to a method of optimisation in future designs. A region of backflow into the enclosure is also present at the front vent opening where a small anticlockwise recirculation occurs. This means that the outlet flow at the front vent is exhausted through a narrower opening than the case without the partition, effectively
Figure 5.20: PIV flow structure for vented leading edge with bleed duct & partition (Case C).
reducing the open area of the vent used to remove high temperature air from the enclosure. If this were to increase further it could prove detrimental to the effective ventilation of the leading edge.

Placing the bleed duct adjacent to the subspar as shown in Fig. 5.21 greatly alters the interaction of the plume with the interior of the leading edge. As the plume is found to be a main driver of the flow structure, any change of its location can have a significant impact upon the resultant enclosure conditions. In this configuration, the plume sets up an anticlockwise recirculation in the rear partitioned space which is fed by air entrained along the lower enclosure surface. This recirculation mixes the air in the leading edge to a greater extent, but as it is mainly confined to the rear of the enclosure with only a small portion travelling over the partition into the front of the enclosure, it mainly just serves to increase the local air temperature as this hot air cannot be easily removed from the leading edge. A region of backflow is noted again at the front exhaust vent, reducing the open area available for exhaustion of air from the enclosure, also contributing to the increase in temperature observed.

Placing the bleed duct in the upper region of the enclosure adjacent to the rear wall as in Fig. 5.22 reduces the influence of the bleed duct on the overall flow structure. Since it is affected by the presence of the confining upper and rear walls of the enclosure, its plume is limited to a small recirculation on either side of the bleed duct. The confinement effect has been shown by Sadeghipour and Razi [86] and Koizumi and Hosokawa [29] (amongst others [5, 28, 87]) to be influential on the flow structure and heat transfer from a heat source, with typically an optimum position available, beyond which the heat transfer can be severely compromised. In this configuration, the majority of the high temperature air is confined to the upper region of the leading edge, producing the sharp rise in temperatures seen in Fig. 5.17 (b). Very little air travels over the partition towards the front section, with the majority of this flow travelling up along the partition surface from the lower region and not directly from the plume flow as was the case for the previous two positions. As such it is not the hot plume flow from the bleed duct which is being exhausted, but merely the (relatively) cooler air from above the lower enclosure surface.

The presence of the partition along with the lack of an entrainment source also means
Figure 5.21: PIV flow structure for vented & partitioned leading edge with bleed duct adjacent to partition (Case D).
Figure 5.22: PIV flow structure for vented & partitioned leading edge with bleed duct in upper rear corner (Case E).
that a secondary plume forms over the bottom surface. Its oscillation alternates between travelling towards the partition and back to rear wall where it forms an anti-clockwise recirculation in this region. Similar to the other partitioned configurations, a backflow into the leading edge is noted at the exit vent. In this case however, it forms its own recirculation cell in the lower region at the front of the partition which travels down the lower surface, over the partition before being exhausted again. In the end very little high temperature air from the upper region of the enclosure where the bleed duct resides can actually be exhausted from the leading edge for this configuration. As such this position cannot be recommended in creating a suitable ventilation solution.

The flow images across the exit vent at the front of the leading edge in Fig. 5.23 highlight how the airflow exiting the leading edge is affected by both bleed duct placement and the presence of a partition. For the unpartitioned leading edge both with and without the bleed duct, a similar airflow is seen crossing the vent boundary. The addition of the cylinder in Fig. 5.23 (b) leads to an increase in the maximum velocity (and mass flow rate) of the air from approximately 0.07 m/s to 0.09 m/s, this being the only difference between the two. The major impact of the introduction of the partition is to produce a region of backflow into the enclosure at the front vent. Whilst in some circumstances this may be beneficial to introduce more cooler air into an enclosure, here this backflow merely recirculates in the lower front section beyond the partition and does not interact to any great extent with the remainder of the enclosure. Its main contribution is to effectively reduce the open area of the vent which is used for exhausting the airflow from within, and hence be detrimental to the efficiency of the ventilation.

5.3.4 Enclosure Ventilation Performance

The physical limitations placed by aircraft wing design on the leading edge leads to very restricted ventilation options when trying to exhaust excessive heat from such enclosures. The investigation presented here relies upon the ventilation openings available on the lower leading edge surface, where drainage holes are located in the actual leading edge structure. Thus far, both the inlet and outlet vents are kept at equal open areas, to simplify the ventilation regime. It is of interest to designers to investigate the impact of the change in the open
CHAPTER 5. LEADING EDGE VENTILATION

Figure 5.23: Airflow crossing exit vent boundary for Case A (a), Case B (b), Case C (c), Case D (d) Case E (e).
area of one vent relative to the other as to whether this has a positive or negative influence on the overall ventilation regime in order to produce a more optimised vent configuration. As a result, the work already outlined is expanded to include this effect and is presented in the following section.

The ventilation performance within the leading edge is presented for the varying enclosure configurations presented in Section 5.3.1, namely the empty, unpartitioned and partitioned leading edge. In order to determine the effect of ventilation configuration upon the leading edge, a vent ratio

\[ R^* = \frac{A_{\text{Front}} - A_{\text{Back}}}{A_{\text{Total}}} \]  \hspace{1cm} (5.1)

is defined, where \( A_{\text{Front}}, A_{\text{Back}} \) and \( A_{\text{Total}} \) are the percentage open area of the front, back and total vent openings in the enclosure respectively. Varying this parameter leads to a sole vent at the rear of the enclosure at \( R^* = -1 \), both vents equally open at \( R^* = 0 \) and only open at the front of the enclosure at \( R^* = 1 \).

The average enclosure \( T^* \) is presented for a varying \( R^* \) under the three different enclosure configurations in Fig. 5.24. The temperature distributions exhibit a general trend whereby the lowest enclosure temperatures are recorded at \( R^* = 0 \), where both vent openings are of similar size at the front and rear of the enclosure. \( T^* \) increases when moving

---

Figure 5.24: Variation in interior air \( T^* \) with \( R^* \) for the Case A (♦), Case B (◦) Case C (□).
CHAPTER 5. LEADING EDGE VENTILATION

towards a rear \(R^* = -1\) or forward \(R^* = 1\) biased venting scenario. The empty enclosure is, as expected, at the lowest overall enclosure temperature. A greater reduction in \(T^*\) is found for going towards a rear biased ventilation \((R^* < 0)\) compared to forward biased, with \(T^*\) being 0.035 at \(R^* = -1\) compared to 0.065 at \(R^* = 1\) for this case. This suggests that having the vent openings biased towards the rear of the enclosure is more beneficial both for a single vent \((R^* = -1)\) and two vent openings of different magnitude \((R^* = -0.6)\) without the presence of a bleed duct. A possible reason for this is that the reduction in the size of the outlet vent at \(R^* = -0.6\) is not sufficient to inhibit the mass flow of air exhausted from the leading edge and as a result \(T^*\) does not increase. Reducing the open area of the rear vent in favour of the front one at \(R^* = 0.6\) does produce an increase in \(T^*\) as a result of a drop in the amount of cooler exterior entrained into the leading edge. This highlights that the interior air temperature is more sensitive to the size of the inlet (rear) vent than the front in a dual vent configuration.

The introduction of the constant temperature bleed duct increases the average \(T^*\) to 0.22 (at \(R^* = 0\)), with the partitioned enclosure slightly lower at 0.2. The vented enclosure with the bleed duct (Case B) shows an almost symmetrical profile, with a minimum \(T^*\) at \(R^* = 0\). A slight increase in \(T^*\) is found for moving towards front biased venting with \(T^* = 0.265\) & 0.28 for \(R^* = -1\) & 1 respectively. For the partitioned case, similar to the empty enclosure in Case A, an increase in \(T^*\) is noted for forward biased venting. Here this change is more pronounced than for Case A, with a large rise in \(T^*\) at \(R^* = 0.6\) which then remains at the same \(T^*\) value of 0.25 at \(R^* = 1\). For \(R^* > 0\), the larger vent opening is beyond the position of the partition, which inhibits the flow of air between the inlet and outlet vents, so having a larger exit vent beyond the partition is of no benefit whatsoever with regards to reducing the interior air temperature. Again it is seen that for the partitioned case, similar to Case A, the interior air temperature is more sensitive to the size of the rear inlet vent than the exit vent beyond the partition. The increase in \(T^*\) from \(R^* = 0.6\) to 1 seen in Case A & B is not present in Case C when the partition is introduced. The average enclosure temperature remains approximately constant in the enclosure when any forward biased ventilation scheme is used with the presence of a partition. At \(R^* = 0.6\), the average temperature is comparable to that of a single vent only configuration \((R^* = 1)\) and
a twin vent configuration seems to provide no benefit as regards to reducing the internal air temperature.

Comparing the average enclosure temperature to the temperature of air exiting at the front vent of the enclosure ($T_A$) in Fig. 5.25 shows an increase in this parameter occurs when going towards a front biased venting configuration. An increase in $T_A$ suggests that either an increase in the temperature of the air being exhausted is occurring or a drop in the average interior air temperature (or both as one will come about as a result of the other occurring). One or both of these effects are beneficial from a ventilation perspective as indeed the overall goal of introducing ventilation was to reduce the interior air temperature via the exhausting of high temperature air from the enclosure. This is present in all three cases investigated. $T_A$ then drops off significantly when only a single vent is open at the front of the enclosure ($R^* = 1$). Here, the single vent opening at the front suffers from a degree of mixing of the entrained and exhausted air occurring which reduces the temperature of the air exhausted from the enclosure, leading to a drop in $T_A$. This is further compounded by the single vent opening being located away from the bleed duct (which was found to augment the entrainment of cooler exterior air into the leading edge when placed directly above the inlet vent) leading to an increase in the average air temperature in the leading edge, reducing $T_A$ further.

In general, it is seen that in a dual vent configuration it is beneficial for $T_A$ to move the
ventilation strategy towards a forward biased configuration. As $R^*$ increases, the venting of the leading edge is biased towards the higher front vent. In a standard displacement ventilation scenario, increasing the height of the exit vent increases the amount of the hotter, more buoyant air that can be easily removed from an enclosure. For $R^* > 0$, the ventilation in the leading edge is based upon a larger vent opening at the outlet than the inlet (which is also at a greater height in the enclosure). This is seen to increase $T_A$. Even though the height difference between the vents is small compared to the overall height of the leading edge, it appears to be enough to produce the increase in $T_A$ seen in Fig. 5.25. An optimum value of $T_A$ exists when $R^* = 0.6$, compared to the optimal $T^*$ at $R^* = 0$ as seen in Fig. 5.24.

The temperature rise ($T_R$) taken between the inlet and outlet vent is presented in Fig. 5.26. From a ventilation perspective, it is beneficial to have a large $T_R$ value as it indicates that the air exiting the leading edge has increased in temperature before being exhausted as a result of the interaction with the air within the leading edge. This produces a reduction in the interior air temperatures as a result of this efficient air removal. A vent $R^*$ ratio of 0 provides an optimum value for $T_R$ for all cases considered, particularly Case B. Similar to $T_A$, an increase in $T_R$ is found for moving towards a forward biased ventilation configuration as opposed to rearward. This is particularly evident for the empty leading edge without the bleed duct (Case A) where $T_R$ does not change significantly between $R^* = 0 - 0.6$. Case
B also shown an increase in $T_R$ for forward biased venting compared to rearwards, however $T_R$ is still at a maximum at $R^* = 0$. The partitioned enclosure in Case C shows no benefit in moving towards either a front or rear biased venting configuration. In this case it is at a maximum when the two vent openings in the leading edge are the same size. For the other configurations, although there is a maximum temperature rise present at $R^* = 0$, some benefit can be gained by placing the vent opening towards the front of the enclosure as opposed to the rear, with $T_R$ greater at $R^* = 0.6$ compared to $R^* = -0.6$ for both the empty enclosure (Case A) and with the horizontal bleed duct (Case B).

The overall enclosure ventilation efficiencies are presented in Fig. 5.27. The greatest ventilation efficiency is found for the unpopulated leading edge. Although the presence of the bleed duct adjacent to the inlet vent opening has been found to increase the amount of entrainment of cooler exterior air, the additional heat load from the bleed duct itself means a higher overall air temperature in the leading edge which reduces the overall ventilation efficiency in this case. The empty leading edge (Case A) and bleed duct only (Case B) configurations also show a similar change in $\varepsilon$ with $R^*$, with an increase in ventilation possible for $R^* \geq 0$, a similar trend to which was also noted for $T_A$ and $T_R$. The curved shape of the enclosure means that the front vent is higher than the rear vent, and this is generally perceived to be advantageous with regards to displacement ventilation performance.
Due to the shape of the leading edge and the placement of ventilation holes, there is not a large difference in the vent heights with regard to the overall enclosure height. Even with this limitation, it can be seen that having the ventilation regime biased towards the front, slightly higher vent tends to increase the ventilation performance compared to having the lower rear vent dominate.

The introduction of the partition in Case C results in a further drop in ventilation efficiency ($\varepsilon$), and also alters the relationship between $\varepsilon$ and $R^\star$. Here, any deviation away from $R^\star = 0$ results in a sharp drop in $\varepsilon$ (60% decrease at $R^\star = -0.6$ & 0.6), which would be detrimental for cooling of components within the leading edge. This also highlights that the introduction of a partition causes a change in the optimum ventilation strategy for the leading edge, with an unpartitioned configuration benefiting from a forward biased ventilation solution, and the partitioned case showing maximum ventilation efficiency at $R^\star = 0$. This change in optimum position implies that the influence of partitioning cannot be underestimated when designing the ventilation within an enclosure.

### 5.3.5 Mass Flow and Air Exchange Rate

A prime objective of ventilation is to remove excessive heat from an enclosure. The idea of venting the leading edge is to reduce the overall interior temperature by exhausting high temperature air through the vent openings. Section 5.3.1 & 5.3.2 show how the temperature distribution within the leading edge changes as a result of the presence of the bleed duct, partitioning the leading edge and also bleed duct positioning. Thermal confinement and the separation distance between the bleed duct and the inlet vent have also been shown to be influential on the thermal distribution. Whilst the temperature profile within the leading edge is a good indicator of the change in the interior conditions as a result of these geometric changes, another such metric of change is the amount of heat exhausted from the leading edge through the vents. This is presented in the following section, based upon the enthalpy change between the inlet and outlet vents.

Fig. 5.28 shows the airflow profiles taken normal to the exit vent boundary. These are taken at $R^\star = 0$. Here the change in the amount of interior air exhausted from the enclosure depending upon the geometric conditions is evident. Evident from this figure is that when
the partition is introduced to the leading edge, the front enclosure vent no longer acts as a pure exhaust vent. In these cases, up to 1/3 of the vent open area is subjected to some degree of backflow into the leading edge, reducing the effective size of the vent for air removal. Partitioning also reduces the total airflow leaving the leading edge. \( U_N \) drops from a maximum of 0.057 m/s for the unpartitioned bleed duct case to 0.037 m/s. This contributes to the drop in ventilation efficiency seen for the partitioned case in Fig. 5.27 For the unpartitioned cases, \( U_N \) is positive at all points across the vent boundary. The empty enclosure in Case A, whilst exhibiting a lower exit flow rate at a lower temperature, still produces the highest ventilation efficiency in Fig. 5.27. This is as a result of the absence of the horizontal bleed duct in the enclosure, with the overall enclosure temperature being significantly lower, \( T^* = 0.01 \) compared to 0.22 for Case B. This is further reflected in \( Q_{EX} \) being lower for Case A than Case B in Table 5.2. Little difference is noted between Case C & D across the vent boundary and this is reflected in the mass flow rates presented in Table 5.2. The bleed duct positioned in the upper left of the enclosure has the lowest mass flow rate out of the enclosure, which is 62% lower than the other partitioned cases and 85% lower than the maximum value recorded in Case B.

The heat exhausted from the cavity varies significantly between the cases considered and highlights the influence of geometrical effects on the ventilation within a non-standard geometry. The introduction of the partition from Case B to Case C produces a 60% drop
Table 5.2: Mass flow of air removed and heat exhausted from the leading edge for Cases A - E

<table>
<thead>
<tr>
<th>Case</th>
<th>(m) (kg/s)</th>
<th>(Q_{EX}) (W)</th>
<th>ACH (1/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>(9.53 \times 10^{-4})</td>
<td>24.9</td>
<td>40.5</td>
</tr>
<tr>
<td>B</td>
<td>(1.25 \times 10^{-3})</td>
<td>41.4</td>
<td>53.2</td>
</tr>
<tr>
<td>C</td>
<td>(5.04 \times 10^{-4})</td>
<td>16.0</td>
<td>21.4</td>
</tr>
<tr>
<td>D</td>
<td>(4.91 \times 10^{-4})</td>
<td>15.9</td>
<td>20.9</td>
</tr>
<tr>
<td>E</td>
<td>(1.89 \times 10^{-4})</td>
<td>6.11</td>
<td>8</td>
</tr>
</tbody>
</table>

in the amount of heat transported out of the enclosure resulting in a drop in ventilation efficiency for this case. Moving the bleed duct towards the partitioned surface in Case D shows little difference in the heat rejection. For this case though the interior air temperatures are higher than for Case C, which is not desirable for the given enclosure conditions. \(Q_{EX}\) is substantially lower for the bleed duct located in the upper region of the enclosure. For this case although the temperatures recorded along the enclosure centrelines are lowest for most of the enclosure height \(Y^* < 0.7\) which would appear beneficial, this is masked by the confinement of the high temperature fluid to the upper region of the enclosure. This, combined with the drop in mass flow rate across the vent boundary contributes to this sharp drop in \(Q_{EX}\) and is not an optimal placement for the bleed duct within the leading edge. The air changes per hour [ACH] (calculated by \(q = nV\), where \(q\) fresh air supply rate \([m^3/h]\), \(V\) is the enclosure volume \([m^3]\) and \(n\) is the air change rate \([1/h]\)) also shows a substantial decrease in the number of times the air is refreshed in the leading edge per hour as a result of the introduction of the partition. This also assumes perfect mixing of the ventilation flow with the interior air which has been shown to not be the case for the leading edge in the current ventilation configuration, so the mean air age (particularly in the upper region of the leading edge where thermal confinement occurs) is likely to be much greater.

5.4 Conclusions

The impact of introducing multiple ventilation openings to the leading edge upon the internal thermal distribution and bleed duct heat transfer has been presented in this chapter. It was found that a reduction in the interior air temperature was possible due to ventilation, except for the single vent on the lower surface. In this configuration the resultant enclosure conditions were almost identical to an unventilated scenario due to the choking
of the ventilation flow at the vent boundary. A dual vent configuration with a distinct inlet and outlet opening performed best, with a maximum reduction in the interior temperatures seen when openings are placed on the upper and lower surface simultaneously. Geometrical constraints imposed by the leading edge construction lead to reduction in ventilation performance, with some reduction in internal air temperature still possible.

For two lower vent openings configuration, the enclosure ventilation consists of both a displacement and mixing component. The lower surface is affected by displacement between the inlet and outlet vent, although this is severely restricted by the minimal height difference between the two openings. In the region above this, mixing of the entrained air and the core bulk air of the enclosure occurs and this region is dominated by the influence of the bleed duct plume, particular at the higher bleed duct input powers. This tends to make the upper region more isothermal as a result and also has the same effect on the rear wall of the enclosure as heat transfer to it increases and the cooling effect from ventilation diminishes. Ventilation of the enclosure does lead to a decrease in the bleed duct surface temperature and a simultaneous drop in the interior air temperature leading to a lower $\Delta T$ between them. Heat transfer is seen to increase as a result, with a 77% rise in $Nu_{BD}$ reported as a result.

Partitioning of the enclosure (introduced to replicate a subspar in the leading edge assembly) tended to impair both the ventilation of the enclosure (by producing a block in the ventilation path between the inlet and outlet vent) and bleed duct heat transfer as a result. It also affected the interaction of the bleed duct plume with the remainder of the leading edge and the resultant flow structure, as it attached to the rear wall and was highly damped as a result compared to the unpartitioned configuration. This reduced the mixing effect of the plume considerably, leading to slightly lower air temperatures in the centre of the leading edge, but with increased confinement of this high temperature plume flow to the upper region of the enclosure. A reduction in convective heat transfer from the bleed duct is noted as a result, with only $Nu_{BD}$ only 44% higher than the unventilated leading edge.

Some optimisation of the ventilation performance is possible based upon both the bleed duct location (relative to the inlet vent) and size of the relative vent openings $(R^* \text{ ratio})$. This is dependant upon geometric conditions, particularly the presence of the partition, with
the optimal ventilation configuration changing as a result. For the unpartitioned enclosure, some benefit is obtained from moving to a forward biased ventilation strategy (based upon ventilation efficiency). This is not seen for the partitioned configuration, as it produces a drop in the overall ventilation efficiency, even more so when departing from equal inlet and outlet vent size ($R^* \neq 0$). The ventilation of the leading edge (and resultant interior temperature distribution and flowfield) is highly dependant upon the geometric configuration and care must be taken that adequate knowledge of the thermal profile and flow conditions is obtained before any ventilation strategy can be chosen.
Chapter 6

Conclusions & Recommendations

The objective of this thesis was to improve the understanding of the thermal distribution and heat transfer within an aircraft wing leading edge enclosure subjected to external solar loading and with the presence of a hot internal bleed duct. To achieve this a comprehensive analysis of the temperature profile within the leading edge was experimentally obtained along with the influences of enclosure ventilation, partitioning and location of the bleed duct. The conclusions obtained are presented here. Recommendations for further investigation are then noted.

6.1 Conclusions

6.1.1 Sealed Leading Edge

- An analysis of the heat transfer from the bleed duct has shown that enclosure effects are dependent upon the Grashof number. At the low Grashof range investigated, the behaviour of the bleed duct can be characterised by a typical correlation for an unconfined cylinder. As the Grashof number of the bleed duct increases, its behaviour begins to resemble that of a confined cylinder.

- The temperature distribution in leading edge is also conditional to the bleed duct Grashof number. For low Grashof numbers, a change in $T^*$ with height is evident. As the relative strength of the plume increases with $Gr$ (along with its mixing effect),

145
the upper region of the leading edge becomes more isothermal, with only the lower region of the leading edge exhibiting any noticeable change in $T^\star$ with height.

- PIV images of the airflow around the bleed duct revealed a change in the flow structure with $Gr$. At low bleed duct $Gr$, all of the air in the region below is entrained by the bleed duct. As $Gr$ increases (along with $T_{BD}$) the presence of a secondary flow structure on the lower portion of the vertical wall due to an increase in its surface temperature becomes evident. This entrains a portion of the air which was previously entrained by the bleed duct. This is found to bypasses the bleed duct completely at the highest $Gr$ value investigated.

- In a multiple cylinder configuration (with a larger constant temperature cylinder present with the bleed duct at $s/D = 2$) no significant influence is noted when the constant temperature cylinder is placed above the bleed duct. Heat transfer is almost identical to the isolated bleed duct in this configuration. This agrees with literature for free arrays of cylinders where the lowest cylinder is unaffected by the presence of higher cylinders. No additional enclosure effect is noted compared to the bleed duct only configuration.

- For the bleed duct placed above the constant temperature cylinder, the effect of confinement from the enclosure and the heating effect from the cylinder below significantly reduces heat transfer. When the lower cylinder is at a higher surface temperature than the bleed duct, convective heat transfer is effectively eliminated ($Nu \sim 1$). When the bleed duct surface temperature increases beyond that of the lower cylinder, convective heat transfer increases, but is still only reaches a maximum of 50% of $Nu_{BD}$ compared to the bleed duct only configuration at $Gr = 3.5 \times 10^5$.

### 6.1.2 Vented Leading Edge

- Location of the vent openings determined the degree of reduction in the interior air temperature, with a greater reduction in the interior air temperature possible with a displacement ventilation configuration. A single vent opening on the lower surface was found to be totally ineffective, with no difference to a unventilated enclosure
noted. The greatest reduction in interior temperature occurred for separate openings on the upper and lower surfaces.

- The limitation of the vent opening positions to the lower surface for operational and structural reasons reduced the effectiveness of the ventilation, however an enhancement in the bleed duct heat transfer was still achievable. Bleed duct heat transfer was found to increase by 77% after the introduction of this ventilation configuration. The bleed duct heat transfer coefficient was also found to be constant across the Gr range investigated.

- Ventilation of the leading edge was observed to be sensitive to the relative open area of the inlet and outlet vents. An increase in ventilation efficiency was noted for the unpartitioned enclosure when the vent openings are biased towards the front of the leading edge. For the partitioned leading edge, ventilation efficiency was found to decrease significantly when moving to a front or rear biased venting configuration. Care must also be taken with regards the choice of ventilation metric when optimising the flow, as a different optimisation may seem apparent to the designer depending upon the amount of information available.

- The presence of the partition created a block in the path between the inlet and outlet vents which led to a reduction in the exterior air entrained and an increase in the bleed duct surface temperature. This led to a reduction in the bleed duct heat transfer by 19%. This was still an improvement of 44% compared to the bleed duct within the unventilated leading edge however.

- The position of the bleed duct in the partitioned leading edge also had a significant influence upon the resultant interior thermal distribution, both as a result of the change in the flow structure and the bleed duct to inlet vent distance. Thermal confinement was also significant when the bleed duct was placed in the upper region of the leading edge.

- Increasing the distance from the bleed duct to the inlet vent increased the interior air temperature due to a reduction in the amount of cooler exterior air entrained into the
leading edge. This led to a reduction in the heat exhausted from the leading edge and also the number of air changes per hour (ACH). Mass flow rate of air exiting the leading edge reduced by 60% when the partition was introduced and by 85% when the bleed duct was positioned in the upper corner of the leading edge.

- An investigation into the flowfield using PIV revealed the bleed duct as the main driver of the flow structure present. Moving the bleed duct had the greatest impact upon the interior flowfield. The proximity of a confining enclosure wall (exterior wall or internal partition) also affected the plume dynamics. An enclosure effect was also noted when the partition was introduced a distance which effectively changed the aspect ratio of the enclosure in which the bleed duct is located. It caused the oscillations of the bleed duct plume to reduce and to attach to the rear vertical wall of the leading edge which was not seen in the unpartitioned configuration.

- Control of the bleed duct plume flow path and interaction with the interior of the leading edge will allow for optimisation of the removal of high temperature air from the enclosure. This is especially true in the partitioned case where the air is required to be exhausted through the front vent opening. In order to reach this opening it first must travel through the narrow gap between the top of the partition and the upper enclosure surface. If it does not, then the recirculation of the plume causes the interior air temperatures increase with a reduction in the ventilation efficiency.

### 6.2 Recommendations

- The experimental test facility used presents an isolated leading edge enclosure which is removed from any external influences from the remainder of the wing structure. This was necessary to understand the fundamental behaviour within the leading edge. The presence of a fuel tank in the region behind the leading edge has the possibility to greatly influence the conditions within the leading edge, especially when a variable fuel load is possible. An investigation into this would provide significant insight into the interaction between the two adjacent compartments.
• A somewhat simplified ventilation strategy was presented in this thesis which used ventilation openings which ran the entire depth of the leading edge. In practical applications this is not a feasible solution and a more realistic configuration suitable for an actual aircraft use could be investigated. This may necessitate a three-dimensional study of the leading edge.

• The conditions imposed upon the leading edge represent worse case scenario conditions, for a high temperature surroundings with no ambient airflow. An investigation into the influence of a local wind upon the ventilation would reveal any enhancements (or otherwise) present as a result. This would be very useful from an aircraft operations perspective.

• With the current drive in aviation design to reduce weight to increase efficiency, a significant proportion of the aircraft structure is now manufactured from composite material. This has markedly different thermal properties to that of aluminium. The influence of these materials from a thermal perspective is often not considered as much as their structural and weight influences, so an investigation into the leading edge constructed of a composite material or a mixture of aluminium and composite would be beneficial.

• The temperature difference across the exterior of the leading edge was also presented here for one value only (30°C). The change of temperature difference across a cavity has been noted to produce a significantly different flow structure [18]. An investigation into a change the parameter will also further the knowledge of the heat transfer regime within the leading edge.

• Enhanced flow control (especially of bleed duct plume) using internal geometry may be possible and give further insight into an optimisation of the ventilation regime where required.
Bibliography


BIBLIOGRAPHY


BIBLIOGRAPHY


BIBLIOGRAPHY


Appendix A

Published Work

Journal Papers


• Moore D., Newport D., Egan V., Lacarac V., “Bleed duct influence on heat transfer in a leading edge aircraft compartment” Aerospace Science & Technology, (under review)

• Moore D., Egan V., Newport D., Lacarac V., “Numerical investigation of the thermal distribution and flow structure within a vented wing leading edge enclosure,” CEAS Aeronautical Journal, (under review)

Conference Papers


Appendix B

List of Equipment

Heaters

- ELEMEX 235mm x 570mm Heating Mat, Serial No. 59288, 300W
- ELEMEX 155mm x 570mm Heating Mat, Serial No. 23644, 250W
- ELEMEX 600mm x 590mm Heating Mat, Serial No. 21260, 500W
- ELEMEX 300mm x 590mm Heating Mat, Serial No. 32654, 500W
- ELEMEX Ø20mm x 540mm Cartridge Heater, 1000W
- Eurotherm 2216e PID Controller Serial No. IR19033
- Variac Claritronic 10534 Serial No. 701112007

Thermocouple Calibration Bath

- Laude Calibration Bath, RM-6, Serial No. 95-0094

Thermocouple Readers

- Stanford Research Systems SR630 Datalogger, Serial No. 58313
- Agilent 34970A Datalogger, Serial No. MY41020168
- Agilent 34970A Datalogger, Serial No. MY44063501
- HP Benchlink Data Logger 3 (Version 4.3.00) Software

PIV System

- Litron Lasers Laserhead Nano L50-1000, Serial No. L170401
• Laser Power Supply LPU-1000, Serial No. 90021/0125 (Laser 1) 90021/0126 (Laser 2)

• TSI Laser Pulse Synchroniser 610035, Serial No. 708113268

• Litron Laser Remote, Serial No. 90015/0169 (Laser 1) 90015/0170 (Laser 2)

• ISEI Automation Traverse, Serial No. 333560

• ISEI Schrittmotor Controller, C142-4 Serial No. 70834010

• Powerview 2MP Plus CCD Camera, Model No 510060. Serial No M52239

• TSI Insight 3G Ver. 9.0.3 Processing Software
## Appendix C

### Leading Edge Geometry Data Points

Table C.1: Data points for the leading edge geometry supplied by Airbus. Dimensions in mm.

<table>
<thead>
<tr>
<th>x</th>
<th>y</th>
<th>x</th>
<th>y</th>
</tr>
</thead>
<tbody>
<tr>
<td>(20, 0)</td>
<td>(536, 144)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(40, 0)</td>
<td>(530, 156)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(60, 0)</td>
<td>(520, 171)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(80, 0)</td>
<td>(500, 190)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(100, 0)</td>
<td>(480, 204)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(120, 0)</td>
<td>(460, 216)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(140, 0)</td>
<td>(440, 227.2)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(160, 0)</td>
<td>(420, 236)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(180, 0)</td>
<td>(400, 245.6)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(200, 0)</td>
<td>(380, 252.5)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(220, 0)</td>
<td>(360, 260)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(240, 0)</td>
<td>(340, 266)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(260, 0)</td>
<td>(320, 270.4)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(280, 0)</td>
<td>(300, 274)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(300, 0)</td>
<td>(280, 278)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(320, 0)</td>
<td>(260, 282)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(340, 0)</td>
<td>(240, 285)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(360, 2)</td>
<td>(220, 288)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(380, 4)</td>
<td>(200, 290.5)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(400, 7)</td>
<td>(180, 292)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(420, 12.5)</td>
<td>(160, 294)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(440, 17)</td>
<td>(140, 295.6)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(460, 24.5)</td>
<td>(120, 296.4)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(480, 33)</td>
<td>(100, 297)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(500, 43)</td>
<td>(80, 298)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(512, 52)</td>
<td>(60, 298.2)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(520, 60)</td>
<td>(40, 298.6)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(530, 74)</td>
<td>(20, 299)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(534, 82)</td>
<td>(0, 300)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(538, 94)</td>
<td>(0, 240)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(540, 104)</td>
<td>(0, 180)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(545, 110)</td>
<td>(0, 120)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(550, 118)</td>
<td>(0, 60)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(540, 128)</td>
<td>(0, 0)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

C1