Copyright Statement

The digital copy of this thesis is protected by the Copyright Act 1994 (New Zealand).

This thesis may be consulted by you, provided you comply with the provisions of the Act and the following conditions of use:

- Any use you make of these documents or images must be for research or private study purposes only, and you may not make them available to any other person.
- Authors control the copyright of their thesis. You will recognize the author's right to be identified as the author of this thesis, and due acknowledgement will be made to the author where appropriate.
- You will obtain the author's permission before publishing any material from their thesis.

General copyright and disclaimer

In addition to the above conditions, authors give their consent for the digital copy of their work to be used subject to the conditions specified on the Library Thesis Consent Form and Deposit Licence.
INVESTIGATION OF HEAT RECOVERY AND COMBUSTION CHARACTERISTICS IN SMALL SCALE COMBUSTION CHAMBERS

by

Valerio Giovannoni

Auckland, New Zealand

November 2017

Supervised by

Dr. Rajnish N. Sharma

and

Associate Professor Robert R. Raine

A thesis submitted in fulfilment of the requirements for the degree of Doctor of Philosophy in Mechanical Engineering, The University of Auckland, 2017
Abstract

Over the last two decades the development of portable devices such as smartphones, laptops, biomedical diagnostic sensors, and GPS (Global Positioning Systems) receivers has led to a miniaturization of their components and to a higher demand of electricity supply that accumulators and batteries cannot fulfil anymore, although they are still widely used today. The most suitable and promising devices to provide electric power to the above mentioned small and portable applications seem to be Ultra-Micro Gas Turbines (UMGTs).

The current research focuses on investigating the causes of efficiency decrease due to heat losses from the hot components and proposes a regenerative combustion chamber to improve the thermal exchanges.

The combustion chamber has been designed in a way that the cold reactants flow in a helicoidal channel which surrounds the inner cylinder where combustion stabilized on a porous medium occurs. Combustion products are then recirculated through a second outer helicoidal channel so that the unburnt mixture receives heat from both sides. Comprehensive tests were performed on the combustor to measure thermal performances and pollutant emissions. Finally the feasibility of using vegetable oils as fuel is investigated and experimental results exposed.

A numerical study has also been performed on a similar model of combustion chamber to provide a further insight of combustion phenomena and to investigate the effects of including a porous medium as flame holder on heat transfer mechanisms. Several simulations using a detailed skeletal mechanism for methane oxidation were run considering different mass flow rates, equivalence ratios and initial reactants’ temperature.

Finally, as a complement of the present research and to provide a better understanding on heat transfer mechanisms in regenerative combustion chambers, another numerical study was performed on a three concentric tubes system with combustion occurring in the inner cylinder. Cold reactants and combustion products were set to flow in the middle and outer channels respectively, in a combined counter-current arrangement. Different mass flow rates were considered and also a parametric study with varying wall thermal conductivities was performed to highlight the influence of axial conduction through walls.
Dedicated to my family
Acknowledgements

I would like to thank my two supervisors Dr. Rajnish N. Sharma and Associate Prof. Robert R. Raine, who constantly supported me and guided me on the right track throughout this study. Their constructive critics and attention to details have always been an inspiration and a powerful motivation.

I am also very thankful to Alan Eaton and Martin Ryder who have given an essential help during my experimental work in the laboratory. I truly appreciated their patience and professionalism during the several modifications to the experimental rig. I would not have been able to complete my study without their support.

I would like to thank Associate Professor K.C. Aw for letting me use his flow controller and the technicians from the workshop at the University of Auckland, in particular Paul Tomlin and Stephen Warrington for building and machining my combustion chamber and the other items.

I also am very grateful to the Energy and Fuels Research Unit (EFRU) for the scholarships received to support my study.

During my studies I am glad to have shared my Ph.D. journey with great people such as Pititat Itsariyapinyo, Pawan Singh, Rohann D’Souza, Ralph Gallegos, Christoph Steinberg, Amir Ali Safaei Pirooz, Frank Chen, Stefano Nava, and Jasnoor Singh. Special thanks to Dr. Zerrin Turkeli-Ramadan for her initial UMGT research at the University of Auckland which provided a starting point for my research.

Special thanks also to Miqdad Hassan, good friend and always a great source of inspiration.

Finally I would like to thank my family and friends who always supported me from home. Without them I would not have achieved so much in my life.
Table of Contents

Abstract ............................................................................................................................... iii
Acknowledgements ................................................................................................................. vi
Table of Contents .................................................................................................................. vii
List of Publications .............................................................................................................. xi
List of Figures ....................................................................................................................... xii
List of Tables ......................................................................................................................... xx
Nomenclature ....................................................................................................................... xxi

1 INTRODUCTION ........................................................................................................... 1
1.1 Scaling Considerations ................................................................................................. 5
1.2 Thesis Aim ..................................................................................................................... 10
1.3 Objectives ..................................................................................................................... 10
1.4 Thesis outline ............................................................................................................... 12

2 LITERATURE REVIEW ................................................................................................. 14
2.1 The UMGT at M.I.T. ................................................................................................. 14
2.2 The project at IHI Corp. ............................................................................................ 15
2.3 The flat-flame combustor at Tokyo Metropolitan University ...................................... 16
2.4 The Belgian PowerMEMS project ............................................................................. 16
2.5 The UMGT coupled with a piezo converter ................................................................ 18
2.6 The UMGT at the Korea Institute of Machinery & Materials ................................. 18
2.7 The Research at the University of Auckland ............................................................. 19
2.8 The UMGT at Onera ................................................................................................. 23
2.9 The UMGT at the University of Rome “La Sapienza” ............................................. 24
2.10 The “Swiss roll” ....................................................................................................... 26
2.11 Lean Premixed Pre-vaporized Combustion for Liquid Fuels .................................. 27
2.12 Premixed combustion using liquid fuels ................................................................. 28
   2.12.1 Liquid fuel combustion using a porous medium ................................................. 28
   2.12.2 Flow blurring injector applied to annular combustor ....................................... 29
2.13 Fuel-film liquid combustion for a TPV power system ............................................ 34
2.14 A Micro Gas Turbine fuelled with vegetable oils .................................................... 35
2.15 Laminar combustion ................................................................................................. 36
2.16 Summary .................................................................................................................... 40

3 UMGT AND EXPERIMENTAL MICRO COMBUSTOR DESIGN ......................... 41
3.1 Brayton Cycle ............................................................................................................. 41
  3.1.1 Improvements to the Brayton cycle ................................................................. 44
3.2 Constraints .................................................................................................................. 48
  3.2.1 Materials ............................................................................................................. 49
3.3 UMGT design parameters ....................................................................................... 51
  3.3.1 Parametric study of the thermal cycle ............................................................. 51
3.4 Combustion chamber ............................................................................................... 60
3.5 Porous medium and flow analysis .......................................................................... 62
3.6 Summary .................................................................................................................... 70

4 EXPERIMENTAL METHODOLOGY ........................................................................... 71
  4.1 Regenerative combustion chamber ...................................................................... 71
  4.2 Porous flame holder ............................................................................................... 73
  4.3 Flow blurring injector ............................................................................................ 74
  4.4 Mass flow controllers ............................................................................................. 77
  4.5 Thermocouples ....................................................................................................... 78
  4.6 Ignition device ......................................................................................................... 80
  4.7 Flame observation ................................................................................................... 81
  4.8 Peristaltic pump ...................................................................................................... 82
  4.9 Exhaust gases analysis ............................................................................................ 83
  4.10 Thermal imaging.................................................................................................... 85
  4.11 Assembled rig ........................................................................................................ 89
  4.12 Insulating materials ............................................................................................. 91

5 EXPERIMENTAL RESULTS ...................................................................................... 94
  5.1 Non-insulated case ................................................................................................. 94
    5.1.1 Combustion products composition ................................................................. 94
    5.1.2 Equivalence ratio and combustion efficiency ............................................... 99
    5.1.3 Visual observation of the flame ....................................................................... 104
    5.1.4 Axial distribution of temperature ................................................................... 110
    5.1.5 Temperature measurements ........................................................................... 113
    5.1.6 Thermal imaging ............................................................................................ 118
  5.2 Insulated case ......................................................................................................... 121
    5.2.1 Visual observation of the flame ....................................................................... 122
    5.2.2 Axial distribution of temperature ................................................................... 127
    5.2.3 Temperature measurements ........................................................................... 130
  5.3 Comparison ............................................................................................................. 132
<table>
<thead>
<tr>
<th>Chapter</th>
<th>Section</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.4</td>
<td>5.4.1</td>
<td>Prediction of droplet evaporation</td>
<td>137</td>
</tr>
<tr>
<td>5.4</td>
<td>5.4.2</td>
<td>Operating parameters</td>
<td>140</td>
</tr>
<tr>
<td>5.4</td>
<td>5.4.3</td>
<td>Composition of combustion products</td>
<td>142</td>
</tr>
<tr>
<td>5.4</td>
<td>5.4.4</td>
<td>Visual observation of the flame</td>
<td>143</td>
</tr>
<tr>
<td>5.4</td>
<td>5.4.5</td>
<td>Profiles of temperature</td>
<td>144</td>
</tr>
<tr>
<td>5.4</td>
<td>5.4.6</td>
<td>Clogging issues</td>
<td>145</td>
</tr>
<tr>
<td>5.5</td>
<td></td>
<td>Summary</td>
<td>146</td>
</tr>
<tr>
<td>6</td>
<td></td>
<td>NUMERICAL STUDY ON COMBUSTION STABILIZED ON A POROUS MEDIUM</td>
<td>147</td>
</tr>
<tr>
<td>6.1</td>
<td></td>
<td>Introduction</td>
<td>148</td>
</tr>
<tr>
<td>6.2</td>
<td></td>
<td>Numerical model</td>
<td>149</td>
</tr>
<tr>
<td>6.3</td>
<td></td>
<td>Validation of the model</td>
<td>150</td>
</tr>
<tr>
<td>6.4</td>
<td></td>
<td>Results and discussion</td>
<td>151</td>
</tr>
<tr>
<td>6.4.1</td>
<td></td>
<td>Adiabatic case</td>
<td>152</td>
</tr>
<tr>
<td>6.4.2</td>
<td></td>
<td>Non-adiabatic case</td>
<td>153</td>
</tr>
<tr>
<td>6.4.3</td>
<td></td>
<td>Effect of flame holder thermal conductivity</td>
<td>154</td>
</tr>
<tr>
<td>6.4.4</td>
<td></td>
<td>Effect of porosity of the flame holder</td>
<td>155</td>
</tr>
<tr>
<td>6.4.5</td>
<td></td>
<td>Effect of external temperature</td>
<td>156</td>
</tr>
<tr>
<td>6.5</td>
<td></td>
<td>Summary</td>
<td>157</td>
</tr>
<tr>
<td>7</td>
<td></td>
<td>NUMERICAL PREDICTION OF THERMAL PERFORMANCES IN A TRIPLE TUBE HEAT EXCHANGER</td>
<td>158</td>
</tr>
<tr>
<td>7.1</td>
<td></td>
<td>Introduction</td>
<td>159</td>
</tr>
<tr>
<td>7.2</td>
<td></td>
<td>Numerical model</td>
<td>160</td>
</tr>
<tr>
<td>7.3</td>
<td></td>
<td>Model validation</td>
<td>161</td>
</tr>
<tr>
<td>7.4</td>
<td></td>
<td>Results and discussion</td>
<td>162</td>
</tr>
<tr>
<td>7.4.1</td>
<td></td>
<td>Adiabatic case</td>
<td>163</td>
</tr>
<tr>
<td>7.4.2</td>
<td></td>
<td>Non-adiabatic case</td>
<td>164</td>
</tr>
<tr>
<td>7.5</td>
<td></td>
<td>Summary</td>
<td>165</td>
</tr>
<tr>
<td>8</td>
<td></td>
<td>CONCLUSIONS</td>
<td>166</td>
</tr>
</tbody>
</table>

Appendices

A1. Flow controllers working principle

A2. Flow rate correction factor

A3. Thermocouple working principle

A4. Thermocouple correction

A5. Thermocouples uncertainty analysis
A6. 17 species, 73 reactions skeletal mechanism ........................................240
A7. Technical drawings ..................................................................................243
References ......................................................................................................252
List of Publications

The following articles and conference papers have been published (or submitted to journals) at the time of the writing:

List of Figures

Figure 1 – Examples of GPS (left) and laptop (right) - Ref. [2,3] ............................................................ 1
Figure 2 – Examples of drone (left) and exoskeleton (right) - Ref. [4,5] ................................................ 2
Figure 3 – Example of drone for agricultural purposes - Ref. [6] .......................................................... 2
Figure 4 - Specific energy and applications [7] .................................................................................... 3
Figure 5 - Specific power vs specific energy of different power systems [8] ......................................... 3
Figure 6 - Exploded view of the M.I.T. turbine assembly .................................................................... 4
Figure 7 - Simple Brayton cycle [15] ................................................................................................... 5
Figure 8 - Surface to volume ratio of the volute vs volute inlet radius [19] ......................................... 8
Figure 9 - Dimensionless heat losses vs surface to volume ratio of the volute [19] ............................... 9
Figure 10 - Section of the M.I.T. UMGT [1] ....................................................................................... 14
Figure 11 - The UMGT of the Belgian PowerMEMS project [22] ......................................................... 17
Figure 12 - Turbine (left) and compressor (right) [22] ......................................................................... 17
Figure 13 - Scheme of the UMGT [23] .............................................................................................. 18
Figure 14 - Scheme of the UMGT [24] .............................................................................................. 19
Figure 15 - Triple tube HEX-combustor [12] .................................................................................... 20
Figure 16 - Pressure losses vs incoming flow velocity [12] ................................................................. 21
Figure 17 - Flat flame operating conditions [12] ............................................................................... 22
Figure 18 - Flammability limits of natural gas-air mixture [11] ............................................................ 22
Figure 19 - Emissions at air mass flow rate of 0.38 g/s [11] ................................................................ 23
Figure 20 - UMGT at ONERA [26] .................................................................................................. 23
Figure 21 - Compressor (left) and turbine (right) [28] ...................................................................... 25
Figure 22 - Regenerative combustion chamber [28] ........................................................................ 26
Figure 23 - Swiss roll burner [32] .................................................................................................... 27
Figure 24 - Heat recirculating burner [35] ......................................................................................... 29
Figure 25 - Flow blurring injector [31] ............................................................................................. 30
Figure 26 - Flow focusing injector [36] ............................................................................................. 31
Figure 27 - Flow blurring injector [36] ............................................................................................. 31
Figure 28 - Droplet distribution of diesel and vegetable oil at the spray and 20 mm axial distance [39] .... 32
Figure 29 - Combustor with porous media and heat exchanger [38] .................................................... 33
Figure 30 - Axially symmetric domain of the combustion chamber model [40] ................................. 34
Figure 31 - Schematic of experimental combustor [41] ..................................................................... 35
Figure 32 - Porous plug burner with flat flame .................................................................................. 37
Figure 33 - Heat transfer within a simple cylindrical combustor with flame holder ............................ 39
Figure 34 - Gas turbine scheme (open cycle) .................................................................................... 41
Figure 35 - Gas turbine scheme (closed cycle) .................................................................................. 42
Figure 36 - Brayton Cycle on T-S diagram ......................................................................................... 42
Figure 37 - Regenerative Brayton cycle ............................................................................................. 45
Figure 38 - Scheme of Brayton cycle with regeneration, intercooling and reheating................................. 47
Figure 39 - T-S diagram of the Brayton cycle with regeneration, intercooling and reheating...............................47
Figure 40 - Properties of selected materials [46]....................................................................................50
Figure 41 - Balje diagram for expanders [50].............................................................................................53
Figure 42 – Simple Brayton cycle efficiency vs pressure ratio ........................................................................54
Figure 43 – Exhaust gas temperature T4 and turbine inlet temperature T3 vs pressure ratio – 0.125 g/s flow rate ........................................................................................................................................54
Figure 44 – Predicted electric power output vs mass flow rate – 3.5 bar pressure ratio ........................................55
Figure 45 – Effect of regeneration on efficiency at different pressure ratios ......................................................55
Figure 46 – Effect of regeneration on fuel consumption at different pressure ratios – 0.125 g/s flow rate ............55
Figure 47 – Effect of regeneration on reactants’ temperature at different pressure ratios – 0.125 g/s flow rate ..55
Figure 48 - Effect of regeneration on fuel consumption at different pressure ratios - 1.5 g/s flow rate ...............55
Figure 49 - Legend ...........................................................................................................................................57
Figure 50 - Simple Brayton cycle - 2.5 bar, 1.5 g/s air mass flow rate .................................................................57
Figure 51 – Regenerative (R=0.2) Brayton cycle - 2.5 bar, 1.5 g/s air mass flow rate .........................................58
Figure 52 – Simple Brayton cycle - 3.5 bar, 1.5 g/s air mass flow rate ...............................................................58
Figure 53 – Regenerative (R=0.2) Brayton cycle - 3.5 bar, 1.5 g/s air mass flow .................................................59
Figure 54 – Scheme of the regenerative combustor (not to scale) ......................................................................61
Figure 55 - 3D model (left) – Mesh with 0.002m element size (right).................................................................63
Figure 56 – Contours of static pressure (0.5 m/s incoming velocity) ....................................................................65
Figure 57 – Contours of velocity (0.5 m/s incoming velocity) .............................................................................65
Figure 58 - Comparison of CFD results and experimental results [12] .............................................................65
Figure 59 - Pressure drop across PM resulting from the CFD simulations (10 mm PM thickness) .................66
Figure 60 - Fluid domain (PM 10mm thick) ....................................................................................................67
Figure 61 - Velocity profiles 3 mm below the PM - 7 mm thick PM ..................................................................68
Figure 62 - Velocity profiles 3 mm below PM - 8 mm thick PM .......................................................................68
Figure 63 - Velocity profiles 3 mm below the PM - 10 mm thick PM .................................................................69
Figure 64 - Pressure drops across PM at different flow rates and different PM thicknesses ...............................69
Figure 65 - Section of the assembly (a), regenerative combustion chamber (b), view of the bottom flange (c), 3D image of the assembly (d) ...........................................................................................................72
Figure 66 - Combustion chamber with porous medium (left) and assembly of the combustion chamber (bottom view) .............................................................................................................................................74
Figure 67 - Flow blurring injector – 3D view (left), cross section (right) ............................................................75
Figure 68 - Images of the flow blurring injector (left) and its components (right) ...........................................76
Figure 69 - Flow blurring injector mounted on the combustor ..........................................................................76
Figure 70 – Sierra Smart Trak2 Series 1000 (left) and Vögtlin Instruments AG Red-y (right) .........................77
Figure 71 - K type thermocouple .....................................................................................................................78
Figure 72 - B type thermocouple .....................................................................................................................78
Figure 73 - Campbell Scientific CR850 datalogger ..........................................................................................79
Figure 74 - ISOTECH Jupiter 650B dry block calibrator ................................................................. 79
Figure 75 - Ignition device .............................................................................................................. 80
Figure 76 - Ignition wire .................................................................................................................. 81
Figure 77 - Image of the flame through the quartz window ............................................................. 81
Figure 78 - Masterflex peristaltic pump ........................................................................................ 82
Figure 79 - Relation between revolutions per minute and flow rate in the peristaltic pump ........ 83
Figure 80 - Kane Analyzer Auto 5-1 .............................................................................................. 84
Figure 81 - FLIR ThermaCAM P25 .............................................................................................. 85
Figure 82 - Radiation paths ............................................................................................................ 85
Figure 83 - Assembled rig ................................................................................................................ 89
Figure 84 - Stainless steel top flange with graphite gasket ............................................................ 90
Figure 85 - Rig scheme ..................................................................................................................... 91
Figure 86 - Insulating wool ............................................................................................................. 92
Figure 87 - Images of the broken ceramic flanges ........................................................................ 93
Figure 88 - Images of the insulated combustion chamber - side (left), bottom (right) .............. 93
Figure 89 - Combustion products composition at 0.04 g/s total mass flow rate .......................... 95
Figure 90 - Combustion products composition at 0.15 g/s total mass flow rate ......................... 95
Figure 91 - Emissions of NOx at different flow rates and equivalence ratios .............................. 99
Figure 92 - Combustion efficiency at 0.04 g/s ............................................................................. 103
Figure 93 - Combustion efficiency at 0.15 g/s ............................................................................ 103
Figure 94 - Flame pictures at different equivalence ratios - 0.04 g/s ....................................... 105
Figure 95 - Flame pictures at different equivalence ratios - 0.06 g/s ....................................... 106
Figure 96 - Flame pictures at different equivalence ratios - 0.08 g/s ....................................... 107
Figure 97 - Flame pictures at different equivalence ratios - 0.1 g/s ............................................ 108
Figure 98 - Flame pictures at different equivalence ratios - 0.12 g/s ........................................ 109
Figure 99 - Axial distribution of temperature - 0.04 g/s .............................................................. 111
Figure 100 - Axial distribution of temperature - 0.06 g/s ............................................................ 111
Figure 101 - Axial distribution of temperature - 0.08 g/s ............................................................ 111
Figure 102 - Axial distribution of temperature - 0.1 g/s ............................................................... 112
Figure 103 - Axial distribution of temperature - 0.12 g/s ............................................................ 112
Figure 104 - Axial distribution of temperature - 0.15 g/s ............................................................ 112
Figure 105 - Temperature T2 (reactants' temperature) at different mass flow rates and equivalence ratios .......................................................... 114
Figure 106 - Temperature T3 (bottom of combustion chamber) at different mass flow rates and equivalence ratios .......................................................... 115
Figure 107 - Temperature T4 (exhaust outlet) at different mass flow rates and equivalence ratios .......................................................... 116
Figure 108 - Fraction of heat transferred from combustion products to cold reactants at different flow rates and equivalence ratios .......................................................... 117
Figure 109 - Thermal image of the combustion chamber - 0.04 g/s, φ=1 .................................... 118
Figure 110 - Thermal image of the combustion chamber. Outer wall temperature - 0.04 g/s, φ =1 .......................................................... 119
Figure 111 - Thermal image of the combustion chamber - 0.06 g/s, \( \varphi = 1 \) ........................................ 120
Figure 112 - Image of the insulated combustion chamber ........................................ 122
Figure 113 - Flame pictures at different equivalence ratios (insulated) - 0.04 g/s .................. 123
Figure 114 - Flame pictures at different equivalence ratios (insulated) - 0.06 g/s .................. 124
Figure 115 - Flame pictures at different equivalence ratios (insulated) - 0.08 g/s ............... 125
Figure 116 - Image of the top side of the combustion chamber - 0.08 g/s, \( \varphi = 1 \) ............ 126
Figure 117 - Flame pictures at different equivalence ratios (insulated) - 0.1 g/s ................. 127
Figure 118 - Image of the top side of the combustion chamber - 0.1 g/s, \( \varphi = 1 \) .............. 127
Figure 119 - Axial distribution of temperature (insulated) – 0.04 g/s .................................. 128
Figure 120 - Axial distribution of temperature (insulated) - 0.06 g/s .................................. 129
Figure 121 - Axial distribution of temperature (insulated) - 0.08 g/s .................................. 129
Figure 122 - Axial distribution of temperature (insulated) - 0.1 g/s .................................. 129
Figure 123 - Temperature T2 (reactants' temperature) at different mass flow rates and equivalence ratios (insulated) ................................................................. 130
Figure 124 - Temperature T4 (exhaust outlet) at different mass flow rates and equivalence ratios (insulated) ................................................................. 131
Figure 125 - Fraction of heat transferred from combustion products to cold reactants at different flow rates and equivalence ratios (insulated) ................................................................. 132
Figure 126 - Comparison of axial distribution of temperature at equivalence ratios 0.85 and 1 – 0.04 g/s .... 133
Figure 127 - Comparison of axial distribution of temperature at equivalence ratios 0.85 and 1 - 0.06 g/s ......... 133
Figure 128 - Comparison of axial distribution of temperature at equivalence ratio 0.85 and 1 - 0.08 g/s .......... 133
Figure 129 - Comparison of axial distribution of temperature at equivalence ratio 0.85 and 1 - 0.1 g/s ........ 134
Figure 130 - Comparison of fraction of heat recovered between insulated and non-insulated case .......... 135
Figure 131 - Comparison of preheating temperatures (T2) between insulated and non-insulated case .......... 135
Figure 132 - Comparison of exhaust gas temperature (T4) at the outlet between non-insulated and insulated case .................................................................................................................. 136
Figure 133 - Properties of selected pure vegetable oils [75]......................................................... 138
Figure 134 - Dependence of kinematic viscosity with temperature of some vegetable oils and diesel fuel [81] 138
Figure 135 - Kinematic viscosity vs temperature of diesel and sunflower oil [81] ................................. 139
Figure 136 - Dependence of density with temperature of some vegetable oils and diesel fuel [81] ......... 139
Figure 137 - Density vs temperature of diesel and sunflower oil [81] ............................................. 139
Figure 138 - Insulated combustion chamber for operation with vegetable oil................................... 142
Figure 139 - Combustion products composition at 0.1 g/s total mass flow rate – vegetable oil .............. 143
Figure 140 - Combustion products composition at 0.12 g/s total mass flow rate – vegetable oil ............ 143
Figure 141 - Emissions of NOx at different flow rates and equivalence ratios – vegetable oil .............. 145
Figure 142 - Flame pictures at different equivalence ratios - 0.1 g/s – vegetable oil ......................... 146
Figure 143 - Axial distribution of temperature (vegetable oil) - 0.1 g/s ...................................... 147
Figure 144 - Axial distribution of temperature (vegetable oil) - 0.12 g/s .................................... 147
Figure 145 - Comparison of axial distribution of temperature at equivalence ratio of 1 when using vegetable oil and LPG - 0.1 g/s .................................................................................. 148
Figure 146 - Preheating temperature of mixture upstream of the porous medium ............................................ 149
Figure 147 - Combustion chamber top view after use with vegetable oil ................................................................. 150
Figure 148 - Combustion efficiency at 0.1 g/s – vegetable oil ............................................................................. 152
Figure 149 - Combustion efficiency at 0.12 g/s – vegetable oil ........................................................................... 152
Figure 150 - Scheme of the combustor model ......................................................................................................... 157
Figure 151 – Mesh 0.00025m element size ............................................................................................................... 160
Figure 152 - Sensitivity analysis - Maximum temperature vs mesh size ................................................................. 160
Figure 153 - Comparison between temperature profiles – 0.08 g/s mixture incoming flow rate, $\phi = 1$ ............. 162
Figure 154 - Main species molar fraction vs axial distance from inlet - 0.08 g/s mixture incoming flow rate, $\phi = 1$ ........................................................................................................................................... 163
Figure 155 - Contours of temperature for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 1 ...................................................................................................................... 165
Figure 156 - Contours of velocity for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 1 ...................................................................................................................... 165
Figure 157 - Contours of CH$_4$ mass fraction for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 1 ........................................................................................................... 166
Figure 158 - Contours of H$_2$O mass fraction for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 1 ........................................................................................................... 166
Figure 159 - Contours of CO$_2$ mass fraction for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7 .............................................................................................................. 166
Figure 160 - Contours of temperature for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7 .............................................................................................................. 168
Figure 161 - Contours of velocity for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7 .............................................................................................................. 168
Figure 162 - Contours of CH$_4$ for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7 .............................................................................................................. 169
Figure 163 - Contours of H$_2$O for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7 .............................................................................................................. 169
Figure 164 - Contours of CO$_2$ for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7 .............................................................................................................. 169
Figure 165 - Contours of temperature (left) and velocity (right) for the adiabatic case at reactants' temperature of 800 K and equivalence ratio of 0.5 .............................................................................................................. 170
Figure 166 - Contours of CH$_4$, H$_2$O and CO$_2$ for the adiabatic case at reactants' temperature of 800 K and equivalence ratio of 0.5 .............................................................................................................. 170
Figure 167 - Contours of temperature for the non-adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 1 .............................................................................................................. 171
Figure 168 - Contours of velocity for the non-adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 1 .............................................................................................................. 171
Figure 169 - Contours of CH₄ for the non-adiabatic case at reactants’ temperature of 298 K, 500 K, 800 K and equivalence ratio of 1 ................................................................................................................................. 171
Figure 170 - Contours of H₂O for the non-adiabatic case at reactants’ temperature of 298 K, 500 K, 800 K and equivalence ratio of 1 ................................................................................................................................. 172
Figure 171 - Contours of CO₂ for the non-adiabatic case at reactants’ temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7 ................................................................................................................................. 173
Figure 172 -- Contours of temperature for the non-adiabatic case at reactants’ temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7 ................................................................................................................................. 174
Figure 173 - Contours of velocity for the non-adiabatic case at reactants’ temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7 ................................................................................................................................. 175
Figure 174 - Contours of CH₄ for the non-adiabatic case at reactants’ temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7 ................................................................................................................................. 176
Figure 175 - Contours of H₂O for the non-adiabatic case at reactants’ temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7 ................................................................................................................................. 177
Figure 176 - Temperature profiles at centreline at equivalence ratios of 0.7 and 1, reactants’ temperature of 800 K and mass flow rate of 0.04 g/s - Comparison between adiabatic case and non-adiabatic case and mass averaged temperature profiles ................................................................................................................................. 178
Figure 177 - Temperature profiles at centreline at equivalence ratio of 1, reactants’ temperature of 298 K and mass flow rate of 0.15 g/s – Comparison between adiabatic and non-adiabatic case ................................................................................................................................. 179
Figure 178 – Contours of temperature gradients in the axial direction in walls and porous medium – non-adiabatic case - equivalence ratio of 1, reactants’ temperature of 298 K ................................................................................................................................. 180
Figure 179 – Main species molar fraction vs axial distance – Comparison between adiabatic and non-adiabatic case at $\phi = 1$ and $\phi = 0.7$ – Mixture mass flow rate 0.04 g/s and reactants’ temperature 800 K ................................................................................................................................. 181
Figure 180 – Contours of temperature with different porous materials at different reactants’ temperatures – non-adiabatic case – equivalence ratio of 1, mass flow rate of 0.04 g/s ................................................................................................................................. 182
Figure 181 – Temperature profiles at centreline with different porous medium conductivities – Equivalence ratio of 1, mass flow rate of 0.04 g/s and reactants’ temperature of 800 K ................................................................................................................................. 183
Figure 182 – Contours of temperature for the non-adiabatic case – Equivalence ratio of 0.5, reactants’ temperature of 800 K and porous medium conductivity of 0.5 W/mK – 0.04 g/s (above), 0.08 g/s (below)….. 184
Figure 183 - Contours of temperature in the non-adiabatic case with different porosities and mass flow rates – equivalence ratio of 1, reactants’ temperature of 800 K ................................................................................................................................. 185
Figure 184 – Contours of temperature in the non-adiabatic case with different external temperatures – equivalence ratio of 0.7, reactants’ temperature of 800 K ................................................................................................................................. 186
Figure 185 - Triple concentric tube heat exchanger model................................................................................................................................. 187
Figure 186 - Triple concentric tube heat exchanger model including porous medium................................................................................................................................. 188
Figure 187 - Double concentric tube heat exchanger model................................................................................................................................. 189
Figure 188 - Double concentric tube heat exchanger model including porous medium................................................................................................................................. 190
Figure 189 – Comparison of analytical and numerical model – Flow temperature vs x coordinate (left), Nu vs axial coordinate (right) – 0.05 g/s total mass flow rate, equivalence ratio equal to 1 .................................................................................................................................................. 191
Figure 190 – Comparison of temperature profiles between analytical formulation and numerical study – 0.05 g/s total mass flow rate, equivalence ratio equal to 1 ................................................................. 197
Figure 191 - Contours of temperature – 0.05 g/s total mass flow rate, 1 ER equivalence ratio...................... 197
Figure 192 – Contours of temperature at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – adiabatic case ............................................................................ 199
Figure 193 - Contours of velocity at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – adiabatic case ............................................................................ 200
Figure 194 – Plots of Nu at 0.5 W/mK (left) and 16 W/mK (right) for the triple tube adiabatic case .......... 201
Figure 195 – Profiles of temperatures of cold mixture in the inner annulus and exhaust gases in the outer annulus along the axial coordinate – 0.5 W/mK; 0.05 g/s (top), 0.1 g/s (center) and 0.15 g/s (bottom) ....................... 203
Figure 196 – Contours of temperature at different flow rates, equivalence ratio of 1 and different wall thermal conductivities ........................................................................................................ 205
Figure 197 - Contours of velocity at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – adiabatic case ............................................................................ 206
Figure 198 - Plots of Nu at 0.5 W/mK (left) and 16 W/mK (right) for the double tube adiabatic case ........... 207
Figure 199 – Contours of temperature at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – non-adiabatic case ............................................................................ 208
Figure 200 - Contours of velocity at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – non-adiabatic case ............................................................................ 209
Figure 201 - Plots of Nu at 0.5 W/mK (left) and 16 W/mK (right) for the triple tube non-adiabatic case ...... 210
Figure 202 - Contours of temperature at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – non-adiabatic case ............................................................................ 211
Figure 203 - Contours of velocity at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – non-adiabatic case ............................................................................ 212
Figure 204 - Plots of Nu at 0.5 W/mK (left) and 16 W/mK (right) for the double tube non-adiabatic case ...... 213
Figure 205 - Heat transfer within the triple tube combustor at different mass flow rates – adiabatic case (left), non-adiabatic case (right) ............................................................................................................. 214
Figure 206 - Heat transfer within the double tube combustor at different mass flow rates – adiabatic case (left), non-adiabatic case (right) ............................................................................................................. 215
Figure 207 - Contours of temperature and velocity at different flow rates, equivalence ratio of 1 and outer wall thermal conductivity of 0.5 W/mK – non-adiabatic case, triple tube system ........................................ 217
Figure 208 - Contours of temperature and velocity at different flow rates, equivalence ratio of 1 and outer wall thermal conductivity of 0.5 W/mK – non-adiabatic case, double tube system ........................................ 218
Figure 209 - Contours of temperature and velocity at different flow rates, equivalence ratio of 1 and walls thermal conductivity of 16 W/mK – non-adiabatic case, triple tube system with porous insert .................. 219
Figure 210 - Contours of temperature and velocity at different flow rates, equivalence ratio of 1 and walls thermal conductivity of 16 W/mK – non-adiabatic case, double tube system with porous insert ........................... 220
Figure 211 – Nu plots at different flow rates for the inner tubes of triple tube and double tube cases with porous medium ...................................................................................................................... 221
Figure 212 - Flow path through Sierra Smart Trak 2 (left) and measuring principle (right) .................................................. 232

Figure 213 - Block diagram of red-y flow controllers ........................................................................................................... 232

Figure 214 – Comparisons between equivalence ratios calculated from emissions and equivalence ratios from the flow meter ........................................................................................................................................ 235

Figure 215 - Scheme of a simple thermocouple [140] ........................................................................................................... 236

Figure 216 - Scheme of a thermocouple with reference junction [140] ................................................................................. 236

Figure 217 – Heat exchanges at the thermocouple bead [60] ................................................................................................. 237
List of Tables

Table 1 - Flame holders and their characteristics [12] ................................................................. 21
Table 2 - Summary of results ........................................................................................................... 59
Table 3 - Results of sensitivity analysis .......................................................................................... 64
Table 4 - Pressure drops across the PM at different incoming flow velocities - CFD results ............ 65
Table 5 - Physical characteristics of sintered steel porous medium ............................................. 73
Table 6 - Kane Automotive Auto 5-1 Analyzer specifications ......................................................... 84
Table 7 - Specifications of Superwool 607 Fibre ......................................................................... 92
Table 8 - Specifications of Duratec750 .......................................................................................... 92
Table 9 - Chemistry of LPG .......................................................................................................... 97
Table 10 – Evaluated physical properties of LPG ......................................................................... 101
Table 11 - Properties of species [68] ............................................................................................ 102
Table 12 – Evaluation of time to pass through the channel .............................................................. 141
Table 13 - Evaluation of droplet evaporation time ......................................................................... 141
Table 14 – Chemistry of sunflower oil .......................................................................................... 144
Table 15 - Governing equations ..................................................................................................... 159
Table 16 - Adiabatic flame temperatures - 0.08 g/s mixture mass flow rate, \( \phi = 1 \) ..................... 161
Table 17 – Laminar burning velocities at different initial temperatures .......................................... 167
Table 18 - Literature review of concentric tubes heat exchangers and small scale heat transfer ...... 187
Table 19 – Governing equations [130] .......................................................................................... 189
Table 20 – Sensitivity analysis ....................................................................................................... 193
Table 21 – Preheating temperature of reactants and temperatures at the outer channel outlet at different mass flow rates and different conductivities in a triple tube system – adiabatic case ............................................. 202
Table 22 – Preheating temperature of reactants at different mass flow rates and different conductivities in the double tube system – adiabatic case .................................................................................................................. 204
Table 23 – Preheating temperature of reactants and temperatures at the outer channel outlet at different mass flow rates and different conductivities in a triple tube system – non-adiabatic case ................................................................. 209
Table 24 – Preheating temperature of reactants at different mass flow rates and different conductivities in the double tube system – non-adiabatic case .................................................................................................................. 213
Table 25 – Preheating temperature of reactants and temperatures at the outer channel outlet at different mass flow rates and in the triple tube system with porous insert ................................................................................................................ 220
Table 26 - Values of N [62] ............................................................................................................ 233
Table 27 - Average values of temperature measurements .............................................................. 238

xx
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Cross sectional area</td>
</tr>
<tr>
<td>$Bi$</td>
<td>Biot number</td>
</tr>
<tr>
<td>$C_2$</td>
<td>Inertial resistance factor</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat at constant pressure</td>
</tr>
<tr>
<td>$D$</td>
<td>Diameter</td>
</tr>
<tr>
<td>$Da$</td>
<td>Damköhler number</td>
</tr>
<tr>
<td>$D_{Hydr}$</td>
<td>Hydraulic diameter</td>
</tr>
<tr>
<td>$d_s$</td>
<td>Specific diameter</td>
</tr>
<tr>
<td>$FA$</td>
<td>Fuel to air ratio</td>
</tr>
<tr>
<td>$Gz$</td>
<td>Graetz number</td>
</tr>
<tr>
<td>$h$</td>
<td>Heat transfer coefficient</td>
</tr>
<tr>
<td>$\Delta h$</td>
<td>Enthalpy difference</td>
</tr>
<tr>
<td>$k$</td>
<td>Ratio between specific heat at constant pressure and specific heat at constant volume</td>
</tr>
<tr>
<td>$K$</td>
<td>Constant</td>
</tr>
<tr>
<td>$K'$</td>
<td>Modified constant</td>
</tr>
<tr>
<td>$k_{eff}$</td>
<td>Effective thermal conductivity</td>
</tr>
<tr>
<td>$k_f$</td>
<td>Fluid thermal conductivity</td>
</tr>
<tr>
<td>$k_s$</td>
<td>Strength coefficient</td>
</tr>
<tr>
<td>$k_w$</td>
<td>Solid wall thermal conductivity</td>
</tr>
<tr>
<td>$M$</td>
<td>Maranzana number</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>Mass flow rate</td>
</tr>
<tr>
<td>$\dot{m}_f$</td>
<td>Fuel mass flow rate</td>
</tr>
<tr>
<td>$n$</td>
<td>Rotational speed [rpm]</td>
</tr>
<tr>
<td>$n_{blades}$</td>
<td>Number of blades</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Theoretical Nusselt number</td>
</tr>
<tr>
<td>$Nu_{k0}$</td>
<td>Nusselt number including axial conduction</td>
</tr>
<tr>
<td>$P$</td>
<td>Perimeter</td>
</tr>
<tr>
<td>$P$</td>
<td>Power</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$Q_{gen}$</td>
<td>Heat generated</td>
</tr>
<tr>
<td>$Q_{lost}$</td>
<td>Heat lost</td>
</tr>
<tr>
<td>$\dot{Q}$</td>
<td>Volume flow rate</td>
</tr>
</tbody>
</table>
\[ r \] \quad \text{Radius}
\[ R \] \quad \text{Regeneration level}
\[ Re \] \quad \text{Reynolds number}
\[ S_i \] \quad \text{Source term}
\[ T \] \quad \text{Temperature [K]}
\[ T_{u0} \] \quad \text{Reference temperature}
\[ U \] \quad \text{Overall heat transfer coefficient}
\[ u \] \quad \text{Peripheral velocity}
\[ u_l \] \quad \text{Laminar burning velocity}
\[ u_{l0} \] \quad \text{Reference laminar burning velocity}
\[ U_{max} \] \quad \text{Maximum peripheral velocity}
\[ V \] \quad \text{Volume}
\[ v_{ax} \] \quad \text{Axial component of velocity}
\[ v_j \] \quad \text{Velocity vector}
\[ W_c \] \quad \text{Eulerian work}
\[ \alpha \] \quad \text{Permeability}
\[ \beta \] \quad \text{Compression ratio}
\[ \gamma \] \quad \text{porosity}
\[ \eta_b \] \quad \text{Combustion efficiency}
\[ \eta_c \] \quad \text{Compression efficiency}
\[ \eta_m \] \quad \text{Mechanical efficiency}
\[ \eta_p \] \quad \text{Polytropic efficiency}
\[ \varphi \] \quad \text{Flow coefficient}
\[ \phi \] \quad \text{Equivalence ratio}
\[ \Phi_{\text{cond}} \] \quad \text{Heat transfer through axial conduction}
\[ \Phi_{\text{conv}} \] \quad \text{Heat transfer through convection}
\[ \mu \] \quad \text{Dynamic viscosity}
\[ \rho \] \quad \text{Density}
\[ \sigma \] \quad \text{Tensile strength}
\[ \sigma_s \] \quad \text{Slip factor}
\[ \tau_{\text{residence}} \] \quad \text{Residence time}
\[ \tau_{\text{reaction}} \] \quad \text{Reaction time}
\[ \omega \] \quad \text{Rotational speed [rad/s]}
Acronyms

GPS  Global Positioning System
UMGT  Ultra-Micro Gas Turbine
M.I.T.  Massachusetts Institute of Technology
K.U. Leuven  Katholieke Universiteit Leuven
MEMS  Micro Electro-Mechanical System
IHI  Ishikawajima-Harima Heavy Industries
UAV  Unmanned Aerial Vehicle
LHV  Lower Heating Value
NGV  Nozzle Guide Vanes
HEX  Heat exchanger
PM  Porous Medium
TPV  Thermophotovoltaic
CO  Carbon monoxide
CO₂  Carbon dioxide
NOₓ  Nitrous oxide
UHC  Unburned Hydrocarbons
O₂  Oxygen
H₂  Hydrogen
H₂O  Water
OH  Hydroxide
N₂  Nitrogen
CH₄  Methane
LPG  Liquefied Petroleum Gas
CFD  Computational Fluid Dynamics
AB  Air-blast
FB  Flow-blurring
TIT  Turbine Inlet Temperature
NTU  Number of Transfer Units
EES  Engineering Equation Solver
AFT  Adiabatic Flame Temperature
1 INTRODUCTION

Over the last 20 years the development of portable devices such as smartphones, laptops, biomedical diagnostic sensors, and GPS (Global Positioning Systems) receivers has led to a miniaturisation of their components and to a higher demand of electricity supply that accumulators and batteries cannot fulfil anymore, even if they are still widely used today. The most suitable devices to provide electric power to the above mentioned small and portable applications seem to be Ultra-Micro Gas Turbines (UMGTs) in fact they are able to generate power in a range from 1 W up to 1 kW with a high energy density. Hydrocarbons burned in air have 20 to 30 times the energy density of the best current lithium chemistry based batteries, so that fuelled systems need only to be modestly efficient to compete well with batteries [1]. Furthermore an UMGT can be fuelled with different types of fuel as liquid or gaseous hydrocarbons, biogases or hydrogen.

In addition, another major issue is represented by the decrease of fossil fuels which may be consumed within the next century, as nowadays they are the most used primary energy source; therefore the common requirement of energy systems is to achieve the cleanest combustion conditions and possibly employing sustainable fuels.

The main applications of UMGTs are related to power production for all those portable or mobile applications requiring small amount of power. These include:

- electronic devices such as laptops, GPS or as a backup for emergency devices (Figure 1);

![Figure 1 – Examples of GPS (left) and laptop (right) - Ref. [2,3]](image-url)
- military applications as exoskeletons for last generation soldiers and for medical purposes, micro-climate cooling systems or UAVs (Unmanned Aerial Vehicles) (Figure 2);

- drones for precision agriculture (Figure 3).

A more exhaustive picture of the possible applications and their specific energy and power requirements is shown in Figure 4.
The first UMGT concept was proposed at the M.I.T. Gas Turbine Laboratory in 1994 thanks to the new manufacturing technique involving etching of photolithographically-
defined planar geometries. With this procedure they were able to produce silicon layers with different shapes extruded on them. The layers were then bonded together to form the UMGT with dimensions 21mm × 21mm × 3.8mm. In Figure 6 the exploded view of the assembly of the UMGT is shown.

![Figure 6 - Exploded view of the M.I.T. turbine assembly](image)

Today there are only few and incomplete prototypes of UMGTs and most of them do not reach acceptable performances. The most common problems that need to be solved are related to combustion instability at micro-scale, rotor failure in operating conditions, bearings wear and lifetime due to high rotational speeds, aerodynamic performances and high strength materials [9-12]. Major advances have been made in the last two decades by several universities including M.I.T. Massachusetts, Stanford University, University of Tokyo, K.U. Leuven in Belgium (with the PowerMEMS project) and Università degli Studi “La Sapienza” of Rome, Italy, however good performances still need to be achieved. The only working prototype of a microscale gas turbine was developed by IHI Corp. which designed and built a 400 W electrical output system where all the components were installed into a portable suitcase [13].

A research on UMGT was recently commenced at the University of Auckland through a Ph.D. thesis work [14] in the Mechanical Engineering Department. This focused on flame stability limits surrounding flat flame combustion that is typically employed for a UMGT. It
has been proven that thermal losses have a great impact on the overall combustion process as they affect the reaction times and flammability limits by lowering the temperature and this could eventually lead to quenching of the flame. Therefore the presently proposed research will extend the mentioned above study to focus on the interactions between combustion and heat transfer with particular focus on cycle regeneration, as they greatly influence combustion dynamics and efficiency.

1.1 Scaling Considerations

A thermal system involves multiple aspects ranging from thermodynamics through structural behaviour to fluid dynamics. Speaking about micro scale systems design, different challenges are involved if compared to large scale machines and optimal component and systems design is different as well.

The thermodynamic cycle which micro gas turbines are based on is the Brayton cycle (Figure 7).

It implies two heat exchanges between the working fluid and an external heat source, one compression and one expansion of the same fluid. Therefore, a conventional gas turbine system consists of a compressor powered by a turbine driven by the expansion of exhaust gases and a combustion chamber. The residual enthalpy in the exhaust flow provides thrust or can power an electric generator. The combustor exit temperature can range from 1200 K to 1800 K in the current machines and the rotor is subject to extremely high peripheral speeds (from 300 to 600 m/s), therefore rotating components are centrifugally stressed to several hundred MPa’s. In addition, the turbine has to withstand high temperatures so the choice of

Figure 7 - Simple Brayton cycle [15]
materials is particularly important. Other important component requirements are low friction and high rotational speed bearings (especially air bearings), tight geometric tolerances and clearances between rotating and static parts to achieve better fluid dynamics efficiency. Additional parameters influenced by scaling down the system include higher viscous forces in the fluid and manufacturing constraints.

Performing a dimensional analysis, an expression indicating the dependence of power density on dimensions can be obtained. Power $P$ is defined as the mass flow rate of fluid $\dot{m}$ evolving in the cycle multiplied by the difference of enthalpy the fluid $\Delta h$ is able to provide, as in Equation (1).

$$P = \dot{m}\Delta h = \rho \dot{Q}\Delta h$$  \hspace{1cm} (1)

Expressing this equation in dimensional parameter, it becomes:

$$P = \frac{kg\ m^3\ kg\ m^2}{m^3\ s^2\ kg} = \frac{kg\ m^5}{m^3\ s^3}$$  \hspace{1cm} (2)

or

$$P \propto \rho D^5 n^3$$  \hspace{1cm} (3)

where $\rho$ is the fluid density, $D$ is the characteristic dimension (typically the outer diameter), and $n$ is the rotational speed. Considering a given pressure ratio, the velocity of the fluid at nozzle exit is independent of the nozzle size (within certain limits) therefore, once the rotor head coefficient has been selected, the tangential speed is constant and independent of turbine size [16]. This means:

$$Dn = 2KU_{max} = constant$$  \hspace{1cm} (4)

where $K$ is a constant of order 1, and $U_{max}$ is the maximum peripheral velocity. The rotational speed is inversely proportional to the diameter. Furthermore:

$$P \propto \rho(2KU_{max})^3 D^2$$  \hspace{1cm} (5)

The expression for the power density can then be obtained as:

$$\frac{P}{V} \propto \frac{K'}{D}$$  \hspace{1cm} (6)
where $K'$ is another constant ($K$ modified), and $V$ is the volume of the machine. This means that the power density increases with scaling down of components.

As indicated above, the thermodynamic cycle is invariant for microscale and large scale machines. On the other hand the fluid mechanics are extremely different. Viscous forces are larger in micro devices, in fact transonic and supersonic speeds can be easily reached if the chosen pressure ratio is between 2:1 and 4:1. This leads to Reynolds numbers in the tens of thousands, which is a small value compared to the $10^6$ of the large scale turbomachinery, and results in higher viscous losses. Aerodynamic studies at low Reynolds numbers are necessary to improve performances. Same importance is given to shock wave losses, and tip leakage. Thermal insulation is also a relevant problem because the high surface area to volume ratio is larger at microscale and this involves higher heat losses (relative to heat generation rates) and consequently efficiency loss. To recover part of the heat lost, in theory the shaft between turbine and compressor could be eliminated to maximize the turbine cooling (by combining compressor and turbine in a “back-to-back” arrangement with a surface in common) but, on the other hand, heating the compressor causes an increase of temperature of the incoming flow (being a non-adiabatic compression), thus a decrease of density of the air which leads to higher power requested by the compressor. Also decreasing the temperature of the turbine influences cycle efficiency as the recoverable work is lower.

The combustion chamber plays an essential role in the design of an effective gas turbine system. The first necessity that should be taken into account is posing a limit to dimensions. Other fundamental requirements of a combustor are high efficiency, low pressure drops, high structural resistance and low emissions. In a conventional size combustion chamber these objectives are easier to achieve as complex structures can be realized with no great problems. On the other side, UMGTs are able to produce a much higher energy density and a higher elaborated flow rate per volume unit.

The dimensions of a combustion chamber are imposed by the necessity of completely developing the reactions, therefore by complying with the residence time of the mixture, which involves a time for achieving complete mixing between the reactants (that scales with the dimensions of the device) and a time for the chemical reaction to complete (that is fixed). The dimensionless number which represents the reaction kinetics is the Damköhler number (Da), which is the ratio between the residence time of the reactants in the combustion chamber and the chemical reaction time. To have a complete reaction Da has to be higher than one and this can be done by decreasing the fluid dynamic time to complete combustion
or increasing the residence time. Increasing residence time also means increasing dimensions, which is not desirable in an UMGT; decreasing reaction time to complete combustion, on the other hand, is possible and can be achieved by preheating reactants (below the autoignition point) or by using catalysts. Power density, as mentioned, is proportional to the inverse of the diameter, and it can also be related to residence time [17] considering that:

$$\frac{P}{V} \propto \frac{\dot{m}}{V} \propto \frac{\dot{m}_f LHV}{V} \propto \frac{\rho}{\tau_{\text{residence}}}$$

(7)

In a micro combustor heat losses are not negligible and it has been demonstrated that the relationship between heat losses and heat generation is proportional to the inverse of the hydraulic diameter of the combustion chamber [18].

$$\frac{Q_{\text{lost}}}{Q_{\text{gen}}} \propto \frac{1}{D_{\text{Hydr}}^{1.2}}$$

(8)

Since the diameter of the combustion chamber are of the order of millimetres (around a hundred times smaller than the conventional ones), the heat losses of a micro combustor are estimated to be about a hundred times higher than the normal-sized combustors.

This is also confirmed by the work presented in [19]. Although only heat losses from the volute are evaluated, it provides an example of the influence of scaling down of components on heat losses (Figure 8 and Figure 9).

Figure 8 - Surface to volume ratio of the volute vs volute inlet radius [19]
As mentioned already, the effect of scaling down the combustor causes a decrease of the Damkohler number, thus the design constraints. Two parameters that can be modified to achieve good performances are the choice of fuel and employing a premixing device to reduce residence time. In this case a flame holder is necessary to prevent flame instability and flashback into the inlet channels.

Another important challenge is related to the materials choice to meet the following physical requirements:

- Cycle maximum temperatures ranging from 1300 K to 1600 K [9,16];
- Elevated peripheral speeds (from 300 m/s to 600 m/s) that translates into highly stressed rotating parts;
- Air bearings to avoid direct contact between rotating and static parts;
- Close tolerances and low clearances.

Small length scale affects material properties and material selection. In this case the constraints are represented by mechanical requirements and fabrication limits. Although the technology is not completely mature yet, the choice of materials is addressed on carbide (SiC) and silicon nitride (Si₃N₄) as their properties include high strength, low density and good oxidation resistance. However further studies are necessary in order to improve manufacturing techniques to produce a flaw-free material. This is feasible in small scale
components as single crystal components are basically perfect at an atomic level; it is much more difficult having a perfect material at large scale in conventional machines.

To optimize cycle efficiency (and make it comparable to the large scale machines) and reduce fuel consumption, a regenerator could be introduced. The purpose of the regenerator is to preheat the combustion air or the reactants (if a premixed combustion is adopted) before the flow enters the combustion chamber. This could be done by recovering heat from the exhaust and transferring it to the compressed unburned gas. The main drawback of employing this solution is the increased complexity and dimensions of the system, as a heat exchanger which requires a considerable amount of surface has to be implemented onto the device. Intuitively, the recuperator needs to be highly efficient and has to limit the pressure drops in the cold and in the hot channels.

1.2 Thesis Aim

The motivation behind this research is the development of a clean and efficient Ultra Micro Gas Turbine for power production. The previous section listed the requirements of such a device, in particular: the necessity of achieving clean combustion, widening flammability limits and having small pressure drop within the combustor including the flame holder. As shown by Turkeli-Ramadan [11,12,14], preheating reactants yields considerable benefits in widening the flammability limits of an air-natural gas mixture, as a result of the increased burning velocity. This also has an effect on the residence time of the mixture in the combustion chamber, thus the total mass flow rate and hence the heat released during combustion can be increased. Not many researches in the literature focus on the performances of the combustion chamber although it is one of the greatest sources of heat losses, due to the elevated temperature difference between its walls and the surroundings. Therefore the aim of this thesis is to provide a deep understanding of interactions between heat transfer and combustion in small scale combustion chambers, to support portable power applications.

1.3 Objectives

The main objective of the experimental study is to investigate thermal performances of a small scale regenerative combustion chamber. Although several combustion chambers were developed for small scale power production devices, a comprehensive study on thermal performances has not been found in literature. The diameter of the cylindrical combustion
The combustion chamber is chosen to be 18 mm with two sets of helicoidal fins surrounding it, allowing the heat recovery from the combustion products. The combustion chamber is tested in order to measure temperatures within the combustion chamber and within the channels where unburned mixture and exhaust gases flow. This allows the evaluation of the heat exchanged between the flows providing an idea of the efficiency of the device. Being clean combustion another requirement of such a burner, the composition of the combustion products is measured and combustion efficiency is evaluated.

Comparisons between insulated and non-insulated combustion chamber are also carried out to highlight the benefits of containing heat losses.

Furthermore a lack of studies on using high viscosity fuels such as vegetable oils at small scale in a premixed combustion configuration is present in the literature. Therefore the second part of the experimental study is focused on testing the same combustion chamber using vegetable oils as fuel, sprayed into the combustor through a flow blurring injector.

The main objective of the first numerical study is to investigate the heat transfer modes and their effects on combustion in an 18 mm diameter combustion chamber with porous medium. A parametric study is performed with varying mass flow rate, equivalence ratio, initial temperature of the mixture, thermal conductivity and porosity of the porous medium. Specifically the objective is to obtain a picture of the heat fluxes through the combustor walls and the porous medium and how they affect the flame.

A second numerical study on a “triple tube” regenerative combustion chamber is also carried out with ultimate goal of investigating heat transfer mechanisms between the flows and thermal performances. The model is the simplification of the actual combustion chamber and it includes three concentric tubes with combustion happening within the inner tube; cold mixture flows in the middle channel and exhaust flow is recirculated in the outer channel. A parametric study is carried out varying mass flow rates, equivalence ratios, thermal conductivities of the walls and also the effects of including a flame holder in the combustion chamber are studied. This study helps understanding the variations of temperature of gases within the channels as well as the heat fluxes through the combustion chamber and heat losses to the surroundings.

In general this research will present a comprehensive set of experimental results on a regenerative small scale combustion chamber for UMGT applications, providing a better understanding of the heat transfer mechanisms happening within it and useful information for designing such devices.
1.4 Thesis outline

This thesis consists of eight chapters.

Chapter 1 presents an introduction to UMGTs and their possible applications. Also a brief summary of the main problems and issues at micro scale that require improvement is highlighted. A series of objectives is listed with a brief outline of the sections exposed in this work.

Chapter 2 presents a literature review, exposing researches from all over the world focused on Ultra Micro Gas Turbines. Research gaps are also highlighted in order to better understand the purpose of the present study.

Chapter 3 describes the fundamentals of a gas turbine design and shows the process and the calculations that led to the design of the regenerative combustion chamber. This has been carried out using the traditional method employing Balje maps and characteristic parameters extrapolated from well-known experimental curves and data. To set the thickness of the flame holder and therefore the characteristics of the flow in the combustor, a numerical study on pressure losses through the porous medium and uniformity of the flow downstream of its bottom surface is also carried out and exposed in this chapter.

Chapter 4 provides details on the experimental setup, and the equipment utilized during the tests. This includes calibration procedures and technical specification of the materials used. Working principles of the equipment is also described as w, with further details included in Appendix A1 and A3.

Chapter 5 is dedicated to the exposition of the results on the regenerative combustor. Measurements include composition of combustion products and subsequent evaluation of the equivalence ratio of the mixture, temperatures at four different points within the combustion chamber and axial distribution of temperature in the combustion chamber. Also visual observation of the flame is performed and pictures at different equivalence ratios and mass flow rates are presented. Evaluation of combustion efficiencies and thermal performances is also presented in this section. A second session of tests on the same combustion chamber insulated from the external environment is carried out and exposed in the second part of this chapter. A comparison between the results obtained in the two cases is made. Finally an investigation of the feasibility of using vegetable oils as fuel in the combustion chamber was carried out and the results and issues encountered are discussed.
Chapter 6 presents the first numerical work, focused on axially symmetric numerical simulations on a cylindrical combustion chamber including porous medium. The purpose is to investigate the interactions between flat flame combustion and heat transfer mechanisms within the combustion chamber. The study includes validation against experimental results and a parametric study with changing equivalence ratio, mass flow rate, incoming mixture initial temperature, porosity and thermal conductivity of flame holder. Results are presented in terms of contour mas of temperature and velocity and also profiles of temperature and exhaust gases compositions are shown and discussed.

Chapter 7 describes another numerical study carried out on an axially symmetric two dimensional model of a three concentric tubes combustion chamber, which is the simplification of the combustor designed and tested for the experimental work. This is focused on heat exchanges between the cold and hot flows in the combustion chamber when combustion is occurring in the inner tube. Again, a parametric study is performed with varying equivalence ratio, mass flow rate and thermal conductivity of the walls. Also the effects on heat transfer mechanism of including a porous flame holder in the combustion chamber are highlighted and discussed.

Finally, in Chapter 8 the conclusions of this work and suggestions for future developments are presented.
2 LITERATURE REVIEW

The interest in small size power generation devices has recently developed thanks to the introduction of new manufacturing technologies based on semiconductors. The utilization of this new methodology enabled the production of millimeter scale systems which could not be machined the same way as large devices (cast, forged, machined and assembled). This industry sector is part of the broader micro electrical-mechanical systems (MEMS) and has been the subject of several researches and publications in the last 20 years. A review of the major works is presented in the following section.

2.1 The UMGT at M.I.T.

The first approach towards micro turbines dates back to the mid-1990s at M.I.T. (Massachusetts Institute of Technology) by Prof. A. Epstein and his research group, who are considered the pioneers in this field. Their proposed UMGT was made up of six silicon wafers and hydrogen was adopted as fuel because of its wide flammability limits and fast reaction times. The centrifugal compressor and the radial turbine had diameters of 8 mm and 6 mm respectively and the peripheral speed of the compressor was designed to reach up to 500 m/s, therefore the correspondent rotational speed that resulted was $1.2 \times 10^6$ rpm [1].

![Section of the M.I.T. UMGT](image10.png)

The turbine inlet temperature was chosen to be 1600 K as design point, thus turbine cooling was fundamental and it was achieved by heat conduction through the structure. The system was designed to provide an output of 17 W of electric power however no tests were performed on this device.
The key conclusions and the implications for future development of this UMGT were identified as follows:

- Non-adiabatic condition negatively affects compressor performance;
- Thermal management has to be improved (thermal barriers, turbine cooling);
- Fluid dynamics and tip clearance (improvement of performances by eliminating casing drag) need investigation;
- Efficient high speed bearings have to be developed (over 1 million rpm rotational speed);
- Materials choice and manufacturing techniques need to be investigated;
- The loss in micro turbine Nozzle Guide Vanes (NGV) is about 10 times higher than the value for conventional sized NGV, therefore accurate studies are necessary to reduce losses.

### 2.2 The project at IHI Corp.

Another interesting project was carried out by a Japanese research group including IHI Corp., The University of Tokyo and Tohoku University. IHI Corp. has already tested and prototyped an Ultra Micro Gas Turbine with 400 W electrical power output operating at 400,000 rpm and it is currently undergoing optimization processes for a future commercialization [13]. Also a 100 W micro turbine with a 10 mm diameter centrifugal impeller was studied and experimentally tested [20]. The configuration of this UMGT is similar to the conventional sized gas turbines and the material used for the compressor is Ti-6Al-4V. The device was designed to operate at a rotational speed of 870,000 rpm using hydroinertia gas bearings. The compressor, the combustion chamber and bearings were tested separately achieving good results in terms of performance. The tests were considered acceptable as the bearings stably operated at 870,000 rpm. The compressor achieved an adiabatic efficiency of 68 % and operational maps have been plotted up to 83 % of the design speed. Two different combustion chambers were investigated and tested for usage with hydrogen and methane. Can-type combustors were selected because of their lower heat losses compared to the annular type combustor. The hydrogen fuelled combustion chamber had a volume of 2 cm$^3$ and the methane fuelled one had a volume of 17 cm$^3$. In both cases self-sustained combustion was achieved along with a combustion efficiency of 99.9%.
2.3 The flat-flame combustor at Tokyo Metropolitan University

The research at Tokyo Metropolitan University was focused on the investigation of flat flame combustion for UMGT using propane as fuel [21]. The dimensions were extremely small as the combustion chamber had a diameter of 25 mm and a height of 4 mm with an exit nozzle of only 1 mm height. The combustion chamber included a 1 mm thick porous plate made of alumina-ceramics, a central shaft, an exit nozzle and it was surrounded by a quartz tube. The study was aimed to find the flammability limits at different equivalence ratios at room temperature and atmospheric pressure. The flame velocity was calculated and it was compared with the incoming mass flow rate. Since the flat flame condition is achievable only when the burning velocity is higher than the incoming flow velocity, it was shown that flat flame could be developed only for very small flow rates (under 0.06 g/s) at ambient conditions. Otherwise, increasing reactants temperature was proposed as a solution to this obstacle, as the burning velocity could be considerably higher. The research also showed how the flammability limits depend on equivalence ratio and air mass flow rate. The flame extinction limit had a minimum when the air flow rate was set at around 0.04 g/s and increased gradually with the air flow rate. It was found that a great contribution for the development of the flat flame was given by the heat conduction through the porous plate to the incoming mixture. Also high combustion efficiency was obtained and results showed that at around 0.5 equivalence ratio it ranged between 99.5 and 100 %.

In a following research of the same research group, the flat-flame combustor was modified with the insertion of a Platinum (Pt) catalyst to increase the reaction rate of methane. Two cases were studied: the first one had the porous plate coated with Pt; in the second case the exit nozzle had a thin Pt film. With the Pt flame holder, catalytic combustion was observed to happen within the holder, no flame was generated and the combustion efficiency was lower than that without a catalyst. In the second case, flat flame was achieved and catalytic reactions also occurred on the surface of the Pt nozzle. Results showed a decrease in CO mass fraction and an increase of combustion efficiency in this last situation.

2.4 The Belgian PowerMEMS project

This research group developed a 1 kW micro turbine which consisted of a 20 mm compressor, a combustion chamber, a turbine, an electric generator and a recuperator [22].
Operating parameters were set at: 1200 K for the turbine inlet temperature, 500,000 rpm rotational speed, 3:1 pressure ratio. The general layout is shown in Figure 11.

![Figure 11 - The UMGT of the Belgian PowerMEMS project][22]

As can be seen from the section in Figure 11, the recuperator is the biggest part of the system and includes a counter-flow micro-channel heat exchanger, consisting of 6 cube-like blocks arranged in an annular shape around the can type combustor. Each block consists of alternating hot and cold plate layers with micro-channels. Numerical simulations of this recuperator (67 mm long and with 60 mm internal diameter) showed an efficiency of 87%; pressure drops in the cold channels were found to be higher than in the hot channels. The combustion chamber was designed to operate with a flow speed of 20 m/s, resulting in a length of 50 mm and a height of 15 mm. In the following steps a geometry with plates in the combustion chamber was adopted, in order to provide sufficient dilution between combustion products and air. No further experimental results on the combustion chamber were shown.

![Figure 12 - Turbine (left) and compressor (right)][22]
The full 3D blades employed in the turbine and compressor are shown in Figure 12. The material employed for the turbine was a ceramic composite Si$_3$N$_4$-TiN with good mechanical, thermal and machining properties and the material used for the compressor was Ti-6Al-4V.

### 2.5 The UMGT coupled with a piezo converter

Another UMGT composed of seven stacked silicon wafers was developed by the National University of Singapore [23]. The novelty of this prototype was the addition of a piezoelectric power converter exploiting the rotation of the turbine. The system was designed to operate with an air mass flow rate of 0.36 g/s, compression ratio of 4:1, rotational speed of 1.2 million rpm and hydrogen mass flow rate of 17 g/h.

![Figure 13 - Scheme of the UMGT](image)

The outer diameters of compressor and turbine were both 4.11 mm and they included respectively, diffuser and nozzle vanes. According to the predictions the system is able to generate a power output of 39 W. CFD simulations of the combustion chamber were performed [23] in order to investigate the temperature distribution and the usefulness of the reactants recirculation channel. Experimental data show the recorded temperature at the turbine exit at different positions over a range of flow rates. The flame was able to sustain at flow rates over 0.04 g/s.

### 2.6 The UMGT at the Korea Institute of Machinery & Materials

In this paper a UMGT able to generate 500 W was designed at the Korea Institute of Machinery and Materials [24] through parametric studies and Computational Fluid Dynamics simulations to investigate the performances of the components. The system included radial
compressor and turbine with a generator mounted on the compressor side and a can type combustion chamber able to recuperate part of the heat in the combustion products through a series of stacked plates. The compression ratio was set at 3.055:1 and the characteristic rotational speed at design point was 0.65, resulting in a rotational speed of 400,000 rpm and mass flow rate of 0.02 kg/s.

![Figure 14 - Scheme of the UMGT](image)

Several parametric studies were performed with varying mass flow rate in order to predict the isentropic efficiency and power (absorbed or generated) of compressor and turbine. No experimental tests were executed.

## 2.7 The Research at the University of Auckland

The research at the University of Auckland started with the Ph.D. thesis of Zerrin Turkeli-Ramadan [14] which followed her previous work in collaboration with Yuasa and Sakurai, aimed at the development of the HEX-combustor [25].

The general idea involved incorporating a heat exchanger in the combustion chamber through a triple tube system where the inner tube acts as combustor, the unburnt reactants flow through the annular space between the inner tube and middle tube, and the exhaust gases are recirculated through the annular space between the middle tube and outer tube.
In this system reactants are preheated by the heat received from both the combustion chamber and the exhaust gases. At the top of the combustion chamber a porous medium was inserted to achieve flat flame combustion. The device was successfully tested and it was proved that increasing reactants’ temperature widens flammability limits but new questions were introduced, especially regarding flat flame burners [10-12]. In any case there was no mechanism to control reactants’ preheating temperature and the sight glass did not allow the complete visualization of the flame shape.

During the research at the University of Auckland, Turkeli-Ramadan [10-12] focused on the combustion aspects, in particular on characteristics of flat flames developed over a flame holder.

Initially the effect of using different flame holders on the combustion characteristics was investigated. The experimental setup was scaled up in dimensions compared to the HEX-combustor, as the combustion chamber diameter was 46 mm. A series of tests were performed using a perforated steel plate, a sintered stainless steel porous insert and an alumina ceramic porous material as flame holders. First the pressure losses across the holders were investigated, as one of the requirements of UMGT systems is containing the pressure
drop across the whole device to within 5% of the operating pressure. The flame holders under investigation and their characteristics are summarized in Table 1.

<table>
<thead>
<tr>
<th>Material</th>
<th>FA024</th>
<th>FA060</th>
<th>FA100</th>
<th>R100</th>
<th>R200</th>
<th>SSFilter</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Alumina ceramic</td>
<td>Alumina ceramic</td>
<td>Alumina ceramic</td>
<td>Stainless Steel</td>
<td>Stainless Steel</td>
<td>Stainless Steel</td>
</tr>
<tr>
<td>Average pore size (µm)</td>
<td>580</td>
<td>230</td>
<td>130</td>
<td>100</td>
<td>200</td>
<td>Unknown</td>
</tr>
<tr>
<td>Porosity (%)</td>
<td>45–50</td>
<td>45–50</td>
<td>45–50</td>
<td>43–48</td>
<td>49–54</td>
<td>Unknown</td>
</tr>
<tr>
<td>Density (g/cm³)</td>
<td>2.3</td>
<td>2.3</td>
<td>2.2</td>
<td>4.1–4.5</td>
<td>3.6–4.0</td>
<td>Unknown</td>
</tr>
<tr>
<td>Thickness (mm)</td>
<td>2.4</td>
<td>1.2,3</td>
<td>1.2,3</td>
<td>3</td>
<td>3</td>
<td>1.6</td>
</tr>
</tbody>
</table>

Table 1 - Flame holders and their characteristics [12]

The pressure drops across different flame holders were measured over a range of velocity from 0 to 2 m/s and at ambient pressure. Results showed that the objective of containing the pressure drop under 5% of the operating pressure was achieved up to an incoming flow velocity of 1.2 m/s with every flame holder. Another conclusion is that the geometric parameters (in particular thickness) of the flame holders have a greater effect on pressure loss than the material of the flame holder.

Figure 16 - Pressure losses vs incoming flow velocity [12]

A second session of tests was aimed to study the flame behaviour with different equivalence ratios and different incoming flow velocities to determine the range where stable flat flame combustion was able to be achieved.

Results showed that flat flame was generated without preheating reactants only for low velocities and for equivalence ratios in the range between 0.45 and 0.9.
The different types of flame holder had little influence on flame stability. One result to be highlighted is that the alumina flame holder FA060t1 cracked because of the elevated thermal stress [11].

In a second study, the effects of preheating the reactants on the flammability limits were explored. The flammability limit when using reactants preheated at 600 K is shown in Figure 18, in the range from 0.38 g/s to 1.34 g/s.

Figure 17 - Flat flame operating conditions [12]

![Flammability limits of natural gas-air mixture](image)

Figure 18 - Flammability limits of natural gas-air mixture [11]
It is interesting to note that the minimum equivalence ratio for stable flames decreases considerably when the reactants temperature is increased to 600 K. Also, NO<sub>x</sub>, CO and CO<sub>2</sub> emissions were measured during the experiments. Figure 18 shows how they change at an air mass flow rate of 0.38 g/s and different equivalence ratios, ranging from about 0.6 to about 1.2, although the operating range for UMGT’s adopting a flat flame is on the lean side. It is important to highlight the fact that for equivalence ratios lower than 1, CO emissions are zero, resulting in clean combustion.

![Exhaust Gas Composition](image)

Figure 19 - Emissions at air mass flow rate of 0.38 g/s [11]

## 2.8 The UMGT at Onera

Another project under development is the DecaWatt program at Onera aimed to realize a prototype of an Ultra Micro Gas Turbine with power output from 50 to 100 Watts [26], designed to operate with hydrogen as fuel. The centrifugal compressor is designed with a diameter of 10 mm and rotational speed of 840,000 rpm for an estimated pressure ratio of 2.56:1.

![UMGT at Onera](image)

Figure 20 - UMGT at Onera [26]
In this configuration the 2.5 cm³ combustion chamber had an annular shape and it operated with a reverse flow always in laminar conditions; a recuperator was not implemented in this study. Numerical tests reported in the literature, showed a comparison between the temperature profiles and energy released within the combustion chamber using reduced reaction mechanisms describing propane and hydrogen oxidation. Results showed very similar temperature contours and the similar flow pattern in the combustor; the only difference was the incomplete combustion when using propane as fuel. The reason for this was the small volume of the combustion chamber related to the reaction time of propane, resulting in a Damköhler number close to 1. The experimental tests were focused on investigating compressor and generator performances; in particular they were tested at 500,000 rpm and 700,000 rpm respectively. Results showed the poor performances of the compressor compared to the numerical predictions and this was attributed to excessive tip clearance, bad alignment between blades and casing and poor pressure recovery by the diffuser. The generator on the other hand showed good results in terms of energy conversion. Future works will be aimed to control the clearance between the casing and the blades to achieve the desired pressure ratio.

2.9 The UMGT at the University of Rome “La Sapienza”

This research group started a project related to an UMGT in 2006 with a feasibility study including series of simulations of the Brayton cycle associated with the micro gas turbine considered [16]. Results showed that the optimal compression ratio was 7 (with considerable increase in dimensions and compression stages) and the maximum achievable efficiency ranged from 6 % to 20 % respectively for the simple Brayton cycle and for the intercooled and regenerated Brayton cycle with a compression ratio equal to 2. After the preliminary design phase, the components of the UMGT were created with the aim of producing a power output of 300 W maintaining a maximum pressure of 2 bar [27,28]. In the following stages, developments were aimed to test the compressor and the original design of the combustion chamber. The ultimate device is divided in two blocks: the first one consists of the compressor/turbine group coupled with the electric generator; the second group consists of the combustion chamber which incorporates a regenerator. The design parameters of the cycle were: rotational speed ranges from 150,000 to 200,000 rpm, temperature range in
the heat exchanger/recuperator from 450 to 1350 K, external diameter of the compressor of 38 mm and power output around 300 W.

The most interesting innovation introduced in this research is the regenerative combustion chamber which is made up of a three layer cylindrical shell. The air from the compressor flows through the internal channel, the exhaust gases in the external one and the combustion happens in the inner cylinder. In this configuration the compressed air is pre-heated by both the exhaust gases and the combustion heat. The flows in the fin passages are countered, in order to maximize the heat exchange. The combustion was not premixed but a pre-whirl was implemented. The combustion chamber was designed to operate with different hydrocarbon fuels either gaseous or liquid but no “hot” test results were found in literature. Pictures of the combustion chamber are shown in Figure 22 and Figure 21.
The results obtained by the research group were focused on drawing the operating map of the compressor expressed as a function of the compression ratio and mass flow rate; compressor efficiency expressed as a function of the compression ratio was also investigated. The values obtained from the tests showed that compressor efficiency ranged from 0.49 to 0.7. Another test was related to the pressure losses in the combustion chamber/regenerator, in particular in the inner fins. A compressed air flow was directed into the channels and the outlet pressure was measured. Results showed that the total pressure drop was lower than 2% of the outlet pressure, thus considered widely acceptable.

2.10 The “Swiss roll”

Another heat recuperating combustor is represented by the “Swiss roll”, which was initially developed by Lloyd and Weinberg [29] in 1974. It was composed of a strip of Inconel 600 with a width of 50 mm wrapped in a spiral with 4.5 turns. They carried out a simple experiment proving the validity of the heat recovery concept and recording the flammability limits of the device operated with methane as fuel. More recently investigations on this type of devices continued with Ronney et al. [30,31] who carried out experiments in order to compare extinction limits of propane-air mixtures in Swiss rolls with and without catalyst located on the walls in the reaction volume. In this experimental work a 3.5-turn square Swiss roll burner with dimensions $70 \times 70 \times 50$ mm was tested over a wide range of Re number and equivalence ratios. Results showed that the catalyst was able to sustain combustion even at extremely low Re number (around 1) while this was not possible without catalyst (minimum Re was about 40). Another characteristic of the catalytic Swiss roll was
the ability to have combustion even at very low temperature. At Re=1.2 the maximum recorded temperature with combustion occurring was 350 K. It was noticed that the minimum temperature requirement to sustain combustion increases when Re increases as a result of the shorter residence time, therefore a faster reaction is required to support combustion. Ronney et al. [31] also worked on an analytical model of a Swiss roll combustor to identify the scaling parameters in order to predict the thermal performances of this device. Subsequently three dimensional numerical simulations were also carried out. Four scaling parameters were found out to be important to characterise the performances of a Swiss roll burner over a wide range of scales, in both laminar and turbulent conditions.

Kim et al. [32,33] carried out several experiments on Swiss rolls with different dimensions and shape factors. Also in this case extinction limits were investigated and compared between the different devices. It was found out that the channel length does not represent an issue in terms of flammability limits; on the other side the channel width and the convective heat transfer between mixture and walls play an important role in flame quenching. The measured exhaust gas concentration of the Swiss roll combustor reflected the chemical equilibrium calculation in terms of \( \text{CO}_2 \); \( \text{NO}_x \) concentration was always below 150 ppm at every mean gas velocity and equivalence ratio considered and the total hydrocarbon concentration was negligible at the channel exit, denoting clean and efficient combustion. It was also found out that the CO emissions increased as the combustor size decreased. Finally the thermal efficiency was also computed; results showed that the maximum efficiency had a value of 85 % and as the mean velocity increased, the efficiency decreased.

![Figure 23 - Swiss roll burner [32]](image)

### 2.11 Lean Premixed Pre-vaporized Combustion for Liquid Fuels

Traditionally in gas turbines the method employed to supply fuel to the combustion chamber is direct injection of liquid fuel (kerosene, oil). Consequently injectors are an
important component of the system, as they have to provide good fuel atomization in order to prevent incomplete combustion and pollutant emissions. On the other hand, this diffusion mode tends to produce high levels of NO$_x$ as they are dependent on the maximum temperature reached in the combustion. To limit this phenomenon water or steam is injected into the combustion chamber with the aim to lower the temperature. However this increases the production of CO as a result of local quenching effects. So the “wet injection” mode requires a compromise between CO emissions and NO$_x$ emissions. Recently, stringent emission standards have led to the development of lean premixed combustion to be utilized in gas turbines for power production and industrial applications, as this method provides low NO$_x$ and CO emissions without injecting water or steam. The utilization of liquid fuels in premixed mode introduces autoignition and flashback problems in the premixing section, as the ignition delay time of the fuel/air mixture is shorter than the mean residence time of the mixture in the premixing section [34].

Recent studies have investigated the feasibility of vaporizing the fuel in an inert environment and mixing it with the combustion air [34] before entering the combustion chamber. In this case, premature autoignition is controlled by the level of inert gas in the vaporization process. This research utilized nitrogen as inert gas and it showed that NO$_x$ emissions were lower than those when using natural gas and a stable combustion was reached. Ideally, as inert gas, exhaust gases can be recirculated and used to vaporize the liquid fuel.

However, employing this solution in Ultra Micro Gas Turbines poses several problems, as a recirculation system and control devices have to be implemented at a micro-scale and this introduces additional design problems and increase of overall dimensions.

### 2.12 Premixed combustion using liquid fuels

#### 2.12.1 Liquid fuel combustion using a porous medium

In this study [35] a 67 mm diameter burner surrounded by two preheating annuli was developed and tested using liquid kerosene as fuel. The preheating annulus was filled with stainless steel beads which had the purpose of vaporizing and mixing the kerosene inserted by gravity from the top end with the combustion air. First a mixture of natural gas and air was used to initialize the combustor and reach the steady conditions; subsequently the kerosene was inserted using a peristaltic pump in the preheating annulus. At the same time the natural
gas flow rate was gradually reduced until the system was operated only on kerosene. The equivalence ratio was kept constant at 0.57.

Measurements of temperature along the system surface showed that the value remains constant along a great part of the wall length. The temperature of the combustion products at the outlet showed a uniform radial profile with temperatures ranging between 1000 K and 1150 K. The emissions measurements show the achievement of clean combustion as the concentration of CO in the exhaust gases was ranging between 4 and 10 ppm. NOx concentration reached values up to 60 ppm at 5.55 kW heat release rate; emissions were considerably lower when decreasing heat released at 2.22 kW. Also percentages of heat loss and heat recirculated are shown. Heat losses ranged between 30 and 50 % of the lower heating value of the fuel while the heat recirculated was included in the range 7.5 % to 10 %.

2.12.2 Flow blurring injector applied to annular combustor

A solution to solve the issues in using liquid fuels in UMGTs came unexpectedly from a discovery on an injector under investigation at the University of Sevilla [36]. It was found out that a geometric parameter affects the spray breakup, therefore atomization properties of the injector (which was later called “flow blurring” - FB). Subsequently further developments at
the University of Alabama [37,38] deeply investigated its behaviour and atomization performances. A scheme of a flow blurring injector is shown in Figure 25.

![Flow blurring injector diagram](image)

Figure 25 - Flow blurring injector [31]

In the flow blurring injector fuel flows through an inner channel and atomizing air is introduced in proximity of the exit nozzle in a radial direction. The outlet of the inner channel has the same diameter as the exit orifice and both sections face each other. The flow blurring injector was developed in 2005 [36] and showed dramatic improvements in spray atomization compared to air-blast injectors. Good atomization efficiency is usually based on maximizing surface production, minimizing droplets coalescence and minimizing gas expenses. Generally a good atomizer is a very complex system and therefore its size increases. The flow blurring injector was an unexpected discovery due to unpredictable flow behaviour. In Figure 26, a usual flow focusing atomizer is shown where the flow follows a “focusing” pattern.
The novelty of the flow blurring injector is very simple and it’s based on a geometric parameter that causes the air to interact with the liquid fuel flow causing an efficient perturbation leading to a pre-atomization of fuel before the nozzle exit. It was observed that the ratio between the height of the lateral orifice $H$ and the diameter of the exit orifice has to be less than or equal to 0.25. That means that the limit condition for this phenomenon to happen is that the lateral passageway must equal the exit orifice area.

$$\frac{\pi D^2}{4} = \pi DH \quad \text{so} \quad \frac{H}{D} = 0.25$$

The edges of the inner tube are sharpened to ease the perturbation of the back-flow air. This allows the disaggregation of liquid fuel particles and provides an atomization of one order of magnitude higher compared to the air-blast (AB) injectors. This is necessary to achieve complete combustion and minimize NO$_x$, CO and particulate matters emissions.

The properties of cold sprays of diesel and vegetable oil fuels have been investigated and compared [39]. Results showed that the droplets’ mass weighted Sauter Mean Diameters
of both fuels produced by the FB were comparable with values of 34 μm and 38 μm, respectively for diesel and vegetable oil (with an Air-to-Liquid mass ratio of 2). Also, diesel and vegetable oil produced nearly identical droplets distribution with the biggest droplet diameter of 125 μm for diesel and 160 μm for vegetable oil at the center of the spray and at 20 mm from the injector outlet. The combustion properties were also studied by the same researcher and they demonstrated that the flow blurring injector is able to produce clean and efficient combustion [37].

![Figure 28 - Droplet distribution of diesel and vegetable oil at the spray and 20 mm axial distance](image)

The system developed was made up of a 1 cm diameter combustion chamber and a counter-flow heat exchanger using the enthalpy of exhaust gases to increase reactants’ temperature. It has been proven in studies listed above that increasing reactants’ temperature contributes to achieve higher thermal performances and complete combustion. In this case kerosene was adopted as fuel, flow blurring injectors were installed to provide good fuel atomization and also a porous media was installed to stabilize the flame.
Figure 29 shows a representation of the combustor composed of a cylindrical combustion chamber, an annular heat exchanger with plane walls surrounding the combustion chamber, and a porous medium to optimize the air-fuel mixing, to stabilize the flame and to achieve lean premixed combustion. The reactants are introduced in the annular gap by the flow blurring injectors and, after being preheated, they flow through the porous material and then burn in the combustion chamber. The porous media is Silicon Carbide (SiC) with porosity of 85 % to minimize the pressure drop. Results showed that 94 % of the heat released was contained in the reaction products and the remaining part was lost to the ambient. The combustion was stable over a range of different air and fuel flow rates and produced a smooth, clean flame, attached to the porous medium.

The same research group also carried out a numerical study [40] on a small scale combustor in order to further investigate the thermal properties and exchanges between combustion products and unburnt mixture. The combustor was scaled up compared to the experimentally tested one and it was modelled using an axially symmetric domain, as shown in Figure 30.
The simulations included radiation using the Direct Ordinate model and combustion was represented by a 1 mm thick heat source area placed just downstream of the porous insert. The validation of the model was made against experimental data on the small scale combustor in terms of radial temperature distribution at the exit plane and temperature of the exterior wall surface. Results focused on investigating the amount of heat exchanged between the fluids and the walls and heat transferred through axial conduction in the solid walls. A parametric study was also performed varying the thickness of the walls, their thermal conductivity and emissivity. Two major heat pathways of heat losses to the surroundings were identified: axial conduction through the combustor tube and radiation heat transfer across the preheating annulus. It was also stated that wall thickness had minor influence on heat losses.

2.13 Fuel-film liquid combustion for a TPV power system

A liquid fuel combustion system for thermophotovoltaic (TPV) applications employing fuel-film technique was developed and tested by Li et al. [41]. This type of system is able to convert radiant heat directly into electrical power therefore potentially it could be coupled with a small scale combustor, where heat losses to the surroundings are important. Figure 31 shows the schematic diagram of the experimental combustor. A porous medium is inserted into the main combustor to separate the liquid fuel trough and the reaction volume. Liquid hydrocarbon is injected from the two inlets at the bottom using a syringe pump while air enters the combustion chamber tangentially through a swirl inlet to enhance mixing. Another
function of the porous medium is to facilitate fuel vaporization absorbing heat from the flame.

An emitter tube made up of Al$_2$O$_3$ and ZrO$_2$ surrounded the combustion chamber to maximize radiation emission over a broad spectrum. The combustor was encased in a return tube to maximize chamber illumination and coupled with GaSb photovoltaic cells to produce electricity. Results showed that the wall of the combustion chamber had a uniform illumination and the flame was confined inside the combustion chamber. It was highlighted that the power output was dependant on combustion efficiency and on the emitter temperature. Two fuels were tested in the newly developed device: n-heptane and pentane. Results showed that the TPV system was able to produce 8.3 W at stoichiometric conditions with a fuel flow rate of 12 mg/s while the output of pentane resulted slightly lower. This study confirmed the feasibility of coupling small scale combustion chambers with photovoltaic cells.

2.14 A Micro Gas Turbine fuelled with vegetable oils

The interest towards use of vegetable oil for stationary power generation and for the transport sector is rapidly growing, especially in Europe, as the price of oil is continuously increasing. So far this type of fuel has been widely tested in reciprocating internal combustion engines but the available literature related to its usage in gas turbines is limited and there are many aspects of these to be investigated. Recently, an Italian research group
started a project focusing on the utilization of vegetable oils in Micro Gas Turbines (MGT) with the aim to study the feasibility of this solution and to propose appropriate modifications for the engine to achieve optimal performances [42,43]. The turbine chosen for the test was able to provide a power output of 25 kVA and performances and emissions were measured with different fuels (diesel fuel, biodiesel and straight rapeseed oil). In particular, biodiesel and rapeseed oil were also preheated before the injection in the combustion chamber. This is necessary to reduce their viscosity, which especially for vegetable oils can be ten or even more times higher than that for diesel. At first the MGT was fed with diesel, subsequently it was fuelled with biodiesel (with and without preheating), with straight vegetable oil and with a blend of these two (25 %, 50 % and 75 % of vegetable oil in the mixture). Results showed that CO emissions were similar for all the fuels tested if vegetable oils are preheated to 120 °C. The preheating was also necessary to ensure smooth operation of the engine and avoiding engine shut down. Despite this, at idle state the MGT could not always run on pure vegetable oil, as flame blow out occurred. Biodiesel, being very close to diesel in terms of physical and chemical properties, did not show any particular problems. An expected result was the higher fuel consumption measured with biodiesel and vegetable oil, as their lower heating value is smaller than that for diesel. At the time of writing, publications regarding the utilization of vegetable oils in smaller scale combustion chamber were not found.

2.15 Laminar combustion

In this section a brief overview of laminar flame propagation will be shown to better focus the attention on combustion phenomena.

A simple laminar premixed flame can be produced by flowing a fuel-air mixture through a porous plug, as shown in Figure 32 [44,45]. The flame front in this case is flat and sits on the upper surface of the porous medium at a fixed distance. The fuel-air mixture flows with a velocity $v$ and, in order to remain stable, the flame must propagate in the opposite direction of the flow with the same velocity. The laminar flame speed $S_L$ is the velocity at which the flame propagates in the cold mixture; it is determined by the rates of the reaction of combustion and by the diffusion of energy and species into the unburnt mixture. The temperature of the mixture is raised by the heat released during combustion to the point that the combustion reaction can proceed at an appreciable rate. Once the reaction starts combustion is very fast and the flame appears very thin.
The flat flame shown in Figure 32 can only be sustained if the mixture is supplied at a velocity below a limiting value that is determined by the rates of diffusion of energy and radicals ahead of the flame and of reaction within the flame.

The laminar flame propagation has been object of many studies starting from the one by Mallard and Le Chatelier [45] who suggested that conduction heating within the mixture up to the ignition temperature of the fuel controls the flame propagation. The equation describing this hypothesis is the following:

\[ f \cdot c_p \left( T_i - T_0 \right) = k \frac{dT}{dz} \]  \hspace{1cm} (10)

where \( k \) represents the thermal conductivity of the gas, \( T_i \) is the ignition temperature, \( T_0 \) is the initial temperature, \( f \) is the mass flux through the flame front. The mass flux is related to the speed \( S_L \) at which the flame propagates into the unburnt mixture and to the density \( \rho_0 \):

\[ f = \rho_0 S_L \]  \hspace{1cm} (11)

If the temperature profile between the initial temperature and the adiabatic flame temperature \( T_f \) is approximated as linear with the flame thickness \( \delta \) according to equation (12):
\[ \frac{dT}{dz} = (T_f - T_i)\delta \quad (12) \]

The flame speed can be calculated as:

\[ S_L = \frac{k}{\rho_0 c_p} \frac{(T_f - T_i)}{(T_i - T_0) \delta} \quad (13) \]

The flame thickness can be related to the flame speed and to the characteristic time for the combustion reaction. The characteristic reaction time can be defined, using the overall fuel oxidation rate \( r_f \), as:

\[ \tau_c = \frac{[\text{fuel}]_0}{r_f} \quad (14) \]

The flame thickness therefore, becomes:

\[ \delta = S_L \tau_c = \frac{S_L [\text{fuel}]_0}{r_f} \quad (15) \]

Rearranging:

\[ S_L = \left[ \frac{k}{\rho_0 c_p} \frac{(T_f - T_i)}{(T_i - T_0) [\text{fuel}]_0} \right]^{1/2} \quad (16) \]

The reaction rate is a non-linear function of temperature as an exponential term is present. Although the reaction may begin at lower temperature, great part of the reaction takes place after the gases have been heated nearly to the adiabatic flame temperature, which is also the final temperature. The activation energies for the combustion of most hydrocarbon fuels are similar, as are the adiabatic flame temperatures and the thermal conductivities of the fuel-air mixtures. The flame temperature is highest near stoichiometric combustion and drops significantly at lower or higher equivalence ratio.

As mentioned, being the flame speed (and combustion) highly dependent on temperature, heat losses play an important role in sustaining the combustion. To have a better picture of the heat losses in a combustion chamber the scheme in Figure 33 can help. As it can be seen the heat transfer mechanisms are different and complex and they have great effects on the combustion products temperature and on the sustainability of combustion. Convection and radiation are the main heat transfer modes that can be found in such a simple system, however axial conduction through the walls has also a great importance as, depending on the gradient of temperature, heat can be “recirculated” upstream of the flame holder and contribute to preheat the reactant mixture.
In this thesis experimental and numerical investigations are carried out in order to gain deeper understanding of these mechanisms.
2.16 Summary

In this chapter a review of the researches focusing on Ultra Micro Gas Turbines worldwide and on fuel blurring injection was presented. Every paragraph introduced and described the work carried out by the different research groups, along with the results and suggestions for future developments. This review also highlighted the gaps found in literature, therefore suggesting the scope of the present work.

The researches listed do not seem to highlight the issues related to heat losses from the combustion chamber nor comprehensive studies on heat performances on heat recirculating combustion chambers were previously performed. Increasing the temperature of the fuel-air mixture has benefits in widening the flammability limits and also allows higher flow rate in the combustion chamber. For this purpose, a design to improve heat recovery, using a further recirculation of combustion products, could be introduced and tested in order to achieve higher preheating temperatures of the reactant mixture. A comparison in terms of thermal performances between an insulated and a non-insulated combustion chamber could be useful to show the pros and cons of limiting heat losses from the combustion chamber.

Studies on premixed combustion of high viscosity fuels, such as vegetable oils, at small scale are not present in literature therefore investigations is necessary.
UMGT AND EXPERIMENTAL MICROCOMBUSTOR DESIGN

As discussed in Chapter 2, the area of study of this research is related to Micro-Electro-Mechanical Systems (MEMS) and in particular to the branch represented by power generation devices able to produce power outputs up to 1 kW. In the case of the UMGT under investigation the power output ranges between 1 W and 50 W, thus including a variety of small portable applications, emergency backup devices, drones, micro-cooling and in general all those instruments and systems requiring a small amount of electric power. Although small scale combustion chambers can be coupled with other electric power producing devices such as thermophotovoltaic cells, in this research the combustion chamber was thought for utilization with an UMGT. In the following section the preliminary design of the compressor-turbine group under different possible operating conditions is presented, with the aim of defining parameters such as operating pressure, temperatures of the Brayton cycle, expected fuel and air flow rates and prediction of the power output. Also a comparative thermal analysis on bigger size systems will be performed to show the benefits of regeneration.

3.1 Brayton Cycle

The Brayton cycle was first proposed by George Brayton for use in the reciprocating oil-burning engine that he developed around 1870. Today it is used for gas turbines only, where both the compression and expansion processes take place in rotating machinery. Gas turbines usually operate according to an open cycle, as shown in Figure 34.

![Gas turbine scheme (open cycle)](image-url)
Air from the external environment at ambient conditions is drawn into the compressor, where its energy content (i.e. temperature and pressure) is increased. The compressed air flows into the combustion chamber, where the fuel is injected and the mixture burns isobarically, usually with excess air. The combustion products then flow into the turbine where they expand, producing work collected on the shaft of the compressor-turbine group (usually by an electric generator). The exhaust gases are then discharged to the atmosphere.

The gas turbine can be utilized also in a closed cycle configuration, where the two heat sources are replaced by two heat exchangers. In this case heat is exchanged at constant pressure, and the fluid operating in the engine is recirculated from the turbine outlet to the compressor inlet. This configuration is schematically shown in Figure 35.

![Figure 35 – Gas turbine scheme (closed cycle)](image)

In both cases the generic Brayton cycle is represented on the T-S diagram in Figure 36 and it is made up of one compression (1-2), one heat addition at constant pressure (2-3), one isotropic expansion (3-4) and one heat rejection at constant pressure (4-1).

![Figure 36 - Brayton Cycle on T-S diagram](image)
In reality the fluid evolution is not ideal, therefore losses have to be taken into account. In Figure 36 an example of a real Brayton cycle is also shown (blue line). During the compression the entropy of the real fluid increases generating a positive effect on the cycle efficiency as less energy input is required from the high temperature source (e.g. combustion of fuel). During the combustion there are friction losses in the combustion chamber that contribute to a pressure drop (point 3’). Also the combustion efficiency needs to be considered, as the fuel calorific value is not entirely transformed into heat. Finally during the expansion of the combustion products in the turbine entropy increases considerably (point 4’) and heat is lost from the turbine.

The highest temperature in the cycle occurs during the combustion process (state 3), and it is only limited by the maximum temperature coupled with tensile stress that the turbine blades can withstand. Excessive stress on turbine also sets a limit on the pressure ratio that can be used in the cycle and on the net work output so a compromise is needed.

The air flow in gas turbines has two essential functions: it supplies the necessary oxidant for the combustion of the fuel and it serves as a coolant to keep the temperature of various components within safe limits. The second function is achieved by drawing into the compressor a larger amount of air than the stoichiometric quantity needed for the combustion. Usually gas turbines operate with a considerable excess of air, with an air to fuel ratio around 30 or above. Therefore, if the mixture flowing into the gas turbine group is approximated as only air, the error caused in the calculations by neglecting the presence of the fuel is not relevant.

The two major application areas of gas-turbine engines are aircraft propulsion and electric power generation. In the first case the turbine produces just enough power to drive the compressor and a small generator to power the auxiliary equipment. The propulsion is achieved by the exhaust gases flowing through the nozzles at high speed, providing the required thrust. They are also used as stationary power plants to generate electricity as stand-alone systems or in conjunction with steam power plants. In these plants the exhaust gases of the gas turbine (at temperatures around 800 - 1000 K) serve as heat source for the steam to evaporate. The gas turbine can also be operated in a closed cycle configuration in nuclear power plants. In this case the working fluid is not air, and a gas with more desirable characteristics can be employed (e.g. helium).

In gas turbine power plants, the ratio of the compressor work to the turbine work, called the back work ratio, is considerably high. Usually more than one-half of the turbine work
output is used to drive the compressor. The situation becomes even worse if the isentropic efficiencies of compressor and turbine are low. This is in contrast with steam power plants, where the back work ratio is only a few percent. In this case a liquid is compressed instead of a gas and the steady flow work is proportional to the specific volume of the working fluid. A power plant with a high back work ratio needs a larger turbine to provide the additional power requirements of the compressor, so the turbines used in gas turbine power plants are larger than those used in steam power plants considering the same net power output.

### 3.1.1 Improvements to the Brayton cycle

Since their first successful development in the 1930s the gas turbines experienced extraordinary growth and advances. In those years turbines had efficiencies of around 17% because of low compressor and turbine efficiencies and low turbine inlet temperatures due to metallurgical limitations. In the years that followed the development of turbines focused on different aspects:

- **Increasing turbine inlet temperature.** This has been the primary approach taken to improve the cycle efficiency. The turbine inlet temperatures have increased dramatically from about 540 °C in the early years to 1400 °C and even higher today. These increases were made possible by the development of new materials and the innovative cooling techniques for the critical components, such as coating the turbine blades with ceramic layers and cooling the blades with the discharge air from the compressor. Maintaining high turbine inlet temperatures with an air cooling technique requires the combustion temperature to be higher to compensate for the cooling effect of compressed air. On the other side, higher combustion temperatures increase the amount of nitrogen oxides (NOx), which are responsible for the formation of ozone at ground level and smog. Using steam as the coolant allowed an increase in the turbine inlet temperature without increasing the combustion temperature. Steam is also a much more effective heat transfer medium than air.

- **Increasing the efficiencies of turbomachinery components.** The performance of early rotating equipment suffered greatly from the inefficiencies of turbines and compressors. However, the advent of computers and advanced techniques as computer-aided design improved the aerodynamic design of these components containing to a minimum fluid dynamic losses. Also advances in manufacturing
technologies and in materials development contributed to increase the components’ efficiency and subsequently, the cycle efficiency.

- **Modifications to the basic cycle.** The efficiency of the simple Brayton cycle can be greatly increased by implementing intercooling, regeneration and reheating techniques. These improvements come at the expense of increased initial and operational costs and they cannot be justified unless the decrease in fuel costs offsets the increase in other costs. In the past, the base-load electric power generation was dominated by large coal and nuclear power plants. However there has been a historic shift towards natural gas-fired turbines because of their higher efficiencies, higher regulation flexibility, lower capital costs, shorter installation times, better emission characteristics and the abundance of natural gas supplies. As a result, gas turbines are used to satisfy power peak requests and also small decentralized applications are rapidly growing.

### 3.1.1.1 Regenerative Brayton cycle

In gas turbine engines, the temperature of the exhaust gas leaving the turbine is often considerably higher than the temperature of air leaving the compressor. Therefore the compressed air can be heated by recovering heat from the hot exhaust gases through a heat exchanger, also known as regenerator or recuperator.

![Regenerative Brayton cycle](image-url)
In Figure 37 a scheme of the regenerative Brayton cycle is shown. The thermal efficiency of the Brayton cycle increases as a result of regeneration since the portion of energy of the exhaust gases that is normally rejected to the surroundings is now used to preheat the air entering the combustion chamber. This, in turn, decreases the energy input (the fuel) requirements for the same net work output. Obviously, the use of a regenerator is acceptable only when the temperature of exhaust gases is higher than the temperature of air leaving the compressor, otherwise the combustion products will be heated, decreasing the efficiency. This situation does not occur very often, as it can happen only with engines operating with an elevated pressure ratio. The temperature of the combustion products entering the recuperator is $T_4$. In the ideal case, the air exits the regenerator at the same temperature ($T_4$) the exhaust gases enter it. The final regeneration temperature will always be lower than $T_4$, depending on the exchanger efficiency.

3.1.1.2 Brayton cycle with intercooling, reheating and regeneration

The net work of a gas turbine system results from the difference between the work produces by the turbine and the work absorbed by the compressor. The energy necessary to compress a gas between two specified pressures can be decreased by carrying out the compression process in different stages and cooling the gas between them. This is called multistage compression with intercooling and as the number of stages is increased, the compression work decreases. Similarly, the output of a turbine operating between two pressures can be increased by expanding the gas in stages and reheating it between them (multistage expansion with reheating). This is accomplished without raising the maximum temperature in the cycle. As the number of stages increases, the process becomes nearly isothermal. The general principle this scheme is based on is that the steady flow compression or expansion work is proportional to the specific volume of the fluid. Therefore, the specific volume of fluid should be as low as possible during a compression process and as high as possible during an expansion process. Figure 38 represents the scheme of the modified gas turbine system.
Generally gas turbines operate with a high excess of air to avoid excessive temperatures so exhaust gases are rich in oxygen. Therefore, reheating can be done by injecting more fuel into the exhaust gases between two expansion states and igniting the mixture. This makes regeneration more attractive since a greater potential exists. The T-S diagram of an ideal two stage gas turbine cycle with intercooling, reheating and regeneration is shown in Figure 39.

Air enters the first stage of the compressor at point 1, it undergoes a first compression at intermediate pressure (point 2) and then it is cooled until point 3. Afterwards it is compressed again to the highest pressure of the cycle (point 4). A first heating occurs in the regenerator and the second heating is represented by the combustion of fuel (from point 5 to 6). The
burned gas enters the first stage of the turbine and expands to state 7, where it enters the reheater. After the second heating it expands again in the low pressure stage of the turbine until point 9 where it enters the regenerator. Finally the gas is cooled again to the initial state, or discharged into the atmosphere.

3.2 Constraints

There are several limitations to the development of a UMGT so that an optimal combination of the operating parameters and component dimensions is necessary to minimize losses.

The first requirement of an UMGT is obviously related to containing dimensions within certain limits. The conventional gas turbine systems are usually composed of multistage compressors and turbines but such configuration is not desirable in a micro-scale device, thus, to maximize the compression ratio and the compactness, a radial compressor and a radial turbine are the optimal choices to satisfy this request. With this type of solution the compression ratio can reach values of up to 4. The biggest constraint of a micro-scale gas turbine is related to the turbine material. In fact, to generate the required specific power with a single stage machine it is necessary to reach turbine inlet temperatures ranging from 1300 K to 1600 K [9,16]. The impossibility at this stage to implement a cooling system for the blades sets a limit on this value. In particular, in conventional size machines, cooling air flows internally through the blade and removes heat from the surfaces. Obviously this cannot be achieved in small scale turbines where dimensions of blades are in the order of millimetres or less. In addition to that, the centrifugal stresses, as well as the specific work, depend on the square of velocity and, since these values can easily reach up to 600 m/s, they can be in the order of hundreds of MPa. Higher peripheral speeds can also lead to fluid dynamic shocks due to exceeding the speed of sound (especially in the inlet section of the compressor), thus resulting in unacceptable losses. So far studies related to UMGT have used the same design techniques and fluid dynamics concepts as the conventional machines, but, as pointed out in the literature [9], the adaptability of these models to micro-scale devices is not certain therefore establishing an optimal design area is needed. An important constraint in the design procedure is the adoption of 2D extruded geometries for the blades, using the photolithography techniques, now widely used for the production of Micro Electro-Mechanical Systems. Developments in such manufacturing technologies are currently in
progress and more complex 3D geometries can be produced. Reynolds number in small devices is lower than in conventional machines and considering the strict constructive limits, realizing a geometry avoiding separation of the flow will be challenging. Furthermore, the flow is usually guided through the channels of the rotor by the gentle curvature of the blades in order to avoid separation and losses due to interaction between the boundary layer and curved surfaces. In an UMGT this process could be constructively difficult or very expensive. A solution to that is represented by reducing the gap between the rotor and the case diminishing the ventilation losses; on the other hand this could result in increasing resistance losses so a compromise is required.

Usually, considering conventional size gas turbines, assumptions on adiabatic flow are not so inappropriate as the amount of heat transferred is generally low (less than 0.5 % of thermal energy available at the turbine inlet). Internal and external heat transfers are therefore neglected and their influence is not taken into account [19]. Micro gas turbines and ultramicro gas turbines cannot be studied with the same assumption otherwise the inaccuracy could be very high. During operation, heat is transferred from the turbine to the outside, to the bearings, to the casing and to the compressor, thus heating the compressor and leading to a drop in turbine performances (as the compressor requires more work to increase air pressure). Consequently operational maps need to be reviewed and corrected for micro-scale applications.

### 3.2.1 Materials

When considering materials for a UMGT the biggest issues arise for the turbine blades. The properties to be considered are the material absolute specific strength and its resistance to thermal shocks at high temperatures, creep, oxidation and the degradation characteristics over fatigue cycles under operating conditions. At the present time, silicon alloys seem to be the most interesting candidate for the realization of turbine blades as they are able to operate at elevated temperatures without cooling. The only weakness of these alloys (and of all the ceramic materials) is the fragile behaviour that made them undesirable for large-scale as well. In fact the inner imperfections can be present in an amount that even a small mechanical stress could lead to a propagation of a crack. In an UMGT dimensions are reduced so that one single piece contains a low number of grains and this problem could be better controlled.
Metals and ceramic materials do not have the required properties to operate at high temperature and high centrifugal stresses at the same time.

Two of the most suitable materials for this type of application are SiC and Si₃N₄ and their characteristics are shown in Figure 40 [46].

According to the ASM International Materials Handbook [47], the sintered silicon nitride (Si₃N₄) tested at room temperature has a tensile strength ranging from 600 to 1200 MPa and at 1200 °C the tensile strength ranges from 500 to 800 MPa. The density of the material is around 3200 kg/m³ and the thermal conductivity at 20 °C ranges from 10 to 55 W/mK. The silicon carbide has a similar density (3300 kg/m³) but the tensile strength is lower than Si₃N₄ and at room temperature ranges from 70 to 300 MPa while at high temperature (1200 °C) is about 250 MPa. The thermal conductivity at 20 °C ranges from 60 to 200 W/mK. Overall these two materials could be used in UMGT applications provided manufacturing techniques are further improved.

![Young's Modulus of Elasticity](image1)

![Flexural Strength vs. Temperature](image2)

Figure 40 - Properties of selected materials [46]
3.3 UMGT design parameters

As previously stated, the power output range considered in this research is from 1 W to 100 W; the preliminary design of the UMGT is discussed next, including operating and constructive parameters of the system. All the constants and the physical properties of fluids and materials are taken from charts and tables found in the literature.

Since one of the purposes of the research is to maintain UMGT scales, and therefore dimensions within certain limits, radial configurations are chosen for both the compressor and the turbine and a diameter of 10 mm is set for both compressor and turbine. The regenerative combustor will not be integrated in the turbomachinery group, as in conventional size gas turbine machines, but it will be studied in the following sections as the main focus of this study. There are a few variables that need to be set to perform the sizing of the machine which can regard the peripheral speed, turbine inlet temperature and compression ratio. In this case the peripheral speed chosen is 200 m/s and the turbine inlet temperature (TIT) is set at 1350 K as these are feasible values [9] given the current state of art of materials and the impossibility of cooling the turbine blades. The selected speed was chosen in order to maintain subsonic conditions at the channel exit section of the compressor, which is the most critical. Different compression ratios and different air mass flow rates have been considered to entirely cover the power range set previously. To generate the desired power output mass flow rates range from 0.04 to 0.5 g/s and compression ratios range from 1.8 to 3.5. In conventional size single stage compressors a compression ratio of 3.5 represents the upper limit that prevents supersonic flow speed exiting the compressor rotor, thus maintaining an acceptable efficiency of the diffuser. Higher values result in shocks that cause flow irreversibilities and efficiency decrease.

To define the maximum peripheral speed of the rotating components, appropriate materials have to be chosen and in this case the Ti-6Al-4V alloy (maximum tensile strength 880 MPa) will be chosen for the compressor (since it is typically used in conventional compressor blades) and Si₃N₄ (maximum tensile strength at 1350 K is 425 MPa) will be considered for the turbine blades [47].

3.3.1 Parametric study of the thermal cycle

In this section the transformations of the operating fluid and the performance of the thermal cycle with reference to Figure 36, are analysed.
In this analysis, polytropic efficiencies of compressor and turbine, mechanical efficiency and several constructive parameters are based on typical values of conventional machines [48] and corrected according the modified Balje maps by Capata and Sciubba [49] for low Reynolds numbers.

The compressor is the first component to be designed. Equation (17) describes the evolution of the fluid during compression and expansion phases and allows the calculation of the final temperature of an ideal gas after the compression or expansion in the ideal case:

\[ T_2 = T_1 (\beta)^{\frac{k-1}{k}} \]  

and in the actual case:

\[ T_2 = T_1 (\beta)^{\frac{k-1}{k \eta_p}} \]  

where \( k \) is the ratio between specific heat at constant pressure and specific heat at constant volume and \( \eta_p \) is the polytropic efficiency of the transformation. The Eulerian work, the power required by the compressor and the compressor isentropic efficiency can then be evaluated.

\[ W_c = c_{p2} T_2 - c_{p1} T_1 \]  

\[ P = \frac{\dot{m} W_c}{\eta_m} \]  

\[ \eta_c = \frac{T_{2i} - T_1}{T_{2r} - T_1} \]  

The constructive parameters are obtained from similitude considerations, in particular using Balje diagrams and Lewis maps for compressors.

The method is based on utilizing dimensionless numbers including the specific rotational speed \( n_s \) and specific diameter \( d_s \) to scale down the machine:

\[ n_s = \frac{\omega \sqrt{Q}}{W^{3/4}} \]  

\[ d_s = \frac{D W^{1/4}}{\sqrt{Q}} \]  

Balje diagrams are similar to the one shown in Figure 41 and they have been obtained experimentally for every type of rotating equipment.
Once the rotational speed has been set, all the other constructive data follow easily and it is possible then to obtain the velocity triangles at the inlet and at the outlet sections with simple geometric relations. The only constraint in this case is the sonic speed at the inlet section which limits the maximum relative speed at the tip of the blades. However, this condition is respected for the compression ratio range considered (from 1.8 to 3.5) as the Mach number is always lower than 0.9 at that section. In the sizing procedure it is also necessary to conveniently choose the flow coefficient, defined as $\varphi = \frac{v_{ax}}{u}$ (ratio between axial component of velocity and peripheral speed). The number of blades is defined with the relation:

$$n_{blades} = \frac{2}{(1 - \sigma_s)}$$  \hspace{1cm} (24)

where $\sigma_s$ is the slip factor assumed to be 0.85 [48]. This results in 13 blades for the compressor and between 6 and 8 for the turbine (in a compressor the flow needs to be “guided” from the inlet to the outlet to limit the losses due to the effect of the pressure gradient).

The procedure doesn’t change for the turbine, and the only limit is imposed by the material of the blades which are subject to extreme thermal and mechanical stresses. The tensile stress is thus calculated using Equation (25):
\[ U_{\text{max}} = k_s \sqrt{\frac{2\sigma}{\rho(1 - \chi^2)}} \]  

(25)

where \( \sigma \) is the material tensile strength, \( k_s \) is a strength coefficient, \( U_{\text{max}} \) is the maximum peripheral velocity and \( \chi \) is a coefficient usually ranging from 0.65 to 0.85.

The fuel used in these preliminary considerations is methane and a lower heating value of 50 MJ/kg has been taken. The thermal efficiency of the Brayton cycle can therefore be calculated once the fuel to air ratio \( FA \) and fuel mass flow rate \( \dot{m}_{CH_4} \) are obtained:

\[ FA = \frac{c_{p3}T_3 - c_{p2}T_2}{LHV \eta_b - c_{p3,CH_4}T_3} \]  

(26)

\[ \eta = \frac{P_{\text{net}}}{LHV \dot{m}_{CH_4}} \]  

(27)

To calculate the efficiency of the regenerated cycle, different regeneration grades (ratio between heat transferred to the reactants and heat contained in the combustion products) have been chosen, and the correspondent efficiency has been evaluated.

From Figure 42 to Figure 48 the results of the preliminary calculations are shown. The values are referred to 0.125 g/s air mass flow (which corresponds to a power output in the range 10 to 15 W) and they show the efficiency of the simple Brayton cycle with varying compression ratio, the temperature of the exhaust gases vs the compression ratio, the change in efficiency with different regeneration grades and the benefits of regeneration in terms of fuel consumption.
Figure 44 – Predicted electric power output vs mass flow rate – 3.5 bar pressure ratio

Figure 45 – Effect of regeneration on efficiency at different pressure ratios

Figure 46 – Effect of regeneration on fuel consumption at different pressure ratios – 0.125 g/s flow rate

Figure 47 – Effect of regeneration on reactants’ temperature at different pressure ratios – 0.125 g/s flow rate

Figure 48 – Effect of regeneration on fuel consumption at different pressure ratios - 1.5 g/s flow rate
Obviously, the cycle efficiencies and the temperature at the end of the expansion do not change with different flow rates, as they depend on the compression ratio.

As can be seen from the figures, the fuel consumption is considerably reduced as the regeneration grade increases thus resulting in an efficiency increase. Considering the two extreme situations, without regeneration and with R= 0.9, the efficiency is almost doubled, so the benefits of employing regeneration is certain. A high regeneration grade however, is not feasible as the reactants may reach the autoignition temperature and ignite before reaching the combustor therefore a compromise is needed, also considering the available heat exchange surfaces.

The last case presented (with an air mass flow of 1.5 g/s - Figure 48) has been validated by comparing to the results obtained from a demo version of a thermal cycle simulation software developed by Delft University of Technology named CycleTempo [51]. This program uses the extended thermodynamic library FluidProp [52] to compute cycle parameters.

The first case considers an air mass flow rate of 1.5 g/s and a compression ratio of 2.5. In the schemes shown in Figure 50 to Figure 53 the devices are represented and also the values obtained from the thermal cycle simulation are shown at different points of the cycle. The parameters are related to the components’ performances (turbine and compressor in particular) and also the thermal properties of the operating gas are shown at different points of the system (in particular pressure expressed in bar, temperature expressed in °C, enthalpy in kJ/kg and mass flow rate in kg/s - rounded at the 3rd decimal figure). A legend is also provided, in Figure 49.

As can be seen from Figure 50, the incoming flow is simulated by a mass source upstream of the compressor (for air) and another one connected to the combustor (for fuel), a heat sink has been introduced to set the turbine inlet temperature to 1350 K and a smoke stack is used to simulate the exhaust gas discharged into the atmosphere. Typical values of efficiencies related to the components were considered [16].
Figure 49 - Legend

Figure 50 - Simple Brayton cycle - 2.5 bar, 1.5 g/s air mass flow rate
Figure 51 – Regenerative (R=0.2) Brayton cycle - 2.5 bar, 1.5 g/s air mass flow rate

Figure 52 - Simple Brayton cycle - 3.5 bar, 1.5 g/s air mass flow rate
The results of the CycleTempo simulations confirm the validity of the calculations shown previously. The only differences in the values can be attributed to approximations related to system efficiencies such as mechanical, electrical and polytropic efficiency. Table 2 summarizes the values of efficiency obtained from the simulations and from the theoretical calculations, for the four cases presented.

Table 2 - Summary of results

<table>
<thead>
<tr>
<th></th>
<th>Thermal efficiency [%]</th>
<th>Power output [W]</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Theoretical Calculations</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.5 bar simple</td>
<td>6.7</td>
<td>149</td>
</tr>
<tr>
<td>2.5 bar R=0.2</td>
<td>7.57</td>
<td></td>
</tr>
<tr>
<td>3.5 bar simple</td>
<td>8.88</td>
<td>187</td>
</tr>
<tr>
<td>3.5 bar R=0.2</td>
<td>9.86</td>
<td></td>
</tr>
<tr>
<td><strong>CycleTempo simulations</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.5 bar simple</td>
<td>6.64</td>
<td>150</td>
</tr>
<tr>
<td>2.5 bar R=0.2</td>
<td>7.4</td>
<td></td>
</tr>
<tr>
<td>3.5 bar simple</td>
<td>8.84</td>
<td>190</td>
</tr>
<tr>
<td>3.5 bar R=0.2</td>
<td>9.69</td>
<td></td>
</tr>
</tbody>
</table>
3.4 Combustion chamber

The flat flame combustion chamber design is a major challenge at microscale as there are several constraints involved. Combustion is regulated by the Damköhler number ($Da$) which is defined as the ratio between the residence time of the mixture in the combustion chamber and the reaction time.

$$Da = \frac{\tau_{\text{residence}}}{\tau_{\text{reaction}}}$$

(28)

This dimensionless number indicates whether the combustion is complete or not. The residence time is the ratio between the volume of the combustion chamber and the mass flow rate and the reaction time is given by the reaction rate of the fuel. Obviously, to achieve complete combustion in the given volume, $Da$ must be higher than 1. To do this, either the volume can be increased, giving the mixture enough time to completely burn, or the reaction rate can be increased, shortening the combustion time. The first option is not feasible at this scale, as one of the requirements of UMGT is to contain overall dimensions; the second option can be achieved by preheating the reactants so that the combustion occurs more rapidly.

Furthermore, as previously mentioned, heat losses through lateral surfaces play an important role in the stability of the flame and in the worst case they can lead to the quenching of the flame. In addition, the necessary condition for having stable flat flame on the surface of a flame holder is that the incoming flow velocity be lower than the laminar flame speed, otherwise the flame will blow off.

The laminar flame speed for methane-air mixtures at atmospheric pressure depending on the initial reactants temperature can be found from the literature and, for the design point chosen for the present work (0.65 equivalence ratio and operating pressure ranging from 1.8 to 3.5 bar), it results in a range from 11.5 to 14.3 cm/s. This value has been obtained from the correlation proposed in [53] for natural gas, and although the equivalence ratios considered range from 0.7 to 1.2, it will be taken as a reference.

$$\frac{u_i}{u_{i0}} = \left( \frac{T_u}{T_{u0}} \right)^{a_T} \left( \frac{P_u}{P_{u0}} \right)^{b_P}$$

(29)

In this equation the references are $T_{u0} = 300 \, K$ and $P_{u0} = 0.1 \, MPa$ and
\[ \alpha_T = 5.75 \phi^2 - 12.15 \phi + 7.98 \]  
\[ \beta_p = -0.925 \phi^2 + 2 \phi - 1.473 \]  
\[ u_{l0} = -177.43 \phi^3 + 340.77 \phi^2 - 123.66 \phi - 0.2297 \]

The initial values of reactants temperature are calculated using the Poisson’s equation related to polytropic compression (it ranges from 360 K at 1.8 bar to 450 K at 3.5 bar). Preheating the reactants affects the laminar flame speed, as can be seen from Equation (29), and this allows increasing the mass flow rate in the combustion chamber, thus the heat release rate and the power generated by the UMGT. For the present study the diameter of the combustion chamber will be 18 mm, with a minimum mass flow rate of 0.04 g/s.

The size of the inner set of fins where reactants flow has been set in order to prevent flame flashback through it. Therefore the cross sectional area of the channel has been set at 1.5 x 2.0 mm. With these dimensions the flow velocity is around 7 m/s (with reactants’ flow rate of 0.04 g/s), which is much higher than the laminar flame speed at the operating conditions so that flame flashback would not occur. The dimensions of the outer fin are set at 2 x 3.5 mm to allow the flow to discharge smoothly without affecting the combustion.

A simplified scheme of the regenerative combustor (not to scale) is shown in Figure 54.

![Figure 54 – Scheme of the regenerative combustor (not to scale)](image-url)

As can be seen from the scheme in Figure 54, the system is composed of one cylinder and two annuli: in the inner tube the combustion occurs, stabilized on the flame holder; the
mixture of compressed air and fuel flows into the middle annulus receiving heat from both the lateral surfaces; the exhaust gases from the turbine flow through the outer annulus and are discharged into the environment. The systems can be approximated as two heat exchangers coupled between them: the inner tube and the middle annulus operate on countered flows; the two annuli operate on parallel flows. Such arrangement allows the compressor-turbine group to be easily installed with the turbine inlet placed as close as possible to the combustor exit, and with the compressor outlet next to the middle fin, minimizing the heat losses to the environment.

### 3.5 Porous medium and flow analysis

The porous medium (PM) is an important component of the present combustion chamber as it is necessary to stabilize the flame on its surface and achieve flat flame combustion. The parameters to be set are porosity, defined as the fraction of the volume of voids over the total volume, and the length. The studies carried out by Turkeli-Ramadan [11,12] at the University of Auckland will be taken as references. In the above mentioned papers the behaviour of different flame holders was studied under several operating conditions. The main aim was to investigate the combustion characteristics adopting several flame holders made of different materials, porosities and thicknesses. Results showed that the most suitable material for the flame holder was sintered steel, as it has the necessary strength to resist thermal stresses. On the other hand alumina ceramic flame holders cracked after a short period of time during the combustion tests.

Another essential constraint in the choice of the flame holder is the pressure loss of the mixture flowing through it. The present study has different incoming flow condition compared to [12], as the incoming flow enters the combustion chamber radially. In addition to that the PM has also an important role in mixing the reactants. In order to obtain a uniform distribution of the flow in the combustion chamber, a preliminary analysis of the flow needs to be carried out to set the thickness of the PM. For this purpose the commercial Ansys Fluent [54] CFD tool will be used.

Initially the tool has been validated by comparing the experimental results of the 3 mm thick sintered steel PM pressure drop obtained in [12] with the numerical results from the simulation. A 3D model is used for the simulation with diameter equal to the test rig used in the above mentioned paper (46 mm), length equal to 120 mm and the flame holder considered.
is the 3 mm thick sintered steel PM. The zone was represented as a 3 mm thick volume in the fluid domain.

The parameters describing the fluid dynamic behaviour of the porous plate, as viscous and inertial resistance, were obtained following the procedure described in the Fluent Manual [55]. Using the results shown in the pressure drop vs incoming velocity plot in [12], it was possible to obtain the requested values. Fluent models porous media by adding a momentum source term $S_i$ to the standard flow equations [55]. This term is composed by two parts: a viscous loss term and an inertial loss term:

$$S_i = - \left( \sum_{j=1}^{3} D_{ij} \mu v_j + \sum_{j=1}^{3} C_{ij} \frac{1}{2} \rho |v| v_j \right)$$

(33)

where $v_j$ represents the velocity components in the x, y and z directions, $\mu$ the dynamic viscosity and $D_{ij}$ and $C_{ij}$ prescribed matrices. In the case of simple homogeneous media the equation becomes:

$$S_i = - \left( \frac{\mu}{\alpha} v_i + C_2 \frac{1}{2} \rho |v| v_i \right)$$

(34)

where $\alpha$ is the permeability and $C_2$ is the inertial resistance factor.

A mesh sensitivity analysis was first performed using 6 different element sizes: 0.002 m, 0.0015 m, 0.0012 m, 0.0009 m, 0.0007 m and 0.0005 m. The number of elements composing the grid in the six cases was respectively 55,614, 125,662, 244,188, 413,775, 1,059,279 and 2,430,972. The average element quality ranged from 0.8573 to 0.93, the
average skewness ranged from 0.07 to 0.1348 and the average orthogonal quality ranged from 0.799 to 0.99. The fluid used was air at 300 K temperature at atmospheric pressure. An incoming velocity of 0.5 m/s was used for this analysis and the results (in Table 3) showed very little differences, in terms of pressure drop.

<table>
<thead>
<tr>
<th>Element size [m]</th>
<th>Pressure drop [Pa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0005</td>
<td>299.257</td>
</tr>
<tr>
<td>0.0007</td>
<td>299.227</td>
</tr>
<tr>
<td>0.0009</td>
<td>297.186</td>
</tr>
<tr>
<td>0.0012</td>
<td>297.158</td>
</tr>
<tr>
<td>0.0015</td>
<td>297.139</td>
</tr>
<tr>
<td>0.002</td>
<td>297.098</td>
</tr>
</tbody>
</table>

Table 3 - Results of sensitivity analysis

The difference in pressure drop from the different mesh configurations studied is very small so the 0.0007 m mesh was chosen for the simulations to minimise computational cost / time.

Four simulations with four different inlet velocity conditions were executed: 0.5 m/s, 1 m/s, 1.5 m/s and 2 m/s. The boundary conditions set in Fluent were: velocity inlet and pressure outlet for the two ends of the domain, 3 mm porous zone representing the sintered steel insert and walls with no slip condition. The parameters related to the porous medium obtained from the experimental data were 145,056 1/m (inertial resistance) and 8,658,902,131 1/m² (viscous resistance). The convergence criteria were based on the residuals and the convergence target (normalized RMS) residuals were set to $10^{-6}$ for all the equations involved. Convergence was reached in all cases after about 200 iterations.

In Figure 56 and Figure 57 the contours of pressure and velocity for an incoming flow velocity of 0.5 m/s are shown.
The pressure drop across the PM for the four inlet conditions are listed in Table 4.

Table 4 - Pressure drops across the PM at different incoming flow velocities - CFD results

<table>
<thead>
<tr>
<th>Incoming velocity [m/s]</th>
<th>Pressure drop [mbar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>2.97</td>
</tr>
<tr>
<td>1</td>
<td>7.25</td>
</tr>
<tr>
<td>1.5</td>
<td>12.89</td>
</tr>
<tr>
<td>2</td>
<td>19.93</td>
</tr>
</tbody>
</table>

The results obtained from the CFD simulations match closely with those reported in [12] in terms of pressure drop compared to the sintered steel PM (R200t3), as can be seen in Figure 58.
After successfully validating the CFD tool against the experimental results, the study focused on the design of the combustion chamber and in particular of the porous media and the flow in the inner tube. The purpose of the following analysis was to set the dimensions of the flame holder in a way that the flow downstream of the PM was as uniform as possible.

In this 3D CFD study the actual operating conditions were used and different thicknesses of porous media were simulated.

Since porous media with different thicknesses (maximum 10 mm) will be employed in the present experimental rig, a series of simulations was performed using the same boundary conditions but varying the porous insert domain’s thickness. The results in terms of pressure drop of the 10 mm thick PM are shown in Figure 59.

One of the requirements during the design of a flat flame combustion chamber system is that the pressure drop across the flame holder is contained within 5 % of the inlet pressure. In the 3 mm thick sintered steel PM (R200t3) case this condition is achieved for all the velocities considered. For the case considering a 10 mm thick PM the condition is valid for inlet velocities up to around 1.5 m/s, which correspond to a total mass flow rate of around 0.46 g/s, thus, according to the operating conditions previously set, it is widely acceptable.

Finally the uniformity of the flow downstream the PM was studied through CFD cold simulations. The fluid domain recreates the geometry of the actual combustor. The mass flow inlet is placed radially at the top end of the chamber having a diameter of 18 mm. The porous
media were simulated by slicing the domain with two planes and applying the physical parameters previously mentioned to the new zone.

![Image](image.png)

**Figure 60 - Fluid domain (PM 10mm thick)**

The inlet mass flow rate was set at different values: 0.04 g/s, 0.08 g/s, 0.12 g/s, 0.22 g/s, 0.3 g/s and 0.5 g/s and the velocity profile 3 mm downstream of the porous medium was examined for all the simulations. Atmospheric pressure was set (101,325 Pa) at the outlet and the physical properties of the porous media were the same as in the previous simulations. The residuals monitors were set at $10^{-4}$ for continuity and for the three velocity dimensions. Results showed that a 10 mm thick PM produces best results in terms of uniformity of the flow downstream when compared to the 8 mm and 7 mm thick PM, especially at high flow rates. The velocity profiles are shown in Figure 61, Figure 62 and Figure 63.
Figure 61 - Velocity profiles 3 mm below the PM - 7 mm thick PM

Figure 62 - Velocity profiles 3 mm below PM - 8 mm thick PM
Given the uniformity of the velocity profiles at different flow rates, the 10 mm thick porous medium will be employed as flame stabilizer for the combustion chamber. In terms of pressure drop, it is contained below 5% of the inlet pressure, thus satisfying the condition for micro scale combustors. The graph in Figure 64 shows the results of the CFD simulation regarding the pressure drops across the PM.

Figure 64 - Pressure drops across PM at different flow rates and different PM thicknesses
3.6 Summary

This chapter was completely dedicated to the design of the combustion chamber for the experimental part of the study. In the first section a description of the Brayton cycle and the improvements that can be applied to increase overall efficiency was presented. Subsequently the constraints and the issues in the design of a small scale combustion chamber were discussed and the procedure for designing conventional size compressors and turbines employing dimensional analysis was shown. This was necessary to estimate the power output of the device after setting the diameter of compressor and turbine at 10 mm. A parametric study of the cycle performances at different operating conditions was carried out, highlighting the thermal efficiencies obtained. To give an idea of the improvements of a regenerated Brayton cycle, different levels of heat recovery rates were considered and the results were compared to those of the simple Brayton cycle. The second part of the chapter was focused on the design of the combustion chamber, based on the compromise between residence time, laminar burning velocity and uniformity of the flow downstream of the porous medium. To support the design, a numerical study of the pressure loss across the porous medium was carried out, validating the results against experimental data. In addition to that, since the mixture was designed to flow radially into the combustion chamber upstream of the flame holder, a numerical study about the uniformity of the flow (therefore of the flame) downstream of the flame holder was also carried out with the ultimate purpose of determining the thickness of the porous medium.
4 EXPERIMENTAL METHODOLOGY

This chapter is dedicated to present the experimental details, including the apparatus, the equipment and the methodology used in this part of the research. The test rig was developed in the Thermofluids Laboratory at The University of Auckland.

4.1 Regenerative combustion chamber

The design process of the combustion chamber considered for this study was based off the analysis presented in Chapter 3, but also went through some modifications to satisfy the following requirements:

- A regeneration of the cold reactant mixture had to occur receiving heat from the combustion products;
- Flat flame had to be achieved using a porous insert;
- Allowance for instruments (thermocouples and ignition wire in particular) had to be considered;
- A sight glass to control any flame flashback upstream of the flame holder had to be designed and implemented;
- Appropriate connections at the inlet and outlet of the combustion chamber;
- A system able to recirculate the combustion products from the inner cylinder to the outer annulus;
- Adequate sealing between the parts.

The first part to be designed was the combustion chamber. Stainless steel was chosen and the results obtained in Chapter 3 were taken as reference for the sizing. Therefore the inner diameter of the cylindrical combustion chamber was set at 18 mm. To allow a proper mixing of air and gaseous fuel and to achieve flat flame combustion, a 10 mm thick stainless steel porous medium with average porosity in the range 0.49 to 0.54 was chosen, considering the acceptable pressure drop across it (evaluated in Chapter 3). A compromise between minimizing the overall dimensions and a smooth development of the flame downstream of the porous insert determined the length of the combustion chamber, which was set at 18 mm. The overall length of the inner cylinder was 30 mm. Therefore the effective volume of the combustion chamber resulted to be 4.58 cm³.
The outer wall of the cylindrical combustion chamber was machined to obtain a helicoidal channel where the cold mixture was supposed to flow. The channel had dimensions 1.5 x 2 mm (depth x height) for an overall length of 0.47 m. Pressure losses within the cold flow channel were estimated to be ranging from about 4500 Pa to 44000 Pa (using the experimental correlation in [56]), depending on the flow rate. This choice was determined in order to avoid flashback through this channel. In fact, it was determined that even at the minimum flow rate, the flow velocity would be much higher than the burning velocity of the mixture. The fuel-air mixture was designed to be injected axially in the channel through a 1.5 mm diameter hole. The preheated mixture would then exit the regenerator radially just upstream the porous medium. In Figure 65 a section and a 3D view of the assembled combustion chamber with the supporting flanges is shown, as well as a picture of the actual chamber.

The three cylinders were assembled by cold pressing them in order to avoid using screws or other types of supports to keep them together. At the top and bottom ends of the
combustion chamber two stainless steel flanges were designed to allow access for the necessary measuring instruments. The section in Figure 65 shows the inlet and the combustion products recirculation cavity in the bottom flange. In this cavity a K type thermocouple is inserted from a 0.5 mm hole on the side of the flange. From the view of the bottom flange, the square hole for the sight glass can be seen with two additional holes for screws supporting the glass. The top flange includes a connection for the exhaust gas analyser on the outlet, a round sight glass to check possible flame flashback upstream of the flame holder and a 0.5 mm hole for a K type thermocouple measuring the preheating temperature of reactants. Four holes were drilled at the flanges’ corners to allow bolts through them to keep the system together. Two additional K type thermocouples were installed at the outlet and at the inlet to measure exhaust gas temperature and cold mixture temperature before entering the combustion chamber. Also a 2 mm hole was drilled in the bottom flange to allow the insertion of the ignition wire and the B type thermocouple. Stainless steel parts were machined in the workshop at The University of Auckland.

4.2 Porous flame holder

The effect of the flame holder material on combustion phenomena has already been investigated in a previous study by Turkeli-Ramadan [12]. In this work sintered stainless steel was chosen as flame holder and it was provided by Siperm Tridelta Gmbh from Germany. A 10 mm thick sheet was machined to obtain a 19 mm porous disc. The physical characteristics of the sintered steel porous insert are taken from the manufacturer and shown in Table 5. Figure 66 shows the porous medium and the combustion chamber assembled together.

<table>
<thead>
<tr>
<th>Filter grade</th>
<th>Density $[g/cm^3]$</th>
<th>Porosity [%]</th>
<th>Laminar flow coefficient $[m^2] \times 10^{-12}$</th>
<th>Turbulent flow coefficient $[m^2] \times 10^{-7}$</th>
<th>Pore size [$\mu m$]</th>
<th>Tensile strength $[N/mm^2]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>R200</td>
<td>3.6 – 4.0</td>
<td>49 - 54</td>
<td>112</td>
<td>300</td>
<td>65</td>
<td>30</td>
</tr>
</tbody>
</table>
To avoid leakages and to make sure the mixture completely flows through the porous volume, a layer of high temperature resistant cement was applied on the lateral surface of the porous insert in order to occlude the pores on the curved surface.

4.3 Flow blurring injector

In the last session of experiments on the regenerative combustion chamber, the feasibility of using vegetable oil in this device is investigated. For the purpose an injector able to produce fine droplets (comparable to diesel droplets in the same injector [39]) even at ambient temperature, to increase the surface to volume ratio and therefore to ease heat transfer and vaporization, had to be used and installed on the combustion chamber. The injector employed is a recent discovery [36], as described in the literature review, and a similar version was built and machined by the workshop at The University of Auckland. The injector is made up of different parts and all of them are removable.
Figure 67 shows a 3D view and the long cross section of the flow blurring injector designed and built for the tests. As it can be seen the overall radial dimension is 15 mm, the length of the main body of the injector is 42.5 mm and the fuel tube was designed with an inner diameter of 1.2 mm. Two inlets are provided for the atomizing air and the primary air respectively. However in this study, the whole air flow rate was used to atomize the liquid fuel and the primary air inlet was sealed. The fuel supply was connected to the bottom end of the fuel tube through silicon tubing. The distance between the top end of the fuel tube and the casing was regulated in order to be lower than 0.25 times the nozzle diameter to allow the backflow of the atomizing air within the fuel tube to generate the first fuel breakup, as noted in [37,39]. The top end of the housing of the injector had a thread which allowed the installation on the bottom supporting flange of the combustion chamber.
During the experiments the combustion chamber was started with a mixture of LPG and air (flowing into the combustion chamber through the atomizing air inlet) to preheat the device and reach steady conditions. This also allowed the vaporization of liquid fuel in the inner channel thanks to the heat drawn from the combustion products. Once the conditions were considered stable enough, the peristaltic pump was started and the fuel injected. At the same time the LPG flow rate was gradually decreased until zero so that the combustion chamber was run exclusively on vegetable oil. For this study sunflower oil was chosen because of its general availability.

The design of the injector was replicated from previous researches where comprehensive investigations of atomizing properties using different liquids and comparison with air blast injector were carried out [36,37,39,57].
4.4 Mass flow controllers

To control the equivalence ratio of the mixture, two flow controllers were necessary for the gaseous fuel and for air respectively. The air flow rate was controlled by a Sierra Smart Trak2 Series 100 with a minimum mass flow rate setpoint of 0.001 g/s of air and 1% accuracy on the whole scale. The other flow controller was a Vögtlin Instruments AG Red-y Smart Series with an operating range from 100 ml/min to 6000 ml/min of air and accuracy of 1% on the whole scale. It was installed in the gas line connected to a LPG bottle with two valves at the inlet and at the outlet respectively.

![Figure 70 – Sierra Smart Trak2 Series 1000 (left) and Vögtlin Instruments AG Red-y (right)](image)

The Sierra mass flow controller allowed the flow regulation through commands and display on the casing. Several configurations were available in the flow controller depending on the type of gas to be measured. Air was included in one of the preset inputs. The Red-y flow controller was fully operated through dedicated software and it was calibrated to operate with air. Therefore a correction on the mass flow rate was applied, which was also based on the equivalence ratio obtained from the analysis of the exhaust gases. The theory and the correction procedure are explained in Appendix A1, as well as the working principle of the thermal mass flow meters. Downstream of the flow controllers the mixing of air and LPG happened in a wye having the outlet connected to the combustion chamber outlet through a silicon tube.
4.5 Thermocouples

The most important requirements of the temperature measuring equipment are the high precision needed and the necessity of fitting into the tiny channels of the regenerative combustion chamber. Four Omega K type thermocouples (Chromel – Alumel) were used to measure the temperature of the flow at different locations in the combustion chamber. The first thermocouple had a 2 mm diameter and it was placed downstream of the junction between the gas line and the air line to record the ambient temperature. The other thermocouples had 0.5 mm diameter and they were respectively located upstream of the porous flame holder to measure the preheating temperature, in the bottom flange where combustion products were recirculated to the outer channel and at the outlet of the combustion chamber. Also a 1.92 mm outer diameter B type thermocouple (Pt-Rh 30%, Pt-Rh 6%) with a ceramic sheath, provided by Servotech Instrumentation Ltd was employed to measure the axial temperature distribution in the combustion chamber. Wires had a diameter of 0.5 mm and terminated in an exposed junction. A sheathed thermocouple was preferred to an exposed wire thermocouple to increase the durability of the instrument, although this could cause disturbance to the flame when inserting it in the combustor; the thermocouple was inserted from the downstream end and this was deemed to minimise any disturbances. Flame characteristics and behaviour were not closely examined in this work as they were studied in a previous research by Turkeli-Ramadan [11,12].

Figure 71 - K type thermocouple

Figure 72 - B type thermocouple
The thermocouples were connected to a Campbell Scientific CR850 datalogger (Figure 73), operated through its dedicated software. The dedicated software had default configurations for the K type thermocouples so that the temperature corresponding to the output voltage was displayed. There was no calibration for the B type thermocouple, therefore NIST (National Institute of Standards and Technology) [58] tables were used to convert the output voltage into temperature.

Calibration of thermocouples was performed using a ISOTECH Jupiter 650B dry block high temperature calibrator able to reach up to 650 °C with a ±0.3 % accuracy. The results of the uncertainty analysis are shown in Appendix A5.
The B type thermocouple was calibrated up to 650 °C but a higher temperature dry block was not available therefore only a limited range of temperature was tested. However the manufacturer specified that the error falls under ANSI standard limit of errors (which is ±0.5 % over 800 °C) which are available on the NIST website [59].

Since the B type thermocouple was subject to very high temperature differences, radiation and axial conduction through the wires cannot be considered negligible, therefore a correction was applied to every measurement. The formula including these heat losses is shown in Appendix A4 [60].

### 4.6 Ignition device

To ignite the fuel-air mixture an energy source was required and the only practicable solution was to use a spark to start the combustion. To generate the spark the device shown in Figure 75 was used consisting of an ignition coil and a condenser, with connecting wires. The device is able to produce a spark on a gap greater than 5 mm, using a voltage of around 40 kV. The positive terminal (Figure 76) was simply a 2 mm diameter stainless steel spigot and it was inserted in the combustion chamber through the bottom flange at a distance of around 2 mm from the sintered steel porous flame holder and the negative terminal was in contact with the outer surface of the combustion chamber. To avoid the spark to “jump” from the ignition wire to the bottom flange, a ceramic sleeve (shown in Figure 77) was machined to electrically insulate the wire (visible in Figure 76).

![Ignition device](image)
As mentioned above, the bottom flange was designed in order to allow for the flame visualization through a sight glass. Given the extremely high temperature the material chosen was quartz and the machined windows were provided by Technical Glass Product Inc., Ohio, USA. The quartz window was secured in place through friction against a graphite gasket forced between the flange and the quartz. The coupling was secure enough to avoid leakages and to prevent the window from falling during operation at high temperature with thermal expansion of the flange.

The pictures were taken using a digital camera Canon PowerShot 200IS. To capture good images of the flame, reflections and sources of light had to be eliminated therefore a dark screen was arranged around the combustion chamber.
4.8 Peristaltic pump

To study the operation of the combustion chamber when fuelled with vegetable oil, a Masterflex L/S P07528-30 peristaltic pump with variable speed motor was employed. This model in particular is able to achieve extremely low flow rates, as low as 0.06 ml/min, as required in this case. The tubing used with the pump is silicone platinum-cured flexible tubing and its internal diameter is 1.6 mm.

![Masterflex peristaltic pump](image)

Figure 78 - Masterflex peristaltic pump

The pump was manually controlled through the commands on the front panel. The display showed the revolutions per minute of the motor, therefore a correlation to convert it to flow rate was necessary and this was done through the calibration of the instrument.
Calibration was performed using water at ambient conditions and the relation between flow rate expressed in ml/min and the rotational speed of the motor in rpm is shown in Figure 79. During the tests one end of the tubing was inserted into the vegetable oil tank and the other end was connected to the liquid inlet of the flow blurring injector.

4.9 Exhaust gases analysis

The composition of the exhaust gases flowing from the combustion chamber were collected and analysed using a Kane Automotive Auto 5-1 device able to measure the volumetric content of CO₂, CO, unburned hydrocarbons HC, oxygen, NOₓ and NO of a dry gaseous mixture. The calibration was made against a calibration gas of known composition and results showed a very accurate match. To eliminate the water contained in the flue gases a very simple but effective condenser was arranged. The condenser where the combustion products were directed was made up of a container with one inlet connection and one outlet connection, immersed in an ice bath. The container, other than the inlet had an outlet connected to the suction line of the Kane analyser which was able to sample 1 l/min of air at ambient temperature. As the mass flow rate flowing out the combustion chamber was always higher than the sampling flow rate of the analyser, a relief valve was also implemented to allow combustion products in excess to flow to the environment. The temperature of the combustion products flowing out of the container was constantly monitored by a thermocouple in order to adjust the readings conversion to compensate changes in gas density. However, for most of the time, this correction was not necessary as the temperature
was very close to the ambient temperature. In Figure 80 the Kane gas analyser is shown and in Table 6 its specifications and accuracy are listed.

![Figure 80 - Kane Analyzer Auto 5-1](image)

**Table 6 - Kane Automotive Auto 5-1 Analyzer specifications**

<table>
<thead>
<tr>
<th>Type of sensor</th>
<th>Resolution</th>
<th>Accuracy</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Carbon Monoxide</strong></td>
<td>Infrared</td>
<td>± 5% of reading</td>
<td>0-10%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>± 0.06% of volume</td>
<td>Over Range 20%</td>
</tr>
<tr>
<td></td>
<td>Fuel cell</td>
<td>± 5% of reading</td>
<td>0-21%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>± 0.1% of volume</td>
<td>Over-Range 48%</td>
</tr>
<tr>
<td><strong>Oxygen</strong></td>
<td>Infrared</td>
<td>± 5% of reading</td>
<td>0-5000 ppm</td>
</tr>
<tr>
<td></td>
<td></td>
<td>± 12 ppm</td>
<td>Over Range</td>
</tr>
<tr>
<td></td>
<td></td>
<td>± 0.1% of volume</td>
<td>10000 ppm</td>
</tr>
<tr>
<td><strong>Hydrocarbon</strong></td>
<td>Infrared</td>
<td>± 5% of reading</td>
<td>0-16%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>± 12 ppm</td>
<td>Over-range 25%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>± 0.5% of volume</td>
<td>0-5000 ppm</td>
</tr>
<tr>
<td><strong>Carbon Dioxide</strong></td>
<td>Infrared</td>
<td>± 5% of reading</td>
<td>0-5000 ppm</td>
</tr>
<tr>
<td></td>
<td></td>
<td>± 0.1% of volume</td>
<td></td>
</tr>
<tr>
<td><strong>Nitric Oxide</strong></td>
<td>Fuel cell</td>
<td>± 4% between 0-4000 ppm</td>
<td>0-5000 ppm</td>
</tr>
<tr>
<td></td>
<td></td>
<td>± 5% between 4000-5000 ppm</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>± 4% between 0-4000 ppm</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>± 5% between 4000-5000 ppm</td>
<td></td>
</tr>
</tbody>
</table>

84
Knowing the composition of the exhaust gases allows the evaluation of the equivalence ratio of the mixture, therefore the validation of the correction factor applied to the flow controller [61,62].

4.10 Thermal imaging

For a deeper understanding of the heat transfer from the combustion chamber, temperature measurements of the external surface of the combustion chamber were also performed. For the purpose, a FLIR ThermaCAM P25 thermal camera was used. Before utilizing the camera a proper and accurate calibration was performed, as the influence of radiation and reflected lights to the external surfaces greatly affects the measurements. The procedure is described next.

![FLIR ThermaCAM P25](image)

Figure 81 - FLIR ThermaCAM P25

When pointing at an object, the camera receives radiation not only from the object itself but also from the surroundings reflected by the object surface, as shown in Figure 82.

![Radiation paths](image)

Figure 82 - Radiation paths
Another contribution to radiation comes from the atmosphere through light scattering or stray radiation from intense radiation sources outside the field of view. Such effects are difficult to quantify, however in most cases they are small enough to be considered negligible. In case they are not negligible, the measurement configuration is likely to be such that the risk for disturbance is obvious, at least to a trained operator. Therefore attention will have to be paid to avoid the disturbance, for example by changing viewing direction or arranging protections against intense radiation sources. Referring to Figure 82 an equation can be derived to calculate the temperature of the object from the calibrated camera output. Assuming that the received radiation power $W$ from a black body source at temperature $T_{source}$ at short distance generates a camera output $U_{source}$ that is proportional to the power input, we can write:

$$U_{source} = CW_{source} \quad (35)$$

where $C$ is a constant. If the source is a grey body with a defined emissivity $\epsilon$, the consequent received radiation would be $\epsilon W_{source}$. The three collected radiation power terms are therefore:

$$E_{object} = \epsilon \tau W_{obj} \quad (36)$$

$$E_{r,ambient} = (1 - \epsilon) \tau W_{refl} \quad (37)$$

$$E_{atm} = (1 - \tau) \tau W_{atm} \quad (38)$$

In equation (36) $\epsilon$ and $\tau$ are the emittance of the object and the transmissivity of the atmosphere respectively. In equation (37), the sources from the surrounding ambient are at temperature $T_{refl}$, and $(1 - \epsilon)$ represents the reflectivity of the object. One simplification adopted in this case is that all the sources within the half-sphere seen from a point on the surface of the object are at the same temperature $T_{refl}$. Also it has been assumed that the emissivity of the surroundings equals 1 according to Kirchhoff’s law. In equation (38) the temperature of the atmosphere is $T_{atm}$ and $(1 - \tau)$ is the emissivity of the atmosphere. Therefore the total received radiation power can now be written as:

$$W_{tot} = \epsilon \tau W_{obj} + (1 - \epsilon) \tau W_{refl} + (1 - \tau) W_{atm} \quad (39)$$
Multiplying each term by the constant $C$ and replacing the $CW$ products with the corresponding $U$ according to the same equation, one obtains:

$$U_{tot} = \epsilon \tau U_{obj} + (1 - \epsilon) \tau U_{refl} + (1 - \tau) U_{atm}$$  \hspace{1cm} (40)

Solving for $U_{obj}$:

$$U_{obj} = \frac{1}{\epsilon \tau} U_{tot} - \frac{(1 - \epsilon)}{\epsilon} U_{refl} - \frac{(1 - \tau)}{\epsilon \tau} U_{atm}$$  \hspace{1cm} (41)

The formula shown in (41) represents the general measurement formula for this particular thermal camera.

Parameters to be supplied from the operator are: the object emissivity $\epsilon$, the relative humidity, the atmospheric temperature, the distance between lens and object and the effective temperature of the surroundings, or the reflected ambient temperature $T_{refl}$. Emittance and reflected ambient temperature required a few measurements in the calibration phase as they depend on the conditions of the surroundings. These effects represent a disturbance to the measurement and adequate compensation is needed. The most important object parameter to set correctly is the emissivity which gives a measure of the amount of radiation emitted from the object, compared to the emission from an ideal blackbody at the same temperature. Normally, object materials and surface treatments exhibit emissivity ranging from 0.1 to 0.95. A highly polished surface can have emissivity below 0.1, while an oxidized or painted surface has a higher emissivity. Oil-based paint, regardless of the colour in the visible spectrum, has emissivity of 0.9 in the infrared region. Human skin has emissivity of around 0.97-0.98. Non-oxidized metals represent an extreme case of perfect opacity and high reflexivity, which does not vary greatly with wavelength. Consequently, the emissivity of metals is low and increases with temperature. Emissivity of non-metals tends to be high, decreasing with temperature [63,64].

To determine the actual reflected temperature to input in the camera setup the reflector method described in the camera manual [63] was used. First an aluminium foil was crumbled to maximize the reflective surface, then “uncrumbled” and attached to a piece of cardboard. The piece of cardboard was placed in front of the object of measurement (the combustion chamber) and the emissivity of the camera was set at 1. The apparent temperature of the aluminium foil was then measured and adjusted in the camera setup. The emissivity was determined in a similar way: first a piece of electrical tape with known emissivity (0.97) was put on a sample object, which in this case was a piece of the same stainless steel used for the
combustion chamber. Then the object was evenly heated up to a known temperature. The tape temperature, which is supposed to be the same as the stainless steel object, was measured by the camera. Then the temperature of the object was measured and the emissivity setup in the camera was manually adjusted to match the tape temperature obtained previously. In this procedure, attention was paid to avoid effects such as forced convection, uneven distribution of temperature on the sample object and reflections on the object from the surroundings.

Emissivity of the stainless steel combustion chamber was estimated to be 0.13 on every side of the combustion chamber and the measured reflected temperature was 20 °C. These values were input in the thermal camera setup, along with the distance of the lens from the object, the ambient temperature and the relative humidity, measured using a wet bulb thermometer. These measurements were repeated before every test, to adjust the parameters to the actual conditions. Since the tests including the thermal camera measurements were performed on consecutive days, the reflected temperature and the emissivity were very close to the one shown above.

This particular model of thermal camera has a spatial resolution of 1.3 mrad, an accuracy of ±2 °C or ±2 % of the reading. The detector is a Focal Plane Array, uncooled microbolometer with a resolution of 320×240 pixels and the spectral range is 7.5-13 μm. Unfortunately the maximum temperature detected by this camera is 500 °C so a complete set of measurements could not be obtained. However thermal images of the combustion chamber when operating at low heat release rates were successfully acquired.
4.11 Assembled rig

In Figure 83 the assembled rig is shown.

As it can be seen the combustion chamber was mounted on two steel supports screwed on a vertical wooden board at a height which allows the flame observation from the bottom flange. The flow controllers were placed on a wooden board and the flows at the outlets joined together using a stainless steel wye. In this rig the outlet of the combustion chamber was connected to an aluminium volume connected to the exhaust gas analyzer through a flexible tube, however this solution was abandoned as water vapour contained in the combustion products used to condensate inside this volume and flow through the pipe. This affected the measurements as the analyzer was designed to operate with dry gases, therefore the pipe connected to the outlet was modified to direct the flow into another container, as described previously, immersed in ice, where condensation happened. Water was collected at the bottom of the new container and emptied manually. In Figure 83 the device able to generate a spark across two electrodes is also shown and connected to the combustion chamber. The blue wire had a spigot as termination and it was inserted from the bottom of the combustion chamber. Once ignited, the wire was removed and the B type thermocouple (not shown here) was inserted in the same slot. When the spark occurred it generated a great electromagnetic field which caused disturbances, especially to the Red-y flow controller and
the acquisition devices connected to a laptop through a USB cable. Since this flow controller was operated exclusively through dedicated software, a USB connection was necessary at all times. This required a separate power supply able to produce stable voltage and current to be installed and connected to the flow controller. The B type thermocouple was supported by an aluminium “arm” connected to a vertical threaded shaft and its axial movement (to measure the axial temperature profile in the combustion chamber) was regulated by a bolt. Also K type thermocouples are shown in Figure 83: a thermocouple was placed in the wye after the mixing between gas and air to measure ambient temperature, two more thermocouples were mounted on an aluminium support above the combustion chamber to measure the preheating temperature upstream of the porous medium and the combustion products temperature at the outlet, another thermocouple was mounted on another aluminium support on the right side of the combustion chamber and inserted into the bottom flange in the recirculating cavity, to measure the temperature at the inlet of the outer channel. To connect the main line to the combustion chamber inlet, a 1.5 mm diameter hypodermic tube was pressed into the inlet and on the other end a silicon tube allowed the main line connection to it. When performing tests under normal conditions (non-insulated) two stainless steel flanges were machined to prevent leakages from the combustion chamber and also a graphite gasket able to withstand extremely high temperatures was cut from a sheet and placed on the two flanges, as shown in Figure 84. To make sure no leakages were present, a portable hydrocarbon detector was used during the testing of the device and also during the operation.

Figure 84 - Stainless steel top flange with graphite gasket
In Figure 85 a simplified scheme of the test rig used is shown.

4.12 Insulating materials

As part of the experiments, to gain deeper understanding of heat transfer and heat losses of the system, the combustion chamber was insulated externally to achieve conditions as close as possible to the adiabatic case. The comparison between insulated case and non-insulated case is very useful in order to quantify heat transfer with respect to a reference situation.

For the purpose the external lateral surface of the combustion chamber was encased in insulating wool (Figure 86) provided by RS Components Ltd., UK.
Specifications of the Superwool 607 Fibre are shown in Table 7.

Table 7 - Specifications of Superwool 607 Fibre

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>350 kg/m³</td>
</tr>
<tr>
<td>Modulus of Rupture</td>
<td>1.2 MPa</td>
</tr>
<tr>
<td>Compressive Stress at 10% deformation</td>
<td>0.3 MPa</td>
</tr>
<tr>
<td>Thermal conductivity at 200 °C</td>
<td>0.05 W/mK</td>
</tr>
<tr>
<td>Thermal conductivity at 400 °C</td>
<td>0.08 W/mK</td>
</tr>
<tr>
<td>Thermal conductivity at 600 °C</td>
<td>0.11 W/mK</td>
</tr>
<tr>
<td>Thermal conductivity at 800 °C</td>
<td>0.15 W/mK</td>
</tr>
<tr>
<td>Thermal conductivity at 1000 °C</td>
<td>0.20 W/mK</td>
</tr>
<tr>
<td>Thermal conductivity at 1200 °C</td>
<td>0.26 W/mK</td>
</tr>
</tbody>
</table>

Slots were obtained in the bulk material in order to fit the combustion chamber and the supporting bolts.

Also additional top and bottom flanges were machined with a material with higher insulating capacity, provided by Goodfellow Cambridge Ltd, UK. The material is called Duratec750 and it is a machinable material manufactured from calcium silicate.

Table 8 - Specifications of Duratec750

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>1400 kg/m³</td>
</tr>
<tr>
<td>Flexural Strength</td>
<td>23 MPa</td>
</tr>
<tr>
<td>Compressive Stress at 10% deformation</td>
<td>55 MPa</td>
</tr>
<tr>
<td>Hardness</td>
<td>80 (Shore D)</td>
</tr>
<tr>
<td>Maximum service temperature</td>
<td>1000 °C</td>
</tr>
<tr>
<td>Shrinkage at 750 °C for 12 hours</td>
<td>0.14/1.1 %</td>
</tr>
<tr>
<td>Thermal conductivity at 750 °C</td>
<td>0.49 W/mK</td>
</tr>
<tr>
<td>Coefficient of thermal expansion at 100-750 °C</td>
<td>$6.6 \times 10^6$</td>
</tr>
</tbody>
</table>
These flanges were properly machined to allow access for measuring instruments as with the stainless steel flanges. In Figure 88 pictures of the insulated combustion chamber are shown. As it can be seen three layers of insulating wool were obtained from a single sheet and stacked one on top of the other to cover the whole vertical length of the combustor. On the top flange a hole to allow the connection to the gas analyser and a 0.5 mm hole to fit a K type thermocouple were obtained, as well as a 15 mm diameter hole for a cylindrical sight glass made of quartz to observe possible flame flashback upstream of the porous medium. The bottom flange was machined to allow the square sight glass, the B type thermocouple placed in the middle of the combustor and the inlet for the mixture. Also two 3 mm stainless steel sheets were machined and installed to increase the flexural strength of the system and to prevent the flanges from breaking. Even in this case, graphite gaskets were used to avoid leakages from the combustion chamber. Springs mounted on the supporting bolts were used to counter thermal expansion of the combustion chamber. At first the flanges were mounted on the combustion chamber without using the stainless steel plates and after a short time of operation they cracked and broke due to excessive flexural stress (Figure 87).

![Figure 87- Images of the broken ceramic flanges](image1)

![Figure 88- Images of the insulated combustion chamber - side (left), bottom (right)](image2)
5 EXPERIMENTAL RESULTS

This chapter presents the experimental results on the regenerative combustion chamber described in the previous section. One of the main objectives of this work is to achieve clean flat flame combustion therefore attention was initially paid to achieve high combustion efficiency. Subsequently temperatures within the combustion chamber were measured at different operating conditions to determine a comprehensive map of its thermal performances. Flame shape and characteristics were also observed through the quartz sight glass located on the bottom flange. Also, to provide further understanding of heat losses from the combustion chamber, thermal images of the outer wall of the combustion chamber were acquired. In the second part the combustion chamber was externally insulated with insulating wool and calcium carbonate flanges were installed to decrease the conductivity, thus the heat losses. Also in this case, thermal performances were measured and compared to the previous case.

5.1 Non-insulated case

Several operating parameters and their combinations were considered in the tests. In the present non-insulated case the mass flow rates ranged from 0.04 g/s to 0.18 g/s, while in the insulated combustion chamber case the mass flow rate varied from 0.04 g/s to 0.1 g/s. The mass flow rate was not increased further because the temperature of the combustion chamber walls became too high and the stainless steel started emitting in the visible spectrum, therefore, to avoid damages and deformations of the combustion chamber, the tests were stopped. In every case the device was started at stoichiometric conditions. Equivalence ratio was varied from rich mixture (around 1.1-1.2) and gradually decreased until flame extinction, keeping the mass flow rate constant.

5.1.1 Combustion products composition

As previously mentioned, the main purpose of the development of this combustion chamber is to achieve clean combustion. Equivalence ratio was calculated from the exhaust emissions so that the proper correction factor could be applied to the gas flow controller, as described in Appendix A2. Exhaust gas measurements were only performed in the non-insulated case.
In Figure 89 and Figure 90 the composition of the main components of the combustion products at 0.04 g/s and 0.15 g/s total mass flow rate is presented at different equivalence ratios.
As it can be noticed the profiles are almost identical and they show the peak of maximum production of carbon dioxide at stoichiometric conditions, as expected [65]. At the same point oxygen concentration drops to zero and the carbon monoxide concentration starts increasing as the mixture becomes rich. When the mass flow rate is set at 0.04 g/s (Figure 89), near the lower flammability limit ($\phi = 0.57$) the concentration of $CO_2$ reached 7.7%, the concentration of $O_2$ was at 9.84% and the $CO$ concentration was zero (or below the resolution of the analyzer, which is close to zero. See Table 6). At stoichiometric conditions the maximum volumetric concentration of $CO_2$ is reached (13.1%) and at the same time the oxygen concentration decreases to values very close to zero (0.1%). As the equivalence ratio is increased after the stoichiometry the concentration of carbon monoxide grows as the combustion cannot be as efficient. When the combustion is rich ($\phi = 1.14$) the $CO_2$ volume concentration decreases to 10.8%, after the peak at stoichiometric conditions. Oxygen is obviously absent in the combustion products and the content of carbon monoxide is 4.3%. In Figure 90 the profiles of the main species contained in the combustion products at 0.15 g/s are shown. In this case values are similar to those in Figure 89. Near the lower flammability limit ($\phi = 0.61$) the volumetric $CO_2$ concentration is 8.6%, the concentration of $O_2$ is 8.97% and the content of carbon monoxide is zero. At equivalence ratio equal to 1 these concentrations reach 13.4%, 0.59%, 0.6% for $CO_2$, $CO$ and $O_2$ respectively. On the rich side of stoichiometry the maximum equivalence ratio considered in this case is 1.14. The concentration of $CO_2$ reaches 11.3%, the carbon monoxide has a volumetric percentage of 4.41% and the oxygen drops to zero. It is worth mentioning that the two cases considered show slightly different lower flammability limits as at 0.04 g/s the lowest equivalence ratio is 0.57 and at 0.15 g/s it results 0.61. The composition of $CO_2$, $CO$ and $O_2$ at other flow rates considered in this study are not shown as they are very similar to the mentioned above cases.

Being LPG (Liquefied Petrol Gas) an undefined mixture of mainly propane and butane, the composition was not known so it was necessary to estimate it from the exhaust gases composition. However, assuming a mixture of 50% of propane and 50% of butane, the composition of the combustion products in the ideal case can be estimated. The chemical reaction at stoichiometric conditions therefore can be written as in Table 9.
Table 9 - Chemistry of LPG

\[
C_3H_6 + C_4H_{10} + \frac{23}{2}O_2 + \frac{23}{2} \times 3.76N_2 = 7CO_2 + 9H_2O + \frac{23}{2} \times 3.76N_2
\]

<table>
<thead>
<tr>
<th>$kg/mol$</th>
<th>44</th>
<th>58</th>
<th>368</th>
<th>1210.72</th>
<th>308</th>
<th>162</th>
<th>1210.72</th>
</tr>
</thead>
<tbody>
<tr>
<td>$mol$</td>
<td>1</td>
<td>1</td>
<td>11.5</td>
<td>43.24</td>
<td>7</td>
<td>9</td>
<td>43.24</td>
</tr>
<tr>
<td>$mol$ $fraction$</td>
<td>0.0175</td>
<td>0.0175</td>
<td>0.203</td>
<td>0.762</td>
<td>0.118</td>
<td>0.152</td>
<td>0.73</td>
</tr>
<tr>
<td>$mass$ $fraction$</td>
<td>0.0262</td>
<td>0.0345</td>
<td>0.219</td>
<td>0.7203</td>
<td>0.183</td>
<td>0.097</td>
<td>0.72</td>
</tr>
</tbody>
</table>

The combustion products mole fractions in Table 9 correspond to wet volume fractions (which include water content) therefore they cannot directly be compared with the measured values shown in the graph. Considering the carbon dioxide concentration, the dry mole fraction (without considering water in the combustion products) was 0.139; this matches quite closely to the experimental value of 0.134. Differences can be attributed to the LPG mixture which may differ from that assumed in the theory as it may contain additional compounds (such as nitrogen) not taken into account in the calculations. However, as the main purpose of this study is to perform an investigation of the thermal performances, this is considered acceptable in this occasion.

The gas analyzer also measured the content of NO\(_x\) in the combustion products. Usually we refer to nitric oxide (NO) and nitrogen dioxide (NO\(_2\)) as NO\(_x\) although NO is the predominant component produced during combustion [65]. The principal source of NO is the oxidation of atmospheric (molecular) nitrogen. However, if the fuel contains significant nitrogen, the oxidation of the fuel nitrogen-containing compounds is an additional source of NO. The description of NO formation from atmospheric nitrogen has been developed by Zeldovich. The main reactions governing formation and destruction of NO from molecular nitrogen are:

\[
O + N_2 = NO + N \tag{42}
\]

\[
N + O_2 = NO + O \tag{43}
\]

\[
N + OH = NO + H \tag{44}
\]

The NO formation rate strongly depends on temperature as well as oxygen concentration, therefore high temperatures and excess of oxygen result in high NO formation.
rates. Thus the NO\textsubscript{x} concentration is expected to reach the highest values in proximity of the stoichiometric conditions. This NO\textsubscript{x} formation mechanism is also known as thermal NO\textsubscript{x}.

During combustion of hydrocarbon fuels, another mechanism is involved in NO\textsubscript{x} formation. This is called prompt NO\textsubscript{x} and it is predominant in low temperature environments with rich flames and when residence times are short. It is the case of conventional size gas turbines, staged combustion systems and surface burners. The concentration of prompt NO\textsubscript{x} is proportional to the number of carbon atoms present per unit volume and is independent of the parent hydrocarbon identity [65].

$$CH + N_2 = HCN + N$$  \hspace{1cm} (45)

$$N + O_2 = NO + O$$ \hspace{1cm} (46)

$$HCN + OH = CN + H_2O$$ \hspace{1cm} (47)

$$CN + O_2 = CN + CO$$ \hspace{1cm} (48)

Another NO\textsubscript{x} formation mechanism exists and it depends on the nitrogen-containing organic compounds present in the fuel. Generally it becomes relevant with liquid and solid fuels, where these compounds are present in greater amount than in gaseous fuels. Usually NO is produced from HCN as intermediate species, along with molecular nitrogen.

NO\textsubscript{x} emissions from the combustion chamber under investigation are shown in Figure 91. Again, total mass flow rates ranging from 0.04 g/s to 0.18 g/s and equivalence ratios from 1.15 down to the lower flammability limit were considered. The first thing to be noticed is the dramatic increase of NO\textsubscript{x} concentration as the mixture approaches the stoichiometric conditions due to the higher heat release rate, which confirms the validity of the thermal NO\textsubscript{x} mechanism explained previously. As the flow rate increases, obviously the volumetric content of NO\textsubscript{x} in the combustion products becomes higher too, reaching 241 ppm at stoichiometric conditions when the flow rate is 0.18 g/s. At high flow rates, despite the shorter residence time in the combustion chamber (which generally contributes to lower NO\textsubscript{x} emissions [66]), the maximum temperature reaches higher values (see temperature measurements below), therefore outweighs the effects of residence time. As the flow rate is decreased, the concentration progressively lowers and at the minimum flow rate (0.04 g/s) is 11 ppm. On the lean side of stoichiometry the NO\textsubscript{x} content is much lower than the stoichiometric conditions. For example at $\phi$=0.7 and 0.18 g/s the concentration is only 35
ppm and it results in only 2 ppm when the flow rate is decreased to 0.04 g/s. Therefore at the chosen design point ($\phi = 0.7$) the NO$_x$ emission can be considered acceptable, compared to conventional gas turbines [67].

![Figure 91 - Emissions of NOx at different flow rates and equivalence ratios](image)

In addition, the concentration of unburned hydrocarbon was also measured by the analyser. Results showed values no higher than 1 ppm of unburned hydrocarbons in the exhaust gases at all the mass flow rates considered. This value confirms that complete combustion occurred in the combustion chamber.

### 5.1.2 Equivalence ratio and combustion efficiency

*Equivalence ratio.* As previously explained, the calibration of the gas flow controller was performed against the equivalence ratio calculated from the exhaust gas composition. The procedure to calculate the equivalence ratio was taken from Stivender [61] and it is based on an oxygen balance used to determine the air to fuel ratio. Generally a hydrocarbon based
fuel can be expressed as a compound containing carbon, hydrogen and oxygen $C_nH_mO_r$. Therefore the complete combustion reaction can be written as:

$$C_nH_mO_r + \frac{n_{O_2}}{\phi} (O_2 + 3.773N_2)$$

$$= n_p(\tilde{x}_{c_aH_b}C_aH_b + \tilde{x}_{CO}CO + \tilde{x}_{CO_2}CO_2 + \tilde{x}_{O_2}O_2 + \tilde{x}_{N_2}N_2$$

$$+ \tilde{x}_{NO}NO + \tilde{x}_{NO_2}NO_2 + \tilde{x}_{H_2O}H_2O + \tilde{x}_{H_2}H_2)$$

In Equation (49) $\phi$ represents the equivalence ratio of the mixture, $n_{O_2}$ is the number of $O_2$ moles required for complete combustion ($n + \frac{m}{4} - \frac{r}{2}$), $n_p$ is the total number of moles of exhaust products and $\tilde{x}_i$ is the mole fraction of the $i$-th species. Equation (49) can be solved in different ways depending on the amount of variables available from the measurements and on the type of measurement (dry, partially dry or wet). In this calculation NO\textsubscript{x} effects on equivalence ratio is considered negligible as its concentration is very low. Therefore there are seven unknowns left: $\phi, \tilde{x}_{H_2}, \tilde{x}_{H_2O}, \tilde{x}_{N_2}, n_p, a, b$ which require seven additional equations to solve the system.

**Carbon balance**

$$n = n_p(a\tilde{x}_{c_aH_b} + \tilde{x}_{CO} + \tilde{x}_{CO_2})$$

**Hydrogen balance**

$$m = n_p(b\tilde{x}_{c_aH_b} + 2\tilde{x}_{H_2O} + 2\tilde{x}_{H_2})$$

**Oxygen balance**

$$r + \frac{2n_{O_2}}{\phi} = n_p(\tilde{x}_{CO} + 2\tilde{x}_{CO_2} + \tilde{x}_{NO} + 2\tilde{x}_{O_2} + \tilde{x}_{H_2O})$$

**Nitrogen balance**

$$\frac{7.546n_{O_2}}{\phi} = n_p(2\tilde{x}_{N_2} + \tilde{x}_{NO})$$

**Sum of mole fractions**

$$\tilde{x}_{c_aH_b} + \tilde{x}_{CO} + \tilde{x}_{H_2} + \tilde{x}_{CO_2} + \tilde{x}_{N_2} + \tilde{x}_{NO} + \tilde{x}_{O_2} + \tilde{x}_{H_2O} = 1$$

To evaluate the mole fraction of water contained in the combustion products the water-gas reaction equilibrium constant is used.

$$K = \frac{\tilde{x}_{CO}\tilde{x}_{H_2O}}{\tilde{x}_{CO_2}\tilde{x}_{H_2}}$$

Values of $K$ usually range between 3.5 and 3.8. As stated by Heywood [65] the effect of the constant value is very small. In this case variation of $K$ from 3.5 to 3.8, affected the
calculated equivalence ratio only from the fourth decimal figure. Therefore a value of 3.5 was taken.

For fuels composed by carbon and hydrogen only, when all species are measured with the same background moisture (which in this case is dry), equation (56), which is based on the ratio of measured and computed oxygen-containing species to measured carbon-containing species, gives the air to fuel ratio.

\[
\frac{A}{F} = 4.773\left(\frac{M_{\text{air}}}{M_f}\right) \left(\frac{\text{CO}_2}{} + \frac{\text{CO}}{2} + \frac{\text{H}_2\text{O}}{2} + \frac{\text{NO}}{2} + \frac{\text{NO}_2}{2} + \frac{\text{O}_2}{}\right) \left(\text{HC} + \text{CO} + \text{CO}_2\right)
\]

(56)

Percentage of molar concentrations of species are in brackets and \(M_{\text{air}} = 28.96, M_f = 12.01 + 1.008y\) where \(y\) is the H/C ratio of the fuel.

The molar concentration of water in the exhaust gases can be expressed as:

\[
(H_2O) = 0.5y \frac{(\text{CO}_2) + (\text{CO})}{(\text{CO})} \left[\frac{K(\text{CO}_2)}{1}\right] + 1
\]

(57)

After obtaining the measured fuel to air ratio from equation (56), the equivalence ratio can be calculated by dividing it by the stoichiometric fuel to air ratio:

\[
\left(\frac{A}{F}\right)_{\text{st}} = \frac{34.56(4 + y)}{12.011 + 1.008y}
\]

(58)

Graphs reflecting the comparison between equivalence ratio calculated from the exhaust gases composition and the equivalence ratio from the flow controller calibration procedure are shown in Appendix A2.

This procedure also allowed the estimation of the chemical formula and physical properties of the LPG mixture used in the experiments. These are shown in Table 10.

<table>
<thead>
<tr>
<th>Chemical formula</th>
<th>Density [kg/m³]</th>
<th>Lower Heating value [MJ/kg]</th>
<th>Specific heat [kJ/kgK]</th>
<th>Stoichiometric fuel/air ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>(C_{3.49}H_9)</td>
<td>2.28</td>
<td>46.048</td>
<td>1.653</td>
<td>0.0643</td>
</tr>
</tbody>
</table>

The following molecular weight and lower heating values of the main species were used to evaluate the lower heating value of LPG (unburned hydrocarbons are expressed in equivalent propane concentration):
Table 11 - Properties of species [68]

<table>
<thead>
<tr>
<th>Species</th>
<th>Molecular weight [g/mol]</th>
<th>Lower heating value [kJ/kg]</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO₂</td>
<td>44.01</td>
<td>-</td>
</tr>
<tr>
<td>CO</td>
<td>28.01</td>
<td>10100</td>
</tr>
<tr>
<td>C₃H₈</td>
<td>44.097</td>
<td>46334</td>
</tr>
<tr>
<td>H₂</td>
<td>2.01</td>
<td>120000</td>
</tr>
<tr>
<td>N₂</td>
<td>28.01</td>
<td>-</td>
</tr>
<tr>
<td>O₂</td>
<td>31.99</td>
<td>-</td>
</tr>
<tr>
<td>NO</td>
<td>30.51</td>
<td>-</td>
</tr>
<tr>
<td>H₂O</td>
<td>18.02</td>
<td>-</td>
</tr>
</tbody>
</table>

**Combustion efficiency.** Knowing the exhaust gases composition allows the estimation of the heat released during combustion therefore gives a measure of the combustion efficiency. It is defined as the fraction of the total heat of reaction of the fuel-air mixture utilized in the combustion process. In this case only CO, H₂ and HC are considered as they represent the significant unburned products from a combustion process. Therefore the combustion efficiency can be written as:

\[ \eta_{comb} = 1 - \left( \frac{\dot{m}_f + \dot{m}_a}{\dot{m}_f} \right) \left( \frac{\sum x_i M_i LHV_i}{LHV_f M_b} \right) \]  

(59)

\[ \eta_{comb} = 1 - \left( \frac{\dot{m}_f + \dot{m}_a}{\dot{m}_f} \right) \left( \frac{x_{CO} M_{CO} LHV_{CO} + x_{H_2} M_{H_2} LHV_{H_2} + x_{HC} M_{HC} LHV_{HC}}{LHV_f M_b} \right) \]  

(60)

The combustion efficiency in the combustion chamber has nearly identical profiles at every flow rate considered in the study, thus only the cases at 0.04 g/s and 0.15 g/s have been chosen as examples and they are shown in Figure 92 and Figure 93.

As it can be noticed, the combustion efficiency at 0.04 g/s is always very high on the lean side of combustion, as it does not drop below 0.996 for equivalence ratios lower than 0.92. At equivalence ratios higher than 1 the efficiency drops dramatically and reaches values around 0.82 at equivalence ratio equal to 1.15. At 0.15 g/s the efficiency presents a similar behaviour with slightly lower values (0.99) from equivalence ratio 0.85 onwards.
At stoichiometric conditions the efficiency reaches 0.97, which is still very high. It can therefore be concluded that the regenerative combustion chamber produces clean and
efficient combustion when operating at lean mixture, in particular at the chosen design point ($\phi = 0.7$). The plots of combustion efficiency match the ones obtained in previous studies [11,25,69] on a similar combustion chamber, although no indications on how the efficiency was calculated are provided. The explanation of the drop in combustion efficiency as the equivalence ratio approaches the stoichiometric conditions is provided by Sakurai et al. [69]. On the lean side of combustion, with the increase in the equivalence ratio the width of the flame reaction zone decreases. At the same time the CO concentration increases and the reaction rate to the chemical equilibrium state is lower than other reactions, therefore the complete oxidation of CO does not occur given the small volume of the combustion chamber. The effect of preheating the mixture on combustion efficiency is not object of this study as it was demonstrated that only NOx emissions are affected by the increased reactants temperature [35]. In particular due to thermal NOx mechanism, the emissions resulted more than double when combustion air was preheated to 430 K, compared to the unheated air case. Negligible differences were noticed in CO emissions when preheating the combustion air.

5.1.3 Visual observation of the flame

In order to understand how the flame shape and characteristics change over the operating range (from 0.04 g/s to 0.12 g/s), the flame was observed through the quartz window placed in the bottom flange and pictures were acquired using a digital camera Canon Powershot 200IS. A comprehensive series of pictures is shown in this section.

In Figure 94 images of the flame at a total mass flow rate of 0.04 g/s are shown. In this case the flame always has a blue colour which becomes darker at low equivalence ratios and gradually becomes brighter as the equivalence ratio increases. Also as the mixture gets richer, the flame has shades of green in some points on the surface of the porous medium and this is confirmed also in previous researches [14]. In every case the flame was stable and from the pictures acquired looked attached on the bottom surface of the porous medium, which indicates that the burning velocity is higher than the incoming flow velocity. This is also due to the preheating temperature of the mixture which is increased in the channels of the regenerative combustor. In this situation, the heat released from the combustion ranges from 66 W to 143 W and the preheating temperature reaches values of up to 610 K.
Figure 94 - Flame pictures at different equivalence ratios - 0.04 g/s
In Figure 95 the flame behaviour described for the previous case is repeated for a total mass flow rate of 0.06 g/s. At low flow equivalence ratios (less than 0.9) the flame has a dark blue colour because of the heat losses to the porous medium. On the other side with rich combustion the flame becomes bright and slightly green. Heat release rates in the considered range varied from 102 W to around 222 W.
For the flow rate of 0.08 g/s, visualisation of the flame at equivalence ratio ranging from 0.8 to 1.12 are shown in Figure 96. In this case the heat released ranged from 132 W to 271 W. Once again when the equivalence ratio is lower than 0.91, the flame shows a dark blue colour. At $\phi = 0.91$ few red glowing spots appear in the central region of the porous flame holder. This happens because the higher heat released from the combustion increases the temperature of the flame holder resulting in the red colour shown in the pictures. This
phenomenon is also due to the increased burning velocity of the mixture, which depends on the equivalence ratio and causes the flame to move closer to the flame holder, therefore increasing its temperature.

Figure 97 - Flame pictures at different equivalence ratios - 0.1 g/s
At a total mass flow rate 0.1 g/s (Figure 97) a red flame area is clearly visible in the visualisations and it spreads across most of the surface of the porous medium with rich combustion.

![Flame pictures at different equivalence ratios - 0.12 g/s](image)

Figure 98 - Flame pictures at different equivalence ratios - 0.12 g/s

In Figure 98 pictures of the flame at a total mass flow rate of 0.12 g/s are shown. As it can be seen the porous medium has a red glowing colour due to heat transferred to it from the flame. Blue spots are always present on the surface of the porous medium because of the
uneven pore size distribution. At high equivalence ratios (for example at 1.09 and 1.12), the rich mixture causes these spots to turn from blue to slightly green as a result of the different gas concentration which yields a different emission spectrum, in accordance to other studies found in literature [14,70].

5.1.4 Axial distribution of temperature

In this study not only the images of the flame were acquired but also the axial distribution of temperature at the centreline of the combustion chamber at the operating condition considered. As explained previously, a B-type thermocouple inserted in the same slot as the ignition wire was used for these measurements. The thermocouple was mounted on a support free to be displaced along the vertical direction only (along the centreline of the combustion chamber). The regulation of the thermocouple from the porous medium was operated manually by screwing a nut into a bolt and a ruler was used to measure the displacement. Temperature measurements were taken at six points along the centreline: at 1 mm from the bottom surface of the porous medium, at 2.5 mm, at 5 mm, at 7.5 mm, at 10 mm and at 12.5 mm. The thermocouple was kept at a distance of 1 mm from the flame holder to avoid damages to the exposed junction.

Although the design point was set at equivalence ratio equal to 0.7, the temperatures were measured only at two equivalence ratios: 0.85 and 1 as representative of the order of magnitude of temperature within the reaction volume. One could expect that the temperature difference in the reaction volume between the cases at equivalence ratio 0.7 and 0.85 would be similar to that between the s equivalence ratio 1 and 0.85. The output voltage from the thermocouple was acquired by the datalogger and then converted to Kelvin using NIST tables. The readings were corrected to include effects of axial conduction through the thermocouple wires and radiation from the thermocouple to the surroundings, as described in the Appendix A4. The profiles of temperature are shown from Figure 99 to Figure 104.
Figure 99 - Axial distribution of temperature - 0.04 g/s

Figure 100 - Axial distribution of temperature - 0.06 g/s

Figure 101 - Axial distribution of temperature - 0.08 g/s
Figure 102 - Axial distribution of temperature - 0.1 g/s

Figure 103 - Axial distribution of temperature - 0.12 g/s

Figure 104 - Axial distribution of temperature - 0.15 g/s
The profiles of temperatures show the same behaviour at all the flow rates considered, with the maximum temperature reached in the combustion chamber located at 1 mm from the porous medium. This suggests that in every case the flame is attached to the flame holder as a result of the burning velocity being higher than the incoming flow velocity. As the thermocouple is moved away from the porous medium the temperature decreases almost linearly with the distance. It also can be noticed that the temperature drop occurring along the length of the combustion chamber is around 500 K to 600 K at the centreline. The correction due to axial conduction and radiation ranges from 50 K to 150 K, depending on the temperature difference. The disturbing effect of the thermocouple in this case was not considered, although it may be present and may affect the flame characteristics and the readings.

5.1.5 Temperature measurements

This section shows the values of temperature recorded by the four K-type thermocouples located within the channels in the combustion chamber, as described previously and as shown in the rig scheme in Figure 85, with the purpose of investigating the amount of heat that can be recovered and transferred to the cold reactants. Results are presented in the operating range from 0.04 g/s to 0.18 g/s total mass flow rate. Ambient temperature was also constantly monitored by thermocouple T1, however since it was always included in the range 294 K to 296 K, its influence on the combustion process and on the other readings can be considered negligible. The thermocouple mounted in the bottom flange cavity (T3) was directly exposed to high temperatures, therefore at flow rates of 0.15 g/s and 0.18 g/s, measuring the temperature was not possible as the values reached were outside the operating range of the instrument as discussed previously. After setting the correct values on the flow controllers, the system was left running for some time to allow the stabilization of the temperature in order to exclude transient and unsteady effects.
In Figure 105 measurements of the preheating temperature with changing equivalence ratio and mass flow rates are shown. The lowest level of preheating occurs at low equivalence ratio and at minimum flow rate (0.04 g/s), as expected. At these conditions the preheating temperature is just over 500 K. As the heat released is increased with increasing equivalence ratio, the preheating temperature rises as expected and reaches a maximum value of around 600 K corresponding to stoichiometric conditions. Increasing the equivalence ratio beyond the stoichiometric conditions causes a decrease in the preheating temperature due to the lower heat released during combustion. Similar trends are shown at higher mass flow rates. Considering the case where the mass flow rate is set at 0.18 g/s, the maximum temperature reached is around 900 K while a minimum temperature of 714 K was recorded at equivalence ratio equal to 0.52. The increase in preheating temperature with the total flow rate is explained with the increased heat release rate from combustion and the increased flow velocity in the cold reactants channel. This results in higher temperature difference and higher heat transfer coefficient which enhance the heat transfer rate, therefore the preheating temperature. This value confirms the validity of the design which prevents flame flashbacks in the cold reactants channel and generally upstream of the flame holder. In fact the
autoignition temperature of LPG is usually lower than 741 K (as stated in the safety datasheet published by the New Zealand gas supplier used in this research [71]) and it depends on the concentration of species in the fuel.

![Figure 106 - Temperature T3 (bottom of combustion chamber) at different mass flow rates and equivalence ratios](image)

The readings from the thermocouple T3 placed in the recirculation cavity (past the combustor) are presented in Figure 106. The combustor exit temperature also varies depending on the heat released during combustion therefore it is maximum at stoichiometric conditions and decreases as the mixture becomes lean or rich. Under the considered conditions the maximum temperature ranges from 1034 K to 1544 K.

In Figure 107 the profiles of temperature at the outlet of the outer channel are shown. This set of measurements helps evaluate the heat exchanger efficiency and the rate of thermal energy flowing out the combustor.
Once again it can be noticed that the higher the heat released during combustion, the higher the maximum temperature reached. Also, considering the highest temperature at different flow rates it can be seen that in every case a crossover point occurs, meaning that the preheating temperature of the reactants becomes higher than the exhaust gases temperature. For example at 0.18 g/s the maximum exhaust temperature $T_4$ at the outlet reaches 840 K while the preheating temperature $T_2$ is 894 K. This may not be desirable as this indicates reversal in direction of heat transfer; some of the heat is transferred back from the unburnt reactants to the combustion products in the outer channel. As highlighted in previous researches [72,73] the crossover point can occur in concentric triple tube heat exchangers and the design needs to be extremely well thought to avoid this to happen. This is also confirmed in the numerical study carried out in this thesis in Chapter 7.

Once the temperatures in the combustion chamber were obtained it was possible to evaluate the energy contained in the flows and to estimate the amount of heat recovered by the cold mixture. In Figure 108 the fraction of heat transferred from the combustion products to the unburnt mixture is presented at the operating conditions considered.
The ‘fraction of heat transferred to reactants’ on the vertical axis of Figure 108 was calculated by dividing the heat gain of the unburnt mixture by the heat released during combustion. This was easily obtained from the previously estimated lower heating value and the fuel mass flow rate. In the cases considered, the maximum recoverable heat reaches around 29% of the heat released during combustion and it occurs at 0.18 g/s. Decreasing the mass flow rate also decreases this value. It can be noticed from Figure 39 that the amount of heat recovered has its maximum when the equivalence ratio is low and progressively decreases as the mixture becomes richer. A possible explanation for this effect could be due to the relationship between laminar burning velocity and the incoming flow velocity. Decreasing the equivalence ratio causes a lower laminar burning velocity therefore, if the mass flow rate is kept constant, the flame will be pushed towards the outlet of the combustion chamber. This means that the flame could result slightly lifted from the flame holder, reducing considerably heat losses to the porous insert. When the flame is attached to the flame holder, heat is transferred from this one to the stainless steel walls and, through axial conduction, to the external surfaces where it is lost to the environment. This effect is further discussed in the numerical simulations presented in Chapter 6.
5.1.6 Thermal imaging

To provide further understanding of the thermal characteristics of the regenerative combustion chamber, infrared measurements of the device were performed using the equipment and the procedure described previously. Unfortunately the operating range of the thermal camera did not allow temperatures higher than 500 °C therefore only two cases at low heat release rates were studied: 0.04 g/s and 0.06 g/s at stoichiometric conditions.

As a reminder, after the calibration, the emissivity was set at 0.13 in the thermal camera setup and the reflected temperature was set at 20 °C and 21 °C in the two test sessions respectively. The camera was placed at 0.3 m from the combustion chamber and the images were acquired once steady conditions were reached.

Figure 109 - Thermal image of the combustion chamber - 0.04 g/s, φ=1
The calibration was made only on the combustion chamber, excluding the supporting bolts holding the two flanges together. In Figure 109 the bolts show a red colour because of the different characteristics of the material, in fact after several hours of tests at high temperature the bolts became oxidized and their colour turned black, therefore the value of emissivity changed (it reached nearly 1) too. This affects also the value of temperature read by the camera as the reflections of the bolts on the stainless steel surface change the value of emissivity in that particular region. Therefore the most reliable measurement is at the mid-height of the combustion chamber, where the effects of possible reflections from bolts and flanges are minimal.

![Figure 110 - Thermal image of the combustion chamber. Outer wall temperature - 0.04 g/s, φ =1](image)

To obtain the temperature of the outer surface, the thermal camera has the possibility to highlight isothermal surfaces and exclude higher temperature areas (which are displayed in grey) from the screen. With this method it can be seen that the temperature on the outer surface of the combustion chamber is very uniform and reaches a value of 311 °C. Small deviations from this value can be expected depending on the region considered. For example the area near the bottom flange which is closer to the flame and hot combustion products can
be expected to have a slightly higher temperature however the measured value in Figure 110 could represent an acceptable average value for the outer surface.

In Figure 111 the thermal image of the regenerative combustion chamber operating at 0.06 g/s mass flow rate and stoichiometric conditions is shown. The reflected temperature was set at 21 °C after the initial calibration. Also in this case the temperature is distributed on the external surface rather evenly and the value detected at the target is 388 °C.

![Thermal image of the combustion chamber - 0.06 g/s, φ =1](image)

Deviations of temperature on the external surface in both cases were contained in the range ±5 °C.
5.2 Insulated case

This section presents the results and discussion of the tests on the externally insulated regenerative combustion chamber, as described previously. The purpose of this second part of the experimental work is to obtain a reference case that can be used as a case scenario close to adiabatic conditions to compare the previous measurements to and also to study the effects of including an external insulation on heat transfer within the combustion chamber. Figure 112 shows a picture of the assembled combustion chamber with insulating wool around the external surface and with ceramic top and bottom flanges. The brown colour on the wool appeared after some time of operation due to heat. Springs were mounted on the four supporting bolts to counter the thermal expansion of the combustion chamber. Also two 3 mm thick stainless steel plates were mounted on the top and bottom ends of the combustion chamber to distribute the pressure uniformly on the ceramic flanges to avoid damages due to flexural stresses. This configuration was chosen to limit heat losses from the combustion chamber by using low thermal conductivity materials to separate hot stainless steel parts from the surroundings, from the connections for the instruments, inlet and outlet. In particular the insulating wool was machined in order to completely contain the combustion chamber to decrease heat losses from the lateral surface. The material provided good insulation at both low and high temperatures as the outer surface of the wool was never higher than 55 °C. The ceramic flanges were used to prevent heat losses in the axial direction and to avoid contact between combustion chamber and other connections. As demonstrated in a numerical research by Dent [40], axial conduction has a very important role in heat lost from the combustion chamber, therefore increasing the thermal resistance of this heat transfer pathway is essential. A 2 mm hypodermic tube was pressed into the combustion chamber inlet and the other end was connected to the silicon tube from the flow controllers to allow the air-fuel mixture in. Other parts in contact with the hot gases or hot stainless steel parts are the three thermocouples measuring axial distribution temperature in the combustion chamber, preheating temperature and temperature at the outlet. These losses could not be excluded from the measurements, except for the heat losses through the B-type thermocouple which are taken into account by correcting the temperature according to the procedure described in Appendix A4.
5.2.1 Visual observation of the flame

The first set of results to be presented relates to several images of the flame taken from the sight glass placed in the bottom flange. The operating conditions whenever possible remained the same as in the non-insulated case. However preventing heat losses caused a great increase in temperatures within the combustion chamber therefore the total mass flow rate (thus the heat released during combustion) had to be limited to 0.1 g/s to avoid structural damages or melting of some of the components.

In Figure 113, pictures of the flame when the flow rate is set at 0.04 g/s are shown. The heat release rates corresponding to the three images ranges from 101 W to 143 W.
As it can be seen the flame characteristics are similar to the non-insulated case, with a uniform flame attached to the porous medium. At low equivalence ratio (0.86) a dark blue stable flame occurred in the combustion chamber; as the equivalence ratio was increased, the flame became brighter as expected and, with rich mixture, shades of green appeared on some regions of the flame [14,70].
In Figure 114 pictures of the flame when the flow rate is set at 0.06 g/s are presented. In this case the heat released from the combustion ranged from 111 W to 222 W. As it can be noticed, at equivalence ratio equal to 0.63 the flame is very dark, almost invisible, because of the low heat release rate. However the flame was stable and looked attached to the flame holder, although very close to the lower flammability limit at those conditions. Increasing the equivalence ratio to 0.78 caused the flame to release a higher amount of heat therefore it became brighter and more visible. At $\phi = 0.89$ the heat being transferred to the porous medium increased so that the colour changed to dark red all over the bottom surface of the porous medium. Blue spots were also present on the surface indicating locations at higher flow of the air-fuel mixture due to the imperfect distribution of pore size. At stoichiometric conditions the red colour became more glowing and intense and increasing the equivalence ratio further led to a brighter red colour and some green spots to appear on the surface.
The next set of images in Figure 115 is related to the flame when the mass flow rate is increased to 0.08 g/s.

![Flame pictures at different equivalence ratios (insulated) - 0.08 g/s](image)

The heat released with varying equivalence ratio at 0.08 g/s ranged from 148 W to 274 W. At the first equivalence ratio considered (0.62) the flame appears as bright blue and uniform on the porous insert. It can be noticed from the edge on the slot of the window, that the internal surface of the ceramic flange becomes slightly incandescent. Increasing the equivalence ratio to 0.71 caused the bottom surface of the flame holder to become red hot as the burning velocity becomes higher than the incoming flow velocity causing the flame to remain attached to the porous holder transferring more heat to it. At $\phi = 0.85$ and at stoichiometric conditions the porous medium showed a very intense orange colour because of the great amount of heat received from the flame. At rich mixture the colour became brighter.
and turned into intense red/orange. This is in accordance with the emission spectrum of a black body in the visible field [74].

A picture of the top side of the combustion chamber was also taken at 0.08 g/s and equivalence ratio equal to 1 and it is shown in Figure 116. At these conditions the insulation caused the stainless steel parts of the combustion chamber to become red hot.

The last set of images show the flame characteristics when the flow rate is set at 0.1 g/s. This was the highest mass flow rate considered in these tests in order to prevent the combustion chamber to be damaged. The heat released in these cases ranged from 185 W to 343 W. As in the previous situations, at low equivalence ratio the flame shows a blue colour and lays uniformly on the surface of the flame holder. At equivalence ratio equal to 0.85 the porous holder becomes incandescent and shows an orange colour which becomes brighter as the equivalence ratio is increased. Also a large portion of the bottom flange started to be incandescent therefore the tests were stopped at this mass flow rate. Even in this case the upper part of the combustion chamber was photographed and as it can be noticed in Figure 118 the combustion chamber and the porous medium are red hot because of the elevated temperature. In this image the two K-type thermocouples T2 (upstream of the porous medium) and T4 (outlet of the combustion chamber) can be seen.
5.2.2 Axial distribution of temperature

As for the test run on the non-insulated regenerative combustion chamber, in this section the readings from the B-type thermocouple are presented and discussed. Also in this case the maximum mass flow rate was limited to 0.1 g/s because of the excessive
temperatures reached and the equivalence ratios considered were again 0.85 and 1. After the acquisition of the readings, the values were converted into Kelvins using NIST tables [58].

Figure 119 shows the profile of temperature along the centreline of the combustion chamber at different distances from the porous medium, as for the previous case: 1 mm, 2.5 mm, 5 mm, 7.5 mm, 10 mm and 12.5 mm. The maximum temperature reached at 0.04 g/s is slightly lower than 1250 K at stoichiometric conditions and it is measured at 1 mm from the bottom surface of the flame holder, meaning that the flame is attached to it. The curves are not perfectly linear at different axial locations. However, since the flame appeared to be in steady conditions at all times from the visual inspection, this can be attributed to minor imprecisions in the measuring equipment, inaccuracies in the vertical position of the thermocouple or disturbance effect due to the presence of the thermocouple itself.

The graphs in Figure 120, Figure 121 and Figure 122 present the axial distribution of temperature in the combustion chamber at equivalence ratio 1 and 0.85 operating at 0.06 g/s, 0.08 g/s and 0.1 g/s.
Figure 120 - Axial distribution of temperature (insulated) - 0.06 g/s

Figure 121 - Axial distribution of temperature (insulated) - 0.08 g/s

Figure 122 - Axial distribution of temperature (insulated) - 0.1 g/s
In every case presented the temperature shows a regular and almost linear behaviour with the distance from the porous medium. As mentioned previously inaccuracies may be present due to disturbance effects by the thermocouple. From these measurements it is also clear that the flame is located very close to the porous medium (i.e. closer than 1 mm) implying that the laminar burning velocity is higher than the flow velocity.

5.2.3 Temperature measurements

This section presents the temperature measurements made in the regenerative combustion chamber when encased in insulating wool.

The temperature T3 in this case was not measured because of the elevated temperatures reached in the recirculation cavity, which were beyond the operating range of a K-type thermocouple. As for the other results obtained for the insulated combustor (and presented earlier), the maximum mass flow rate was limited to 0.1 g/s. Figure 123 shows the variation of the preheating temperature T2 at different flow rates and equivalence ratios.

Figure 123 - Temperature T2 (reactants' temperature) at different mass flow rates and equivalence ratios (insulated)
Also in this case the peak of temperature occurs around the stoichiometric conditions at every flow rate considered, as the heat released from the combustion is highest. When the flow rate is 0.04 g/s, it can be seen that the maximum preheating temperature is slightly higher than 700 K; increasing the flow rate causes the preheating temperature to rise to 878 K, 949 K and 1035 K when flow rates are 0.06 g/s, 0.08 g/s and 0.1 g/s respectively.

In Figure 124 the temperature (T4) profiles at the outlet of the combustion chamber are plotted. In this case the highest temperatures before leaving the combustion chamber are 735 K, 898 K, 973 K and 1076 K at increasing flow rate steps at 0.04 g/s and the peak occurs in the proximity of the stoichiometric conditions.

Also a graph showing the fraction of heat transferred to the reactants from the combustion products was plotted and it is presented in Figure 125. As for the non-insulated case, the greatest amount of preheating of the cold mixture is on the lean side of combustion ($\phi \approx 0.65$) and it reaches 46 % of the heat released during combustion at 0.1 g/s. At lower flow rates this percentage drops to 32 % at 0.04 g/s and becomes 22 % when the mixture is rich. Again this can be explained remembering the relationship between laminar burning velocity and incoming flow velocity, described for the graph in Figure 108. Although
insulating materials are present in this case, the same effect applies as heat losses cannot be completely avoided but only limited.

**Figure 125** - Fraction of heat transferred from combustion products to cold reactants at different flow rates and equivalence ratios (insulated)

### 5.3 Comparison

In this section a direct comparison between non-insulated and insulated cases along with a discussion of the results is presented. The profiles of corrected temperature are shown from Figure 126 to Figure 129 at increasing flow rates from 0.04 g/s to 0.1 g/s. It can be noticed that the maximum temperature obtained in the combustion chamber (which is always close to the porous medium) in the non-insulated case is always slightly higher than the insulated case. This could be due to the higher preheating temperature of reactants which causes a higher laminar burning velocity therefore the flame is pushed towards the porous medium, increasing the heat losses from the flame to it. At low flow rates the difference between the maximum temperatures in the two cases is much higher than at high flow rates; for example at 0.04 g/s the difference is 163 K, at 0.06 g/s it is 85 K, while at 0.08 g/s it is 63 K but at 0.1 g/s they are nearly identical (only 12 K difference).
Figure 126 - Comparison of axial distribution of temperature at equivalence ratios 0.85 and 1 – 0.04 g/s

Figure 127 - Comparison of axial distribution of temperature at equivalence ratios 0.85 and 1 - 0.06 g/s

Figure 128 - Comparison of axial distribution of temperature at equivalence ratio 0.85 and 1 - 0.08 g/s
As the distance from the flame holder increases the temperature along the centreline in the insulated case decreases less rapidly than that in the non-insulated case and, after a “crossover” point, at the bottom of the combustion chamber it results always higher. Therefore it can be concluded that insulating the combustion chamber does not contribute greatly to the maximum temperature reached in the combustion chamber, however it helps limiting the temperature drop due to heat losses. When the combustion chamber was insulated, a greater amount of heat from the combustion products was transferred to the cold reactants. The graph showing this is presented in Figure 130 where the different sets of data previously discussed have been grouped onto the same plot. The benefits of employing an insulated combustion chamber in terms of regeneration of the cold mixture are evident, as in the insulated case the heat recovered is slightly more than double. This will also be confirmed in the numerical study presented in Chapter 7.

For a complete comparison, the preheating temperature (T2) and the exhaust temperature at the outlet (T4) are also presented. Error bars are not shown in these graphs for clarity.
Figure 130 - Comparison of fraction of heat recovered between insulated and non-insulated case

Figure 131 - Comparison of preheating temperatures (T2) between insulated and non-insulated case
In Figure 131 the reactants’ preheating temperature in both cases is presented. It can be seen that the difference of temperature $T_2$ between the two cases increases with the mass flow rate; at stoichiometric conditions the difference is equal to 95 K at 0.04 g/s, 186 K at 0.06 g/s, 205 K at 0.08 g/s and 240 K at 0.1 g/s. Insulating the combustion chamber also causes the exhaust gases to exit at a higher temperature that can be used for other purposes, i.e. further preheating of the gaseous or liquid fuel employed. Figure 132 shows the comparison of temperature $T_4$ with insulated and non-insulated combustion chambers. An interesting thing to be noticed is how the insulated combustion chamber does not show any crossover point as the temperature of the combustion products is always higher than that of the reactant mixture. This means that the heat flux towards the reactants is maximized. The final temperature also depends on the extension of the heat exchanger surface between the two flows, however in ultra micro gas turbines the overall dimensions are a constraint (otherwise the classification of “micro” would not apply) therefore a compromise between size and efficiency is needed.

One of the purposes of this study is to investigate the thermal exchanges within the regenerative combustion to provide useful information in the design process of such devices. One important detail that emerges from the measurement is the presence of the crossover
point between the temperature of the reactant mixture flowing in the inner channel and the
temperature of the exhaust gases flowing in the outer channel. As highlighted, this only
occurs when the combustion chamber is not insulated, thus when heat losses are not limited.
On the contrary, when insulating materials encase the combustion chamber, heat losses are
contained and the final preheating temperature is always lower than the exhaust gases at the
outlet. This indicates that heat is always directed towards the “cold” mixture, approaching the
ideal behaviour of a triple concentric tube heat exchanger, where the final temperatures tend
asymptotically to the same value. This effect can also be noticed in the numerical study
conducted as part of this research and presented in Chapter 7.

Considering the temperatures shown in Figure 132, the main drawback of having an
insulated configuration is the elevated temperature of the combustion products flowing out
the combustor. This also results in a decrease of thermal efficiency, if the enthalpy contained
is not utilized in another process downstream of the combustion chamber outlet.

5.4 Operation with vegetable oil

In this section the results obtained on the combustion chamber fuelled with vegetable oil
are presented. Another goal of the present research was to investigate the feasibility of using
vegetable oils as fuel in a premixed configuration at small scale. The regenerative combustion
chamber was designed with the main purpose of preheating the reactant mixture before
combustion and the successful outcome was presented in the previous sections. However the
design did not exclude the possibility of using a liquid fuel in the combustor, as the heat
recovered from the combustion products could be used to vaporize the fuel in the helicoidal
channel. For the purpose a fuel blurring injector, as described previously, was built and
mounted on the 6 mm thread designed on the bottom flange.

The physical characteristics of vegetable oils are very similar to those of diesel fuel,
especially the ones involved in combustion phenomena, and this is the main reason why they
are strongly considered as a replacement of fossil fuels. In Figure 133 characteristics such as
density, flash point, cetane number, net calorific value, and kinematic viscosity of diesel and
different types of oils are shown [75].
As it can be seen the properties of oils are not very distant from those of diesel fuel, except for viscosity which could be ten times higher. One approach to reduce viscosity is preheating the oil before injection and combustion. Many researches have proven the feasibility of this practice, especially in reciprocating engines, with results in terms of emissions and torque generated very similar to those of diesel [76-80]. Under this point of view the higher density of oils compensates the lower heating value compared to diesel fuel. Several researches showed that the viscosity of vegetable oils is comparable to that of diesel when its temperature is around 100 °C [81]. Figure 134 and Figure 136 show respectively the variation of kinematic viscosity and density with temperature up to 140 °C.

![Figure 133 - Properties of selected pure vegetable oils](image)

<table>
<thead>
<tr>
<th>Oil</th>
<th>Density at 20 °C</th>
<th>Flash point (°C)</th>
<th>Cetane number</th>
<th>NCV (kJ/kg)</th>
<th>Kinematic viscosity (mm²/s)</th>
<th>Pour point (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel</td>
<td>836</td>
<td>93</td>
<td>50</td>
<td>43,800</td>
<td>3-7.5</td>
<td>&lt;-5</td>
</tr>
<tr>
<td>Cotton oil</td>
<td>921</td>
<td>243</td>
<td>35-40</td>
<td>36,780</td>
<td>73</td>
<td>-1</td>
</tr>
<tr>
<td>Palm</td>
<td>915</td>
<td>280</td>
<td>38-40</td>
<td>36,920</td>
<td>95-106</td>
<td>31</td>
</tr>
<tr>
<td>Copa</td>
<td>915</td>
<td>-</td>
<td>40-42</td>
<td>37,100</td>
<td>30-37</td>
<td>20-28</td>
</tr>
<tr>
<td>Rapeseed</td>
<td>915</td>
<td>320</td>
<td>32-36</td>
<td>37,400</td>
<td>77</td>
<td>-11</td>
</tr>
<tr>
<td>Sunflower</td>
<td>925</td>
<td>316</td>
<td>35-37</td>
<td>37,750</td>
<td>55-61</td>
<td>-5</td>
</tr>
<tr>
<td>Soybean</td>
<td>920</td>
<td>330</td>
<td>36-38</td>
<td>37,300</td>
<td>58-63</td>
<td>-4</td>
</tr>
<tr>
<td>Groundnut</td>
<td>914</td>
<td>258</td>
<td>39-41</td>
<td>39,330</td>
<td>85</td>
<td>9</td>
</tr>
<tr>
<td>Jatropha curcas</td>
<td>920</td>
<td>240</td>
<td>45</td>
<td>38,850</td>
<td>55</td>
<td>3</td>
</tr>
<tr>
<td>Flax</td>
<td>940</td>
<td>241</td>
<td>35</td>
<td>39,307</td>
<td>45-50</td>
<td>1.7</td>
</tr>
<tr>
<td>Corn</td>
<td>915</td>
<td>277</td>
<td>98</td>
<td>39,500</td>
<td>60-64</td>
<td>-1.1</td>
</tr>
</tbody>
</table>

![Figure 134 - Dependence of kinematic viscosity with temperature of some vegetable oils and diesel fuel](image)
Figure 135 - Kinematic viscosity vs temperature of diesel and sunflower oil [81]

Figure 136 - Dependence of density with temperature of some vegetable oils and diesel fuel [81]

Figure 137 - Density vs temperature of diesel and sunflower oil [81]
5.4.1 Prediction of droplet evaporation

When designing the combustion chamber, in particular the inner annulus, the necessity of preheating and vaporizing vegetable oil droplets was taken into account and predictions of evaporation time of droplets were carried out. The approach uses the quasi-steady assumptions coupled with the continuity, species conservation and energy equations applied to a droplet \([82,83]\), that provide the following relation:

\[
d^2 = d_0^2 - Kt
\]  \hspace{1cm} (61)

where \(d\) is the droplet diameter (mm) at time \(t\) (s), \(d_0\) is the initial diameter of the droplet at time \(t = 0\) and \(K\) is the evaporation constant for a given temperature \(\left(\frac{mm^2}{s}\right)\). Equation (61) assumes that the droplet temperature is in equilibrium with its surroundings and remains constant in both space and time. Also in equation (61) convection and transport phenomena are not taken into account. The evaporation constant was taken from a previous research found in the literature \([82]\) and, although sunflower oil was not the object of the study, it was approximated with cottonseed oil, given their similar composition. In this case the estimated evaporation time was compared to the time necessary for the droplet to go through the inner channel length.

An approximate estimation can be done assuming the droplet to be small enough to be transported by the flow and assuming the density of the flow to be the same as the density of air in the inner channel. This was evaluated by solving the heat transfer equations in a triple tube system explained in Chapter 7. It was found out that the flow temperature in the inner channel had a sudden increase (reaching about 850 K) in the region just downstream of the inlet because of the high Nusselt number due to the entrance effect and because of the elevated difference of temperature between the hot flows and the cold mixture. The mixture temperature kept increasing along the length of the channel to values over 1000 K. Therefore an average air density of 0.4 kg/m\(^3\) was taken, which corresponds to a temperature of around 900 K. This allowed an estimation of the flow velocity, therefore the time to pass through the channel (the length of the channel is 0.47 m), as shown in Table 12.
Table 12 – Evaluation of time to pass through the channel

<table>
<thead>
<tr>
<th>Mass flow rate [g/s]</th>
<th>Flow velocity [m/s]</th>
<th>Estimated time to pass through the inner channel [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.04</td>
<td>33</td>
<td>0.014</td>
</tr>
<tr>
<td>0.06</td>
<td>50</td>
<td>0.0094</td>
</tr>
<tr>
<td>0.08</td>
<td>67</td>
<td>0.007</td>
</tr>
<tr>
<td>0.1</td>
<td>83</td>
<td>0.0057</td>
</tr>
<tr>
<td>0.12</td>
<td>100</td>
<td>0.0047</td>
</tr>
</tbody>
</table>

The size of the droplets was taken from a study in the literature investigating the spray characteristics of a flow blurring injector using different liquids [39]. It was shown that the injector was able to produce droplets with an average Sauter Mean Diameter of 36.6 µm in the outer region of the cone spray and 54.4 µm along the axial coordinate. These values were chosen for evaluating the evaporation time of a droplet in this study, as the present injector was built with the same characteristics as the one in the mentioned study. The evaporation constant $K$ measured experimentally by Daho et al. [82] is shown in Table 13 at four different temperatures, which represent realistic values of the flow in the inner channel when the combustor is in steady conditions.

Table 13 - Evaluation of droplet evaporation time

<table>
<thead>
<tr>
<th>Temperature [K]</th>
<th>Evaporation constant [mm$^2$/s]</th>
<th>Estimated evaporation time (Droplet SMD=36.6 µm) [s]</th>
<th>Estimated evaporation time (Droplet SMD=54.4 µm) [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>800</td>
<td>0.2</td>
<td>0.0067</td>
<td>0.0148</td>
</tr>
<tr>
<td>850</td>
<td>0.4</td>
<td>0.0033</td>
<td>0.0074</td>
</tr>
<tr>
<td>900</td>
<td>0.55</td>
<td>0.0024</td>
<td>0.0054</td>
</tr>
<tr>
<td>950</td>
<td>0.6</td>
<td>0.0022</td>
<td>0.0049</td>
</tr>
</tbody>
</table>

It can be seen from Table 13 how the predicted evaporation time for a droplet with Sauter Mean Diameter of 36.6 µm results always lower than the time to go from the inlet to the end of the channel except at high mass flow rate. The only case when evaporation is not completed at the end of the channel is when the flow rate is set at 0.12 g/s and the evaporation constant is 0.2 mm$^2$/s. When considering a bigger SMD (54.4 µm) evaporation does not occur every time but only for selected combinations of evaporation times and time to pass through the channel. It is necessary to remember that the evaporation constant does not take into account forced convection nor transport phenomena on the droplet, as the evaporation was experimentally
investigated in steady conditions under natural convection. In the actual situation, forced convection and radiation effects are present therefore the estimation is deemed to be on the conservative side.

5.4.2 Operating parameters

In the tests with vegetable oils only two different flow rates were considered: 0.1 g/s and 0.12 g/s. The combustion chamber was started with a mixture of Liquefied Petroleum Gas (LPG) and air both flowing through the atomizing air line and once steady conditions were reached, vegetable oil was pumped through the injector and the gas flow rate was gradually decreased until the combustor operated only on vegetable oil. The procedure was reversed when shutting off the system: vegetable oil was gradually decreased and at the same time the LPG flow rate was increased to allow cleaning of the combustion chamber. Ignition was achieved through a spark between the porous plate and an ignition wire inserted from the bottom flange. Generally steady conditions were reached after a period of time ranging from 15 to 20 minutes when operating with LPG as fuel. Operating conditions were considered steady when the temperature within the combustion chamber (specifically at 5 mm from the porous flame holder) was stable (±2°C) over a period of one minute. Having a high temperature in the combustion chamber was important in this case to ensure the vaporization of vegetable oil therefore the heat release rate from the combustion was kept high. In addition, the insulation was maintained on the combustion chamber for the same reason (Figure 138).

![Insulated combustion chamber for operation with vegetable oil](image)

During this session of tests, generally combustion was able to be self-sustained only with vegetable oil however few issues related to the presence of solid combustion products
deposition were encountered. Due to this, it was not possible to obtain comprehensive sets of data, as the combustion chamber required frequent cleaning. These issues are exposed in a following section.

### 5.4.3 Composition of combustion products

In this section the results from the combustion products composition when using vegetable oil (sunflower) are presented.

![Figure 139 - Combustion products composition at 0.1 g/s total mass flow rate – vegetable oil](image1)

As Figure 139 and Figure 140 show, the concentrations of the major species follow the expected trend with the peak of production of carbon dioxide located at stoichiometric
conditions. The volumetric percentages of CO\(_2\) reach 14.9 % when the mass flow rate is set at 0.1 g/s and 15.1 % at 0.12 g/s. Considering the lowest equivalence ratios (0.56 at 0.1 g/s and 0.59 at 0.12 g/s) in these tests, the volume fraction of carbon dioxide in the combustion products was 7.9 % at 0.1 g/s and 8.2 % at 0.12 g/s. Similarly to the experiments with LPG, the concentration of oxygen is higher where the mixture is lean and gradually decreases when approaching the stoichiometric conditions. In these two cases oxygen concentration reaches 8.9 % when the mass flow rate is 0.1 g/s and 8.2 % at 0.12 g/s. As the equivalence ratio increases close to stoichiometric conditions (\(\phi = 0.99\)), these values decrease to 0.59 % at 0.1 g/s and 0.39 % at 0.12 g/s mass flow rate. Concentrations of carbon monoxide, as expected, reached maximum value at stoichiometric conditions and decreased as the mixture became lean. In particular the percentage of CO was 0.08 % when the mass flow rate was set at 0.1 g/s and the equivalence ratio at 0.56 and 0.1 % with 0.12 g/s mass flow rate and 0.59 equivalence ratio. At stoichiometric conditions this percentage increased to 0.5 % at 0.1 g/s and 0.43 % at 0.12 g/s. Emissions of hydrocarbons were always lower than 5 ppm, and, although taken into account in the combustion efficiency calculations, they were not shown.

To have a reference to compare the emissions measurements to, the oxidation of sunflower oil is briefly exposed in this paragraph. The composition of vegetable oil is not well defined as it is a blend of different fatty acids, however several researches [84,85] showed that sunflower oil has a predominant component of linoleic acid (C18:2) and a lower content of oleic acid (C18:1). Usually the composition of sunflower oil is made up of 14-40 % of oleic acid, 48-74 % of linoleic acid and the remaining 10 % consists of palmitic and stearic acid. Therefore, for the purpose of calculating chemical equilibrium and stoichiometric fuel to air ratio, sunflower oil can be approximated as exclusively linoleic acid (being its main constituent): C\(_{18}\)H\(_{32}\)O\(_2\).

### Table 14 – Chemistry of sunflower oil

<table>
<thead>
<tr>
<th>kg/mol</th>
<th>mol fraction</th>
<th>mass fraction</th>
</tr>
</thead>
<tbody>
<tr>
<td>280</td>
<td>0.0083</td>
<td>0.116</td>
</tr>
<tr>
<td>800</td>
<td>0.2083</td>
<td>0.334</td>
</tr>
<tr>
<td>1316</td>
<td>0.7834</td>
<td>0.55</td>
</tr>
<tr>
<td>792</td>
<td>0.14</td>
<td>0.33</td>
</tr>
<tr>
<td>288</td>
<td>0.125</td>
<td>0.12</td>
</tr>
<tr>
<td>1316</td>
<td>0.735</td>
<td>0.55</td>
</tr>
</tbody>
</table>

\[ C_{18}\text{H}_{32}\text{O}_2 + 25\text{O}_2 + 25 \times 3.76\text{N}_2 = 18\text{CO}_2 + 16\text{H}_2\text{O} + 25 \times 3.76\text{N}_2 \]
Table 14 shows the chemistry of the oxidation of sunflower oil along with the molar weight, mole fraction and mass fraction of the single species. The stoichiometric fuel to air ratio can be obtained from the calculation of the combustion products and it is equal to 0.131. Considering the dry composition of the combustion products, the mole fraction of carbon dioxide is 0.16 at equivalence ratio equal to 1. Discrepancies between the experimental measurements and the calculated mole fractions in the combustion products may be due to the composition of the fuel being different than the one predicted. The theoretical chemical equilibrium assumed the composition to be exclusively linoleic acid, whilst in reality it is a mix of different acids. Also the purity of the oil is unknown, as it may contain other compounds. Despite these approximations, the values measured through the experiments are not very distant from the theoretical ones and in this case they can be considered acceptable.

![Figure 141 - Emissions of NOx at different flow rates and equivalence ratios – vegetable oil](image)

Also the emissions of NO\textsubscript{x} were acquired and they are presented in Figure 141. As the case with LPG, the concentration of NO\textsubscript{x} was low (not higher than 21 ppm) on the lean side of combustion in both cases and it increases up to 88 ppm as the combustion approaches stoichiometry. These results are comparable to the measurement of the combustion products composition when using LPG as fuel. In fact in that case, the maximum concentrations of NO\textsubscript{x} at stoichiometric conditions at 0.1 g/s and 0.12 g/s reached 80 ppm and 97 ppm respectively.
5.4.4 Visual observation of the flame

In this session of experiments only a set of pictures representative of the three different operating points at 0.1 g/s were taken to show the flame appearance when operating with vegetable oil. The pictures in Figure 142 were taken in a mirror reflecting the bottom flange of the combustion chamber, as the injector mounted on the rig did not allow the camera to be placed under the combustor.

As it can be seen at low equivalence ratio (0.56) the flame appears dark blue because of the heat lost to the flame holder, similar to the observations in the experiments with LPG.
Increasing the equivalence ratio to 0.7 causes a higher heat rate released from combustion and the flame becomes brighter. At stoichiometric conditions the flame has a bright blue colour uniformly distributed on the bottom surface of the porous flame holder. The red glowing area in the upper left corner of the sight glass is due to the presence of the B type thermocouple which was incandescent because of the elevated temperature in the combustor.

5.4.5 Profiles of temperature

While operating on vegetable oil, measurements of temperature along the axial coordinate, as done for the previous tests, were taken at the two flow rates considered but in this case only at stoichiometric conditions. Even in this case the acquisition of a comprehensive set of data was not possible because of clogging issues in the combustion chamber and in the porous flame holder. Further explanations of this are given in the following section.

Figure 143 – Axial distribution of temperature (vegetable oil) - 0.1 g/s

Figure 144 - Axial distribution of temperature (vegetable oil) - 0.12 g/s
Figure 143 and Figure 144 show the profiles of temperature plotted from the acquisitions at 0.1 g/s and 0.12 g/s respectively. The same correction adopted for the previous cases was used for these measurements. The temperatures were acquired at six different points representing different axial distances from the bottom surface of the flame holder: 1 mm, 2.5 mm, 5 mm, 7.5 mm, 10 mm and 12.5 mm. The temperature was recorded once the output voltage was stable enough to consider it steady. When the flow rate was set at 0.1 g/s the maximum temperature in the combustion chamber reached 1858 K and it was located at 1 mm from the porous medium, meaning that the flame was attached to the bottom surface of the porous medium, as happened in the previous experiments. As the distance of the thermocouple from the flame holder increased, the temperature decreased due to the effects of heat exchanged with the surrounding walls. At the bottom of the combustion chamber the temperature obtained was 1474 K. Again, at 0.12 g/s the temperature shows a peak in the proximity of the bottom surface of the flame holder corresponding to a temperature of 1926 K and gradually decreases with the distance from the porous medium to a temperature equal to 1558 K.

As Figure 145 shows, the axial profiles of temperature in the three experimental situations considered are very similar as the maximum temperature is reached near the bottom surface of the flame holder in all the cases. At 1 mm from the flame holder the temperatures of the combustion products are in the range 1817 – 1858 K. As the thermocouple moves towards the bottom of the combustion chamber, the temperatures in the two insulated configurations show the same behaviour and approximately the same values of temperature. In the non-insulated case, due to the absence of the additional external thermal resistance, the temperature decrease is more evident.

Figure 145 - Comparison of axial distribution of temperature at equivalence ratio of 1 when using vegetable oil and LPG - 0.1 g/s
During the tests also the preheating temperature of the unburnt mixture were acquired and Figure 8 shows the profiles at the considered equivalence ratios at 0.1 and 0.12 g/s total flow rate. As expected at stoichiometric conditions the preheating temperature reached the maximum as the heat release rate from combustion is maximum. In particular at 0.1 g/s it reached 1014 K at equivalence ratio equal to 1 and 912 at lean combustion (\(\phi=0.56\)). The trend was similar when the flow rate was set at 0.12 g/s as the maximum temperature resulted 1034 K at stoichiometric condition and decreased to 962 K at 0.59 equivalence ratio.

![Figure 146 - Preheating temperature of mixture upstream of the porous medium](image)

5.4.6 Clogging issues

As previously mentioned, during the experiments with vegetable oil, it was not possible to run the combustion chamber for a long period of time without incurring clogging of some of the components. In particular the average running time on vegetable oil was measured to be between 20 to 40 minutes, during which the acquisitions showed in this section were made. When dismantling the combustion chamber, solid deposits (possibly coke) were found on the top surface of the porous medium, as Figure 147 shows. Also the graphite gasket used to prevent leakages between the top flange and the combustion chamber remained stuck to the stainless steel surface; however it was removed mechanically without difficulties. After some time (around 20-25 minutes) running exclusively on vegetable oil, the flame turned gradually darker and weaker, due to the lower heat released because of the progressive clogging of the pores. Eventually combustion occurred only on half of the bottom surface of the flame holder because of pore blockage. This led to flame extinction as the flow was not able to pass through the porous medium.
Another area with grease deposits and solid combustion products was the inner surface of the top flange in contact with the unburnt mixture. On this surface the deposits were not as solid as those on the porous medium; on the contrary they were greasy and easy to remove. The bottom surface of the porous medium did not seem to be affected by the use of vegetable oil, as it appeared dark and oxidized, similar to that during the operation with LPG.

A possible explanation for the deposits of grease on the top flange can be attributed to oil condensation due to heat losses to the external environment. Although using ceramic flanges to limit heat losses, it may not have been enough to prevent condensation of oil upstream of the porous medium. At considerably elevated temperatures, the oxidation of oil probably started but did not complete, causing the formation of solid deposits on the upper surface of the flame holder (Figure 147) and compromising the operation of the combustor.

To further investigate the clogging issues, a session was run to acquire the temperature in the combustion chamber near the porous medium (leaving the B type thermocouple at 1 mm from the flame holder) over time. The purpose of this was verifying the decrease in heat released from the flame due to clogging of the porous medium. During this test, run at stoichiometric conditions at 0.1 g/s, the initial temperature was 1746 K. After around 10
minutes the temperature at the same location decreased to 1713 K and, after 20 minutes, 1632 K. About 8 minutes later, the flame extinguished at a temperature of around 1448 K (the output voltage was not very stable as it was constantly decreasing however this is the closest recorded value before flame extinction). This suggests that clogging could be the main reason of flame extinction as the heat released gradually decreased over time.

Whenever blocking happened, the combustor had to be disassembled and thoroughly cleaned. The procedure involved both mechanical cleaning of the surfaces of the combustion chamber and flushing of the channels and porous medium with a solution of warm water and caustic soda. This allowed the grease to react with the caustic soda to produce a soapy compound which was washed away. To clean the inner channel where the cold reactants flow the peristaltic pump was used in order to circulate the liquid solution through it. Also compressed air was used to remove the remaining water from the components. These simple steps allowed a good cleaning of the combustion chamber without damaging the surfaces that would otherwise result from mechanical cleaning.

Knowing the exhaust gases composition allowed the calculation of the combustion efficiency, which is shown in Figure 148 and Figure 149 for the two flow rates considered. The solid deposits were taken into account in the combustion efficiency calculation by weighting the porous medium after each test using a high precision scale (with accuracy of 0.01 mg in the range from 0 to 100 g). The flame holder was then cleaned as previously explained and weighted again. The resulting difference in weight (which was 12.3 mg at 0.1 g/s and 22.1 mg at 0.12 g/s) was related to the time of operation with vegetable oil and added to the combustion efficiency calculation. It can be seen how the efficiency is nearly 1 at low equivalence ratio and progressively decreases as the equivalence ratio increases. In particular at stoichiometric conditions it reaches 0.958 at 0.1 g/s and 0.961 at 0.12 g/s total mass flow rate.
The present section demonstrated the feasibility of using vegetable oils in a premixed configuration which can be applied to Ultra Micro Gas Turbines. Further adjustments and improvements are necessary to avoid clogging issues encountered in these tests and to ensure condensation of oil does not occur. For example, the geometry of the cold flow channel could be improved to maximize heat transfer to the reactants; a parametric study on geometric parameters and channel dimension would provide further insights. Also the effect of flame
holder porosity on clogging issues should be investigated as this would have effects on both heat transfer and clogging. Alternatively one could use a different combustion technique which is for example the liquid film combustion, as successfully reported in [41].

5.5 Summary

This chapter presented the results from an experimental study carried out on the newly developed regenerative combustion chamber. In the first part the design process, the requirements of the combustor and the issues encountered during the first tests were presented. Also a comprehensive description of the equipment and measuring tools used for the tests was reported, along with the calibration procedures of instruments. The second part of the chapter was entirely dedicated to the exposition of the results obtained. Results of the tests on the non-insulated combustion chamber were presented first. Initially an analysis of the combustion products, which allowed the calculation of the equivalence ratio of the mixture, was showed. In this section the measurement of the concentration of carbon dioxide, carbon monoxide, unburned hydrocarbons and NOx was carried out and presented for two different mass flow rates. Subsequently the combustion efficiency was calculated employing a procedure based on the dry composition of the combustion products. Also a visual inspection of the flame was performed and comprehensive sets of pictures taken from the sight quartz in the bottom flange were presented, with the purpose of showing the behaviour and characteristics of the flame at different equivalence ratios, at different mass flow rates and therefore at different preheating temperatures. Several measurements of temperature were also acquired during the tests. The first series of graphs is related to the axial distribution of temperature in the combustion chamber at different axial distances from the porous medium, using a B-type thermocouple. The following graphs show the preheating temperature, the temperature of the combustion products in the recirculating cavity at the bottom of the combustion chamber and the temperature of the combustion products at the outlet respectively. These measurements allowed the evaluation of the regeneration level occurring in the combustor at different operating conditions. The same tests were repeated on the combustion chamber in an insulated configuration, in order to limit heat losses to the environment and highlighting the differences with the previous case. Results showed that a higher amount of heat is transferred to the reactant mixture when the combustion chamber is insulated however on the other side a greater amount of heat contained in the exhaust gases is
dispersed to the environment. Finally results in terms of emissions and axial distribution of
temperature in the combustion chamber were shown when the combustion chamber was
operated exclusively on vegetable oil. Results showed that using vegetable oil as fuel is
possible however improvements are still necessary as issues were encountered during the
tests.
6 NUMERICAL STUDY ON COMBUSTION STABILIZED ON A POROUS MEDIUM

In this chapter a comprehensive two dimensional numerical study with the purpose of investigating the combustion process and how it is affected by the presence of a flame holder is presented.

6.1 Introduction

Numerical simulations at small scale are very important as experimental verifications aimed at detecting flame shape, stability and temperatures in micro-scale combustion chambers would be extremely difficult.

Flame quenching also is a major issue at small scale and researches [86] pointed out the importance of thermal exchanges between flame and wall and radical adsorption by the walls which causes non homogeneous chemistry. Another condition for the flame to be sustainable is that the incoming flow velocity needs to be lower than the burning speed. If this does not occur the flame is “pushed” towards the outlet and eventually blows off. Computational Fluid Dynamic (CFD) simulations of micro combustion is a relatively recent topic and most studies mainly focus on the effects of combustor size, combustor geometry and operating conditions on the flame shape.

Flame stability limits of methane-air and propane-air mixture at different incoming flow temperatures (298 K, 400 K and 500 K) have been investigated in [87]. Different tubes diameters were used in the experiment ranging from 200 to 420 μm. It was found out that stability limits are widened by increasing mixture’s initial temperature. Supporting experimental data, a numerical study of premixed methane/air flames was also carried out [88] using a 17 species and 25 reactions mechanism, highlighting the flame characteristics.

Two dimensional simulations of a methane/air flame occurring on a perforated plate have been performed in [89]. Particular attention was given to the flame characteristics in steady conditions. The varying parameters were incoming flow velocity, equivalence ratio, plate thermal conductivity and distance between adjacent holes. Results showed how increasing thermal conductivity increases the flame stand-off distance from the plate and eventually becomes constant for even higher values. On the other hand the distance is zero
when the thermal conductivity is very low. It is also showed that when the distance between adjacent holes increases the flame holder’s temperature increases given the higher heat losses through the solid material.

Simulations of combustion between parallel plates has been also investigated using methane/air and propane/air mixtures [90,91] one step mechanisms, with particular interest on distance between the plates, effect of wall thermal conductivity and flow velocity on combustion characteristics. It was found out that axial conduction through the wall, and therefore its thermal conductivity, plays an important role in flame stability and the optimal range of wall thermal conductivity was showed to be 3 to 5 W/mK.

In another research [92], combustion in a 2 mm diameter cylindrical combustor was analysed, at equivalence ratio of 0.9 and different inlet velocities. The main objective was to highlight the effect of combustor size, of the incoming flow conditions and of the slip-wall condition, however high residuals ($1\times10^{-3}$) for convergence were applied and the solution in this case could not be considered realistic and reliable. The effects of initial mixture temperature and the prediction in terms of species concentration have been recently investigated [93] using a 16 species and 41 reactions mechanism, pointing out the necessity of applying low target residuals ($1\times10^{-6}$) on the solution reliability and the beneficial effect of preheating reactants on flame stability and flammability limits.

However none of the above listed publications are related to numerical studies involving combustion (using a detailed skeletal mechanism) stabilized over a flame holder and its interactions with it and with side walls. The flame holder allows the flame to be stabilized and controlled, with the additional purpose of preventing flashback and allowing a proper mixing of the fuel air mixture. On the other side this is likely to affect the thermal behaviour of the combustion chamber as it might introduce heat losses through it.

Therefore this study investigates the combustion properties of a methane-air mixture in a cylindrical combustor of fixed size and geometry under actual operating conditions and how they are affected by heat transfer through the flame holder and through the walls. Methane is chosen as gaseous fuel for the simulations, as it is still recognized as the most viable and available fuel for micro scale applications, thanks to its high energy density and storage safety.

In the following section the numerical model is described.


6.2 Numerical model

Since the analysis will be set as axial symmetric, only half domain is modelled. The length of the combustor is 100 mm, its radius is 9 mm and a 10 mm thick porous medium has been modelled in the fluid domain at a distance of 10 mm from the inlet (Figure 150). To visualize the temperature gradients and the heat transfer through it, 1 mm thick walls also were modelled and assumed to be stainless steel (with thermal conductivity $k_w=16$ W/mK) for all the simulations.

![Figure 150 - Scheme of the combustor model](image)

The present analysis uses the commercial computational fluid dynamics software tool ANSYS Fluent 16 [54], able to model flows, mass, momentum, energy and heat transfer equations. This version of Fluent includes the CHEMKIN package from Reaction Design [94] which is a code able to solve detailed reaction mechanisms. The reaction mechanism employed in this analysis is a reduced skeletal mechanism including 17 species and 73 reactions [95] simulating methane combustion and the thermodynamic and transport databases are imported from the CHEMKIN database [96,97]. Compared to one or two steps combustion mechanism, the present one offers more accurate results as it includes more intermediate reactions. The domain considered is two dimensional and axially symmetric to save computational time. The validity of this skeletal mechanism has been proved in numerical works and comparisons with other reaction mechanisms, including the well-known GRI-Mech 3.0 [93,98]. The simulations were run on a 4 cores Intel i7-4770 with 8 GB RAM and on the High Performance Computing cluster [99] available at the University of Auckland.

The pressure based solver is used for the computational process with a laminar model, as the Reynolds number is never higher than 1500. The species transport model including volumetric reactions is enabled and the laminar finite-rate turbulence-chemistry interaction allows the calculation of chemical source terms using Arrhenius expressions. The
initialization is achieved first by running a “cold flow” simulation for 200 iterations and then, after enabling volumetric reactions, patching a temperature of 2200 K in the domain to simulate a spark. The coupling between pressure and velocity is guaranteed by the SIMPLE algorithm and a second order upwind spatial discretization is used. Recently the importance of setting low residual target was proven [93] and in this research they are set to $1 \times 10^{-6}$ for continuity, energy and species concentration. The physical characteristics of the porous medium are taken from a previous experimental research where the pressure drop across different types of flame holder was investigated [12]. ANSYS Fluent 16 models the porous domains adding a momentum source term to the fluid flow equations [55]. This term relates to simple homogeneous porous media and it is composed by two parts: a viscous loss term and an inertial loss term, as shown in (1):

$$S_i = -\left(\frac{\mu}{\alpha} v_i + C_2 \frac{1}{2} \rho |v_i| v_i \right)$$

where $\alpha$ is the permeability, $v$ is the flow velocity and $C_2$ is the inertial resistance factor. The needed values are extrapolated from the experimental data in [12], where the pressure drop across the porous media was plotted against the incoming flow velocity. To study the influence of the porous material on heat transfer mechanism three different thermal conductivities are chosen and compared: 0.5 W/mK, 16 W/mK (corresponding to stainless steel conductivity) and 200 W/mK (corresponding to aluminium conductivity). The simulations assume isotropic porous medium and steady conditions. Fluent calculates an effective thermal conductivity [55] of the porous medium as the volume average between the fluid conductivity $k_f$ and the solid conductivity $k_s$ (based on the porosity $\gamma$), as shown in (63):

$$k_{eff} = \gamma k_f + (1 - \gamma) k_s$$

Porosity is kept at a value of 0.5 when the holder’s thermal conductivity is varied. In a following section the effects of changing porosity are analysed. The physical properties of the mixture are calculated using the following laws: ideal gas law for the mixture density, mixing law for the specific heat, mass weighted mixing law for thermal conductivity and viscosity and kinetic theory for mass diffusivity. Kinetic theory is used to calculate thermal conductivity and viscosity, piecewise polynomial expressions depending on temperature (implemented in Fluent [54]) are used for specific heat of single species. The governing equations used by Fluent are summarized in Table 15. $\mu$ is the molecular viscosity, $\tau_{eff}$ is the
stress tensor, $J$ is the diffusion flux of species $i$ and $j$, $S_h$ is a term source including the heat of chemical reaction defined according to molecular weight of species $M_{w,i}$ and Arrhenius molar rate of creation or destruction of species $\bar{R}_{i,r}$ [100].

<table>
<thead>
<tr>
<th>Table 15 - Governing equations</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Continuity (2D axi-symmetric):</strong></td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td><strong>Momentum conservation:</strong></td>
</tr>
<tr>
<td><strong>Energy in fluid regions:</strong></td>
</tr>
<tr>
<td><strong>Energy in solid regions:</strong></td>
</tr>
<tr>
<td><strong>Species:</strong></td>
</tr>
<tr>
<td><strong>Convective heat transfer:</strong></td>
</tr>
<tr>
<td><strong>Radiative heat transfer:</strong></td>
</tr>
</tbody>
</table>

The inlet is defined as a mass flow inlet and the values used in the simulations vary (0.04 g/s, 0.08 g/s, 0.15 g/s, 0.3 g/s and 0.5 g/s of mixture); different initial mixture’s temperatures are investigated: 298 K, 500 K and 800 K. Three different equivalence ratios are considered in this study (1, 0.7 and 0.5) and the mass fractions of methane and air at the inlet are calculated accordingly. All tests are conducted at atmospheric pressure (101325 Pa) and the walls include no slip boundary condition and zero diffusive species flux. An external convective heat transfer coefficient of 12 W/m$^2$K is chosen according to the empirical correlation for natural convection from horizontal cylinders developed by Churchill and Chu [101] (reported in [64]). Standard value of emissivity of steel (0.3) is applied to the model. As a complement of the numerical study, the effect of porosity also is studied and the considered
values are 0.1, 0.5 and 0.9. Finally a series of simulations is run setting the external temperature at 800 K.

A preliminary mesh sensitivity analysis was performed. The mesh was made up of square elements and the considered element sizes were 0.0005 m, 0.00025 m, 0.0002 m, 0.00015 m and 0.0001 m.

The mesh with 0.00025m element size is shown in Figure 151. Other meshes considered are not shown as only the element size is different.

![Mesh 0.00025m element size](image)

Figure 151 – Mesh 0.00025m element size

Results in terms of maximum temperature were identical for all meshes but the 0.0005 m grid showed difficulties to converge, therefore 0.00025 m element sizes were chosen for all simulations. Figure 152 shows the results in terms of maximum temperature with different meshes.

![Sensitivity analysis - Maximum temperature vs mesh size](image)

Figure 152 - Sensitivity analysis - Maximum temperature vs mesh size

### 6.3 Validation of the model

The validity of the numerical model is proven by running Computational Fluid Dynamics simulations of a reference case and comparing the results in terms of adiabatic
flame temperature (AFT) derived from a code developed by the French research centre Cerfacs [102], based on the chemical kinetics software Cantera [103]. A further verification is subsequently obtained by validating the code against the experimental data and theoretical calculations extrapolated from the work of Bechtel [104], as it is the only available data in literature. Although the experimental results in [104] show only an area slightly longer than 4 mm, they are able to capture the flame structure and the concentration of the main species. The comparison with the experimental data from Bechtel [104] is performed by properly overlaying the profiles obtained from the numerical simulation with the values extrapolated from the above mentioned work. Simulations are run with a fixed incoming mixture mass flow rate (0.08 g/s, which corresponds to a velocity of 0.26 m/s at 298 K), stoichiometric air to fuel ratio and adiabatic conditions. Four values of mixture’s initial temperature are considered: 298 K, 500 K, 700 K and 800 K.

Table 16 shows a comparison between the adiabatic flame temperatures calculated with the Cerfacs code and the numerical simulations run with Fluent.

<table>
<thead>
<tr>
<th>Reactants’ temperature</th>
<th>Cerfacs AFT</th>
<th>Fluent AFT</th>
<th>Percentage difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>298 K</td>
<td>2225 K</td>
<td>2228 K</td>
<td>0.13 %</td>
</tr>
<tr>
<td>500 K</td>
<td>2321 K</td>
<td>2327 K</td>
<td>0.25 %</td>
</tr>
<tr>
<td>700 K</td>
<td>2412 K</td>
<td>2420 K</td>
<td>0.33 %</td>
</tr>
<tr>
<td>800 K</td>
<td>2456 K</td>
<td>2465 K</td>
<td>0.36 %</td>
</tr>
</tbody>
</table>

The incoming mixture flow rate is set to 0.08 g/s, the equivalence ratio equals 1 and the reactants’ initial temperature is 298 K. Figure 153 shows the predicted centreline temperature profile in the combustor overlaying the above mentioned experimental data. For the purpose of validating the combustion mechanism, the zone where the flame occurs is considered, which is just downstream the porous domain and has a thickness slightly lower than 1 mm, which matches the computational results reported in [93].
The temperature profile shows excellent agreement with the experimental data. The only difference that can be noticed is the slightly higher temperature before the combustion occurs. This can be explained with the presence of the porous medium which is directly heated by the flame and in turn preheats the mixture before the combustion. The temperature gradient, as can be seen in Figure 153, is also slightly different. Downstream the flame, the temperature profile seems to increase with the distance, not following the experimental values, however this can be due to heat losses during the experiments, which are obviously not present in the numerical simulations.

Under the same initial conditions, the molar concentrations at the centreline of selected species involved in the reaction are shown in Figure 154. The species considered are H\textsubscript{2}O, CO\textsubscript{2}, CO, CH\textsubscript{4}, O\textsubscript{2} and H\textsubscript{2} and again only the zone where the flame occurs is shown. Generally the agreement of the numerical model with the experimental data and with theory from Bechtel [104] is reasonable enough, however there are some minor differences. Sudden changes in the values of mole fraction (e.g. H\textsubscript{2}O, CO, H\textsubscript{2}) are due to discretization length along the axial coordinate.
Figure 154 - Main species molar fraction vs axial distance from inlet - 0.08 g/s mixture incoming flow rate, $\phi = 1$

The profiles upstream the flame, match very well with experimental data and with theoretical results from Bechtel [104]. This behaviour is also maintained in the initial region of the flame in each graph. Most of the times the molar concentration gradient across the flame is slightly different compared to the experimental results, but this could be attributed to the presence of the flame holder which increases the temperature before the combustion happens. In the graph showing the CO$_2$ profile, the numerical simulation provides lower values of molar concentration just downstream the flame, starting at a distance of about 0.0208 m from the inlet (0.8 mm from the surface of the flame holder). As the distance increases, the trend seems to align with theory and experiments. Satisfactory agreement is obtained for the H$_2$O, CH$_4$ and O$_2$. The profiles of CO and H$_2$ both display peak values in the
flame surroundings and, although the shapes are similar with respect to the experimental data, the values are somewhat different. The reason of these discrepancies could be attributed to inaccuracies of the measuring equipment and also to the numerical model setup in fact the presence of the flame holder, as said, could alter the combustion dynamics. Nevertheless the comparison in general shows very good agreement and the model provides accurate enough prediction of the combustion phenomena. In the following section the results of the computational study are presented.

6.4 Results and discussion

6.4.1 Adiabatic case

As a reference case numerical simulations with the adiabatic condition applied were run first. The contours show only one half of the combustor, as the problem was defined as axial symmetric. The outer wall was not modelled at all to reduce the computational time and all the other variables and parameters were changed as described in the previous section. Figure 155 shows the contours of temperature in the combustor at equivalence ratio equal to 1 and at different reactants’ temperatures and incoming mass flow rates. As expected, in every case the adiabatic flame temperature is reached and the flame shape varies depending on the flow conditions. At low flow rates, i.e. 0.04 g/s, 0.08 g/s and 0.15 g/s, the contours of temperature show flat flame occurring on the downstream surface of the porous medium. This means that the burning velocity is equal or higher than the incoming flow velocity, being this parameter dependant on reactants’ temperature [105]. This behaviour can also be noticed from the contours of temperature in the porous domain. When the flame holder’s temperature increases, the burning velocity is higher than the incoming flow velocity and heat is transferred from the flame to the porous medium. This heat can become considerable, and results show that the temperatures in the porous domain reach values of around 1800 K and 1600 K, when the incoming flow rate is 0.04 g/s and reactants’ temperatures are 800 K and 500 K respectively, which could physically damage the porous material. In the adiabatic case heat is obviously conserved inside the domain and there are not dispersions to the environment. However, in case of non-adiabatic external walls, this could lead to flame quenching. As the flow rate increases, the incoming velocity becomes much higher than the burning velocity and the flame shape changes from flat to conical, gradually lifts from the flame holder and eventually blows out. When this situation occurs the numerical simulations
took longer time to converge because of flame instabilities. The different shape of the flame is caused by the wall boundary condition. There were cases where convergence was not obtained because some residuals related to minor species were oscillating however continuity, energy and major species concentration residuals were low enough (order of magnitude of $10^{-5}$) to consider the solution reliable.

Figure 155 - Contours of temperature for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 1

Figure 156 - Contours of velocity for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 1
As said, the first condition for a flame to be sustainable is that the burning velocity has to be lower than the flow velocity. However the dimensions of the combustor are also to be considered, in fact in microscale combustors (rather than mesoscale combustors) the effects
of transport properties have also to be taken into account. In a previous numerical work [93] on a 2 mm diameter combustor, it was demonstrated that at 1 m/s incoming velocity, 300 K reactants’ temperature and stoichiometric conditions the flame was not able to sustain, as the calculated burning velocity at that condition is around 43 cm/s [105]. Laminar burning velocities of methane at stoichiometric conditions and atmospheric pressure can be calculated using equation (64) proposed in [105] or, alternatively, equation (65) suggested in [53] for natural gas can be adopted, although less accurate, because of the slightly different gas compositions.

\[
\begin{align*}
u_t &= 10 + 0.0003717 \times T^2 \quad \text{(64)} \\
u_l &= u_{t_0} \left( \frac{T_u}{T} \right)^\alpha \left( \frac{P_u}{P} \right)^\beta \quad \text{(65)} \\
u_{t_0} &= -177.43\phi^3 + 340.77\phi^2 - 123.66\phi - 0.2297 \quad \text{(66)} \\
\alpha &= 5.75\phi^2 - 12.15\phi + 7.98 \quad \text{(67)} \\
\beta &= -0.925\phi^2 + 2\phi - 1.473 \quad \text{(68)}
\end{align*}
\]

In equations (64) and (65) \(T\) represents the initial mixture temperature, \(P\) the operating pressure, \(T_u\) and \(P_u\) are reference values of temperature and pressure (300 K and 0.1 MPa), \(u_{t_0}\) is the reference laminar burning velocity defined in equation (66), \(\alpha\) and \(\beta\) are coefficients obtained experimentally and depending on equivalence ratio.

<table>
<thead>
<tr>
<th>Initial temperature ([\text{K}])</th>
<th>Burning velocity calculated with eq. (64) ([\text{cm/s}])</th>
</tr>
</thead>
<tbody>
<tr>
<td>298</td>
<td>43</td>
</tr>
<tr>
<td>500</td>
<td>103</td>
</tr>
<tr>
<td>800</td>
<td>248</td>
</tr>
</tbody>
</table>

In the present situation, the adiabatic case with incoming mass flow rate of 0.3 g/s (which corresponds to about 1 m/s at 298 K and atmospheric pressure) in Figure 155 shows a stable conical flame. The flame starts developing near the wall, therefore the maximum temperature is close to the wall, and the heat diffuses towards the axis until the mixture reaches the ignition temperature. Where the flame is conical and stretched the maximum temperature in the domain is reached only locally because of its limited size. On the other side when flat flames occur, the average temperature at the domain outlet reaches the
maximum value. Small conical flames can be noticed in two cases in Figure 155 where the incoming flow conditions are 500 K, 0.3 g/s and 800 K, 0.5 g/s. The flame is slightly lifted from the porous medium which means that the effect of radial heat transfer within the fluid increases the burning velocity and counters the incoming flow velocity.

The same set of simulations was repeated with equivalence ratio equal to 0.7 and 0.5 and the contours of temperature and velocity are shown in Figure 160, Figure 161 and Figure 165.

![Temperature contours](image1)

**Figure 160 - Contours of temperature for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7**

![Velocity contours](image2)

**Figure 161 - Contours of velocity for the adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7**
Figure 162 - Contours of CH$_4$ for the adiabatic case at reactants’ temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7

Figure 163 - Contours of H$_2$O for the adiabatic case at reactants’ temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7

Figure 164 - Contours of CO$_2$ for the adiabatic case at reactants’ temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7
As expected, at lower equivalence ratios the flame is not able to stabilize at high flow rates especially when reactants’ temperatures are low. In particular at $\phi = 0.7$ and 298 K the flame occurred only when the flow rate is 0.04 g/s and increasing reactants’ temperature up to 800 K shows that combustion happens in every situation considered. At $\phi = 0.5$ the contours in Figure 165 show that combustion is possible only when the mixture is preheated at 800 K and the mass flow rate does not exceed 0.08 g/s, which corresponds to an initial velocity of about 0.71 m/s. In any case flame propagation upstream the flame holder does not occur.

### 6.4.2 Non-adiabatic case

In this section simulations related to the non-adiabatic case are performed and a comparison with the adiabatic case is made. Heat losses to the surroundings are introduced into the model according to the equations written in Table 15, with external temperature of 298 K, convective heat transfer coefficient equal to 12 W/m$^2$K (typical value for natural convection from horizontal cylinders) and emissivity equal to 0.3. In this study, to better highlight heat transfer within the combustion chamber and in particular interactions between fluid, walls and porous medium, a constant value of emissivity typical of polished stainless steel was used. Other works in literature carried out investigation on the effects of material emissivity on heat transfer (e.g. [93]).
Figure 167 - Contours of temperature for the non-adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 1

Figure 168 - Contours of velocity for the non-adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 1

Figure 169 - Contours of CH₄ for the non-adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 1
The contours of temperature, velocity, CH$_4$ mass fraction, H$_2$O mass fraction and CO$_2$ mass fraction at stoichiometric ratio are shown from Figure 167 to Figure 171.

From the contours of temperature in Figure 167 it can be seen how the flame shape changes compared to the adiabatic case. The effect of heat transfer towards the external environment through convection and radiation in general reduces the maximum flame temperature as heat is lost through the side walls to the surroundings. Another major cause of heat loss is the presence of the flame holder, especially at the lowest flow rates where the burning velocity is higher than the flow velocity. At a mass flow rate of 0.04 g/s this happens independently of the mixture’s initial temperature. The maximum flame temperature at reactants’ temperature of 298 K is only 2029 K while the adiabatic case showed a maximum temperature of 2226 K, meaning a decrease of around 10 %. When the mixture is at 800 K,
the maximum temperature reached is 1923 K which is extremely lower (almost 30\%) than the adiabatic case (2466 K). The comparison of maximum temperatures between adiabatic and non-adiabatic conditions is showed in Figure 176, taking as a reference the case where the mixture enters the domain at 800 K and 0.04 g/s mass flow rate. As the flow rate is increased the incoming velocity becomes equal or higher than the burning velocity and the flame gradually moves downstream, reducing the heat lost to the flame holder. At 298 K and 0.08 g/s the maximum temperature is at the centreline and it is slightly higher (2252 K) than the adiabatic flame temperature. This effect is enhanced when the mass flow rate is 0.15 g/s and the flame is lifted about 2.5 mm downstream the holder, in fact in this case the maximum temperature reaches 2363 K. Superadiabatic temperatures in both situations are caused by the recirculation of the heat transferred from the flame to the wall upstream the flame, as pointed out in [90]. Axial conduction in the walls also contributes to raise the temperature in the porous medium and consequently the reactants’ temperature before combustion, allowing superadiabatic conditions and widening flammability limits. The profiles of temperatures showing this effect are presented in Figure 177. When a stretched conical flame is generated (500 K and 0.3 g/s, 800 K and 0.5 g/s), the heat recirculation through the walls allows the flame to be sustained and, despite the non-adiabatic conditions, the adiabatic flame temperature is reached. Despite the superadiabatic temperature reached in the combustor volume, the heat losses to the surroundings are still considerable as they result about 30\% of the heat generated during combustion. The results showing contours of temperature and velocity in the non-adiabatic case at equivalence ratio of 0.7 are presented in Figure 172 and Figure 173. When the equivalence ratio equals 0.5 no stable flame was observed; at a mass flow rate of 0.04 g/s and reactants’ temperature of 800 K the flame seemed to be flickering close to the porous domain, however the solution did not reach convergence and the residuals were too high compared to the targets to consider the solution reliable.

![Figure 172 -- Contours of temperature for the non-adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7](image)
Figure 173 - Contours of velocity for the non-adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7

Figure 174 - Contours of CH₄ for the non-adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7

Figure 175 - Contours of H₂O for the non-adiabatic case at reactants' temperature of 298 K, 500 K, 800 K and equivalence ratio of 0.7
Figure 176 - Temperature profiles at centreline at equivalence ratios of 0.7 and 1, reactants' temperature of 800 K and mass flow rate of 0.04 g/s - Comparison between adiabatic case and non-adiabatic case and mass averaged temperature profiles

Figure 177 - Temperature profiles at centreline at equivalence ratio of 1, reactants' temperature of 298 K and mass flow rate of 0.15 g/s – Comparison between adiabatic and non-adiabatic case

In Figure 178 the axial gradients of temperature in the outer walls and in the porous domain are shown. A comparison is made between two cases at equivalence ratio equal to 1
and reactants’ temperature of 298 K; the flow rates considered are 0.04 g/s (left) and 0.15 g/s (right) as they represent respectively the conditions where heat is lost to the flame holder and where heat is recirculated through axial conduction in the wall. Contours show that the gradient of temperature is maximum when heat is directly transferred from the flame to the porous medium reaching values of 2.22×10^5 K/m. In the other situation, heat is lost from the flame to the lateral wall and the gradient of temperature directed from the porous domain to the fluid downstream has negative value, meaning that heat transfer helps increasing the temperature of the mixture. In this case the highest value of temperature gradient is around 1.62×10^5 K/m and it is located within the porous domain, close to the inlet surface.

![Figure 178 - Contours of temperature gradients in the axial direction in walls and porous medium – non-adiabatic case - equivalence ratio of 1, reactants' temperature of 298 K](image)

Figure 179 presents the evolution of the main species’ molar fraction involved in the combustion at the centreline along the axial coordinate. A comparison is made between adiabatic and non-adiabatic case at equivalence ratios of 1 and 0.7. The mixture mass flow rate is set at 0.04 g/s and the initial temperature is 800 K. As it can be seen from the plots of CH_4, H_2O, CO_2 and O_2, the profiles have same shape and slope in the thin reaction zone just downstream the flame holder. In particular the fuel consumption exhibits the same values in the adiabatic and non-adiabatic case. Temperature plays an important role in completing the chemical reactions, in fact in the non-adiabatic case the decrease in temperature due to heat losses helps the oxidation of the remaining CO. This is clearly shown in the CO and OH graphs, which can be considered as indicators of the reaction progress. As the axial distance increases, the temperature lowers in the non-adiabatic case, reducing to values close to zero the intermediate species (e.g. CO and OH) while it does not happen in the adiabatic case where the adiabatic flame temperature is maintained downstream the flame. As expected these trends are similar to the ones obtained in [93], although the absolute values are somewhat different given the diverse boundary conditions.
6.4.3 Effect of flame holder thermal conductivity

In the previous section the effects of the flame holder on heat exchanges has been illustrated and it was clear that under certain conditions it is helpful to preheat the incoming unburnt mixture. On the other hand at very low flow rates simulations showed that it is the main reason of heat losses from the flame. Therefore, to better understand heat transfer mechanisms, an analysis of the thermal behaviour of the combustor with different porous
material as flame holder was performed and the results are presented in this section. Three different thermal conductivities were considered: 0.5 W/mK, 16 W/mK and 200 W/mK. The wall thermal conductivity was kept at 16 W/mK, typical value of steel. The effects of different wall conductivities were investigated in a previous researches [90,91] and are not considered in the present study.

![Temperature contours](image)

*Figure 180 – Contours of temperature with different porous materials at different reactants’ temperatures – non-adiabatic case – equivalence ratio of 1, mass flow rate of 0.04 g/s*

The direct comparison between the contours of temperature with different porous materials at equivalence ratio of 1 and mass flow rate of 0.04 g/s is shown in Figure 180. The contours of temperature show that there are not considerable differences between the 16 W/mK and 200 W/mK cases, as the maximum temperatures and the flame shapes are nearly identical. When the porous medium thermal conductivity is decreased to 0.5 W/mK, the behaviour is different as the maximum temperature is higher than the other two situations and only a small amount of heat is dispersed through the porous medium. In the first series of simulation, when the reactants’ temperature is 298 K, a low PM thermal conductivity is beneficial as the maximum temperature reaches 2262 K, which is slightly higher than the adiabatic temperature at those conditions. Although the flame is attached to the porous medium, heat is transferred to the outer wall because of the high heat resistance offered by the flame holder and, again through conduction, the mixture gets slightly preheated upstream of the porous insert. The same condition occurs at higher initial temperature. In Figure 181 the temperature profiles comparing the considered conductivities when the initial temperature is 800 K are shown.
Figure 181 – Temperature profiles at centreline with different porous medium conductivities – Equivalence ratio of 1, mass flow rate of 0.04 g/s and reactants’ temperature of 800 K

The insulation provided by the low thermal conductivity material allows the flame to be sustained even when the equivalence ratio is 0.5, under certain conditions. Simulations showed that at 0.5 equivalence ratio combustion occurred at both 0.04 g/s and 0.08 g/s mass flow rates, like in the adiabatic case, and only a small quantity of heat is lost (Figure 182).

6.4.4 Effect of porosity of the flame holder

In this series of simulations the effect of the porosity of the flame holder on the combustion properties and on heat transfer mechanisms is investigated. The porosity of a material is defined as the ratio of the pore volume to the total volume and it ranges from 0 to 1. Therefore it is a measure of the void volume (or solid material) contained in the flame holder and this may affect heat losses through it. Two different porosities are considered, other than the previous predictions which were run at porosity equal to 0.5: 0.1 and 0.9. The results presented are related to the case where the incoming mixture is at 800 K and the mass flow rate is 0.08 g/s with equivalence ratio of 1. The choice of this particular situation is due to the fact that burning velocity is always higher than the incoming flow velocity causing heat losses to the flame holder. Thus, changing porosity with these boundary conditions will show how the temperature profiles and the combustion dynamics vary.
In Figure 183 the results of the simulations run at different mass flow rates and different porosities are presented. It can be seen that increasing porosity has a positive effect on temperature as the void fraction in the porous domain is high; thus the insulating effect is dominant as the mixture’s thermal conductivity is much lower than the one of the solid material.

Considering the case where the mass flow rate is set at 0.08 g/s, the maximum temperatures are: 2180 K when the porosity is 0.1, 2234 K when the porosity is 0.5 and 2399 K when the porosity is 0.9. The trend is the same for different flow rates, excepted for 0.3 g/s and higher where the flame is slightly lifted from the flame holder (about 0.8 mm on average) and heat is mainly lost through the walls.

In conclusion the previous two sections showed that having an insulating flame holder has obvious benefits in terms of heat losses, maximum temperature in the combustion chamber and flame sustainability. In addition to this, reducing the porosity also helps reducing the pressure drop across the porous medium.

6.4.5 Effect of external temperature

Finally a series of simulations with the purpose of investigating the effects of the external temperature on combustion was performed. This could represent the situation where the reactants flow in an annulus surrounding the combustion chamber in the opposite direction of the combustion products (like for example the ones presented in [25,106]). Obviously this model does not fully represent such a counter flow heat exchanger, however it is interesting analysing how the heat transfer is affected when the temperature of the surroundings is higher than the ambient temperature. The study was run at different flow rates.
rates and different equivalence ratios with an initial reactants’ temperature set at 800 K. In this section only the results at equivalence ratio of 0.7 are presented, as they are the most significant, and a comparison is made with the case where the external temperature is set at 298 K.

![Figure 184 – Contours of temperature in the non-adiabatic case with different external temperatures – equivalence ratio of 0.7, reactants’ temperature of 800 K](image)

The effect of the different external temperature on the combustion process can be examined in Figure 184. Contours of temperature at equivalence ratio of 0.7 and different flow rates are shown maintaining the initial mixture temperature at 800 K for both cases. When the simulations are run setting the external temperature at 800 K the amount of heat lost to the surroundings is lower than the case at ambient temperature. As a consequence the maximum temperature in the combustion chamber is higher (around 60 K higher in every case), thus the burning velocity is increased. This phenomenon is clearly visible at 0.3 g/s where the flame occurs at different axial distances from the porous medium. At low flow rates the increased burning velocity “pushes” the flame closer to the flame holder and raises its temperature. Simulations run at 0.5 equivalence ratio showed that the flame is able to be sustained at 0.04 g/s mass flow rate when the external temperature is 800 K, therefore widening the flammability limits.

### 6.5 Summary

Combustion dynamics are strongly affected by heat losses from the flame to the surroundings as they are one of the main causes of flame quenching. The present chapter was aimed at investigating the impact of the flame holder on heat losses and therefore on the combustion process in an 18 mm diameter cylindrical combustion chamber. Two dimensional numerical simulations employing a commercial CFD tool able to solve detailed methane
reaction mechanisms including 17 species and 73 reactions were performed in order to predict temperature, velocity and species’ concentration profiles. At small scale numerical simulations play an important role as experimental measurements may result very difficult. The adiabatic case was successfully validated and taken as a reference. The rest of the simulations were executed to model non-adiabatic conditions and different parameters such as flame holder’s thermal conductivity and porosity were varied. Also the effect of increasing the external temperature was analysed and combinations of several mass flow rates, equivalence ratios and mixture’s initial temperatures were considered. Results showed that at low flow rates a high amount of heat is dispersed through the flame holder as the burning velocity is much higher than the incoming flow velocity. To limit this loss a low thermal conductivity material and a high porosity is beneficial. These two physical parameters also have influence on widening flammability limits and on stabilizing the flame. At higher flow rates when the flame is lifted from the porous medium, heat is mainly transferred to the walls and, through axial conduction, to the flame holder which in turn contributes preheating the reactants, increasing the maximum temperature achieved in the combustor. Imposing an external temperature of 800 K reduces heat transfer to the surroundings and this increases burning velocity, allowing better flame stability and higher flow rates to flow in the combustion chamber.
NUMERICAL PREDICTION OF THERMAL PERFORMANCES IN A TRIPLE TUBE HEAT EXCHANGER

As a complement of the present research, a study based on two dimensional numerical simulations of a three concentric tubes recuperator involving combustion in the inner tube is presented.

7.1 Introduction

The importance of increasing the initial temperature of the mixture before combustion in micro combustors has been highlighted in several researches [10,25,38,107,108] which show the positive effect not only on fuel consumption, therefore on Brayton cycle efficiency concerning gas turbines, but also on flammability limits and flame sustainability at micro scale. The benefits of preheating reactants in terms of widening the flammability limits have been also extensively numerically studied in previous researches [86,88,90-93]. In Ultra-Micro Gas Turbine (UMGT) systems a major problem is represented by the huge heat losses to the environment from the hot parts of the device. It has been proven that heat losses are dependent on the inverse of the hydraulic diameter [18]:

\[ \frac{Q_{lost}}{Q_{gen}} \propto \frac{1}{D_{Hydr}^{1.2}} \]  

(69)

A convenient method to exploit in a useful way, the heat lost to the surroundings, is to recirculate the combustion products for heat extraction to increase the temperature of the mixture before combustion. This practice has been used in recent studies on micro and meso scale combustion chambers [1,25-27,38,109], including the present experimental work. Using a three concentric tube configuration represents a suitable and compact configuration to exchange heat between three flows.

Studies on “triple tube” heat exchangers in the literature are extensive and they include experimental, numerical and analytical approaches. One of the first formulations was proposed in [110] for a steady counter-current flow configuration where the cold medium
flowed through the inner annulus and the heating fluids flowed through the inner tube and the outer annulus. A mathematical model was developed including a thermal energy balance in the axial direction and the equations were solved by applying the Laplace transform approach. A few simulations in the turbulent regime were run employing water at 15 °C as the cooling medium for a liquid from a food product to be cooled at 30 °C from 80 °C. Results showed an increased overall heat transfer coefficient with the insertion of an annular region within the inner pipe and a reduction of the overall length of around 25 

Another study on a similar triple tube heat exchanger was performed in [111,112] where the coupled heat transfer equations were reduced into a set of second order differential equations and a mathematical formulation was presented. The assumption made included perfect insulation of the system from the surroundings, thermally and hydrodynamically developed flows and constant fluid properties. In the second part of the work several case studies were considered by changing the mass flow rate of the fluids and changing the radii ratios of the tubes. It was concluded that the ratio between the radii of the tubes is very important with respect to the heat exchanger performances and the improvement with respect to a double tube heat exchanger with the same characteristics was shown.

In [72] an approach that is able to compute the overall heat transfer coefficients for an insulated triple tube heat exchanger was proposed. The analysis was carried out for both co-current and counter-current flows and the aim was to cool a fluid flowing in the inner tube and in the outer annulus using a hot fluid passing through the inner annulus. Also in this case, the equations governing heat transfer were reduced to a set of second order differential equations and applying the boundary conditions, the overall heat transfer coefficients were obtained. Furthermore, the axial temperature profiles were plotted for different flow configurations. In a following publication [73] the same authors investigated experimentally the triple tube system and compared the results with the analytical model. They found that in certain cases (co-current runs) there was a “crossover” point where the temperature of the hot fluid in the outer annulus became equal, and then lower, than the temperature of the cooling fluid, thus decreasing the overall efficiency. They also formulated a further equation to compute the point of crossover. Summary of results showed that the counter-current arrangement provided better performance over the co-current configuration. Also, in a co-current configuration, utilizing a doubletube system in some cases yields better performance than a triple tube heat exchanger, due to the occurrence of the “crossover” in the temperatures.
Another mathematical approach has been solved in [113] where the fundamental equations were discretized using Galerkin’s weighted residual method and results in terms of dimensionless temperature and length were presented using water as the fluid. Assumptions included steady state, constant mass flow rates, un-insulated system and different inlet temperatures for the three pipes. Values for the overall heat transfer coefficients were taken from known correlations.

A non-insulated triple tube heat exchanger designed for dwellings and able to recover wood burning domestic flues has been analysed theoretically in [114]. Combustion flue gases flowed in the inner tube, ventilation air in the inner annulus and combustion air flowed through the outer annulus. A counter-current arrangement was modelled between flue gases and ventilation air and a co-current configuration between ventilation air and combustion air. In the formulation, local heat transfer coefficients were taken from correlations in the literature and radiation was also considered, to calculate an average radiative heat transfer coefficient. The governing equations were put in a dimensionless form and a parametric study was performed with varying length and adding fins to the tubes. Results were presented in terms of Number of Transfer Unit and effectiveness; comparisons with the experimental tests on a rig developed by the same author showed differences in heat transfer rates of lower than 5 %.

The recent development of small scale systems has introduced a new research topic represented by heat transfer in small channels through flowing fluid and walls. When reducing dimensions, the ratio between wall thickness and channel diameter increases and this affects heat transfer. In the literature there are several studies [115-122] on heat transfer at micro scale (sub-millimeter diameter channels) pointing out the importance of axial conduction through walls, especially at low Reynolds numbers, which results in an increase in the fluid temperature. As a consequence of axial conduction in the walls, the Nusselt number is decreased when compared to those derived from known correlations [123]. It is noted that for a fully hydrodynamically and thermally developed laminar flow in a circular duct, the Nusselt number is a constant number if axial conduction is neglected: it equals 4.36 in the case of constant heat flux boundary condition applied at the wall and 3.66 in the case of constant wall temperature.

A dimensionless number called \textit{axial conduction number} or \textit{Maranzana number} (from the author’s name) to determine whether axial conduction through walls can be neglected is proposed in [118]. In this work an exact analytical model able to predict fluid temperature
profile along the axial coordinate including axial conduction is developed considering the flow between two parallel plates. The axial conduction number is defined as the ratio between heat transfer through axial conduction in the wall and convective heat transfer in the flow:

$$M = \frac{\Phi_{\text{cond}}}{\Phi_{\text{conv}}} = \frac{r^2 NTU}{Bi} \quad (70)$$

where $r$ is the ratio between wall thickness and wall length, $Bi$ is the Biot number based on wall thickness and $NTU$ is the number of transfer units. Results of different simulations allowed the author to state that when $M \leq 0.01$ the axial conduction effects can be neglected. Also this work showed that an optimal wall thermal conductivity exists that is able to maximize heat exchanger efficiency.

Another analytical model with the aim of comparing the calculated Nusselt number with other numerical simulations and experimental works was proposed in [119]. The model is built starting from the energy balance of a control volume on the wall with constant heat flux on the outside surface and allows the estimation of the fluid temperature at any section. The equation obtained shows the ratio between the Nusselt number including axial conduction and the Nusselt number neglecting it:

$$\frac{Nu_{k_0}}{Nu} = \frac{1}{1 + \frac{k_s A_s Nu}{k_f A_f (RePr)^2}} \quad (71)$$

where $k_s$ and $k_f$ are the solid and fluid thermal conductivities, $A_s$ and $A_f$ are the cross sectional areas of the solid and the fluid. It can be noticed how increasing the solid cross sectional area and solid thermal conductivity contributes to increasing axial conduction heat transfer. On the other hand a higher value of the Reynolds number means a lower axial conduction heat transfer. In the same publication several comparisons in terms of Nusselt number are made between the model including equation (71) and some numerical and experimental work on microchannels with different shapes and boundary conditions (such as [115,118,124,125]). Results showed very good agreement with the reference literature, providing a reliable validation. Also a parametric study was performed by changing several parameters including the type of fluid (water and air), Reynolds number, diameter of the channel and ratio of the cross sectional area of the solid to that of the fluid. Axial conduction effects in the walls were proven to be much more relevant when the flowing fluid is a gas.
rather than water and also, as mentioned previously, wall thermal conductivity and wall thickness play an important role in axial conduction.

The literature regarding small scale concentric tube heat exchangers with internal combustion is very limited. There are a few experimental researches on combustors incorporating different configurations of heat exchangers \([22,25,26,106,126]\), such as the “swiss roll” combustor, the “can” type combustor and a cylindrical combustor surrounded by a stacked plates recuperator.

<table>
<thead>
<tr>
<th>Author</th>
<th>Exchanger Geometry</th>
<th>Fluid medium</th>
<th>Methodology</th>
<th>Object of study</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gomaa et al.</td>
<td>Three concentric plain cylinders not insulated from surroundings, Counter-current flows</td>
<td>Hot liquid flows in the inner annulus. Cold liquid flows in the inner tube and outer annulus</td>
<td>Analytical formulation solving a system of second order ordinary differential equations</td>
<td>Parametric study varying mass flow rate and radii.</td>
<td>Relative size of the tubes affects thermal performances. Comparison with a double tube system with same outer diameter</td>
</tr>
<tr>
<td>Batmaz</td>
<td>Three concentric plain cylinders insulated from surroundings. Counter-current flows and co-current flows</td>
<td>Hot fluid flows in the inner annulus. Cold fluids flow in the inner tube and outer annulus</td>
<td>Analytical formulation to solve for the overall heat transfer coefficients</td>
<td>Case study aimed at plotting temperature profiles using water and propylene glycol as fluids</td>
<td>Overall heat transfer coefficient was calculated and temperature profiles plotted. The effectiveness of a triple tube system was showed to be higher than that of a double tube system</td>
</tr>
<tr>
<td>Quadir</td>
<td>Three concentric plain cylinders not insulated from surroundings. Co-current and counter-current flows</td>
<td>Hot fluid flows in the inner annulus. Cold fluid flows in the outer annulus and normal fluid flows in the inner tube</td>
<td>Analytical formulation using Galerkin’s weighted residuals method. Global matrix solved and dimensionless boundary conditions are applied</td>
<td>Comparison of mathematical model with experimental data. Parametric study varying radii and fluid properties</td>
<td>Results from analytical formulation match the experimental data. Effects of NTU on temperature profiles are evaluated</td>
</tr>
<tr>
<td>Lin, Kandlikar</td>
<td>Several microchannels geometries considered</td>
<td>Air and water at different Reynolds numbers</td>
<td>Development of an equation of Nusselt number including the effect of axial conduction in the walls</td>
<td>Comparison of the proposed analytical model with different experimental and numerical works in literature</td>
<td>Parametric analysis varying Re, area ratio and fluid showed the importance of axial conduction through walls</td>
</tr>
<tr>
<td>Zuritz</td>
<td>Three concentric plain cylinders</td>
<td>Cooling water flowing in the inner annulus. Hot process products in the inner tube and outer annulus</td>
<td>Analytical formulation including Laplace transform</td>
<td>Design problem. Determination of exchanger length and heat transfer rates</td>
<td>Comparison between formulations showed good match</td>
</tr>
<tr>
<td>Maranzana</td>
<td>Parallel plates with thickness of 500 µm and channels 100 µm wide</td>
<td>Cold water at 10°C, hot water at 60°C</td>
<td>Analytical formulation using quadrupole method</td>
<td>Studying the influence of axial conduction in the walls on the exchanger efficiency</td>
<td>Development of two analytical models including axial conduction through walls. Introduction of non-dimensional number M</td>
</tr>
<tr>
<td>Gemaa et al.</td>
<td>Three concentric plain cylinders</td>
<td>Chilled water in the inner tube, hot water in the inner annulus and tap water in outer annulus</td>
<td>Comparison between experimental and analytical results</td>
<td>Parametric study varying flow pattern, annulus spacing, hot water temperature</td>
<td>Development of correlations for Nu, friction factor and heat exchanger effectiveness</td>
</tr>
</tbody>
</table>
The experimental work presented in [25] shows the tests on a prototype combustion chamber made up of three plain concentric tubes with flat flame combustion in the inner tube. The combustion chamber diameter was 14 mm and the outer diameter was 28 mm. The cold fuel-air mixture flows into the inner cylinder and the combustion products are recirculated through the outer cylinder in order to maximize the heat transfer to the reactants. The paper reports the experimental results for a heat release ranging from 11 W to 244 W, using methane as a fuel. Emissions were detected and clean combustion was achieved, with over 99% of combustion efficiency under lean conditions. Also the reactants temperature before combustion was measured and results showed that at 0.06 g/s air mass flow rate and an equivalence ratio of 0.85, it reached 795 K. At this elevated temperature, the burning velocity increases considerably and consequently the flammability limits are widened as well [93].

Despite the above mentioned study, no other studies are present in literature on micro scale three concentric tubes combustion chambers including a concentric heat exchanger. In Table 18 a recap of the studies related to concentric tubes heat exchangers is summarized.

As highlighted previously, in UMGTs heat losses are a major problem and, in order to utilize the enthalpy contained in the combustion products, the most feasible solution is to recirculate the exhaust gases to increase the temperature of the mixture before combustion. This also results in higher burning velocity allowing a higher flow rate, therefore higher power output. In the literature no numerical works including a detailed combustion mechanism in a triple tube heat exchanger were found. Therefore, the purpose of this study is to numerically investigate the thermal exchanges in a cylindrical combustion chamber externally surrounded by annuli acting as a recuperator, and utilising a detailed reaction mechanism for simulating combustion. After validating the heat transfer model against analytical formulation that incorporates axial conduction through the walls, attention will be paid to the differences between a three concentric tubes configuration rather than a two concentric tubes configuration (without recirculating the combustion products). Also thermal exchanges from varying wall thermal conductivity and the effects of including a flame holder in the combustion chamber on heat transfer mechanisms will be investigated numerically.

## 7.2 Numerical model

The commercial Computational Fluid Dynamics code ANSYS Fluent 17 [128] was used for the present simulations. With this tool it is possible to model flows, mass, momentum,
energy and heat transfer equations, with the possibility of importing detailed reaction mechanisms. In this case a 17 species and 73 reactions mechanism [98] able to describe methane oxidation was used for the simulations along with the CHEMKIN thermodynamic and transport properties database [96,97]. The validity of this mechanism has been widely checked against experimental data in previous researches [93,129] in terms of adiabatic flame temperature, flame thickness, reaction zone characteristics and composition of combustion products therefore in this study a further comparison is not considered necessary. Simulations were completely run on the NeSI Pan Cluster [99] at the University of Auckland employing 24 cores for each simulation which usually took around two days to reach convergence. The governing equations used by Fluent are shown in Table 19.

### Table 19 – Governing equations [130]

<table>
<thead>
<tr>
<th>Equation</th>
<th>Description</th>
</tr>
</thead>
</table>
| **Continuity (2D axi-symmetric):**          | \(
\frac{\partial}{\partial x} (\rho v_x) + \frac{\partial}{\partial r} (\rho v_r) + \frac{\partial}{\partial r} (\rho v_r v_r) = 0
\) |
|                                                      | \( \frac{1}{r} \frac{\partial}{\partial r} (r \rho v_r v_r) + \frac{1}{r} \frac{\partial}{\partial r} (r \rho v_r v_r) \) |
|                                                      | \( = - \frac{\partial p}{\partial x} + \frac{1}{r} \frac{\partial}{\partial r} \left[ \rho \left( \frac{\partial v_x}{\partial x} + \frac{2}{3} (\nabla \cdot \mathbf{v}) \right) \right] \) |
|                                                      | \( + \frac{1}{r} \frac{\partial}{\partial r} \left[ \rho \left( \frac{\partial v_x}{\partial r} + \frac{\partial v_r}{\partial x} \right) \right] \) |
| **Momentum conservation:**                   | \( \frac{\partial}{\partial x} (r \rho v_x v_r) + \frac{1}{r} \frac{\partial}{\partial r} (r \rho v_r v_r) \) |
|                                                      | \( = - \frac{\partial p}{\partial x} + \frac{1}{r} \frac{\partial}{\partial r} \left[ \rho \left( \frac{\partial v_x}{\partial x} + \frac{2}{3} (\nabla \cdot \mathbf{v}) \right) \right] \) |
|                                                      | \( + \frac{1}{r} \frac{\partial}{\partial r} \left[ \rho \left( \frac{\partial v_x}{\partial r} + \frac{\partial v_r}{\partial x} \right) \right] - 2 \mu \frac{v_r}{r^2} + \frac{2 \mu}{3} (\nabla \cdot \mathbf{v}) \) |
| **Energy in fluid regions:**                 | \( \nabla \cdot (\mathbf{v} (\rho E + p)) = \nabla \cdot \left[ k \nabla T - \sum_i h_i J_i + (\bar{\tau}_{\text{eff}} \cdot \mathbf{v}) \right] + S_h \) |
| **Energy in solid regions:**                 | \( \nabla \cdot (\rho \mathbf{v} h) = \nabla \cdot (k \nabla T) \) |
| **Species:**                                  | \( \nabla \cdot (\rho \mathbf{v} Y_i) = - \nabla \cdot \hat{J}_i + M_{\text{mol}} \sum_{r=1}^{N_R} \hat{R}_{i,r} \) |
| **Convective heat transfer:**                | \( q_c = \hat{h}_c (T_{\infty} - T_{\infty}) \) |
| **Radiative heat transfer:**                 | \( q_r = \varepsilon \sigma (T_{\infty}^4 - T_{\infty}^4) \) |
| **DO radiative model:**                     | \( \nabla \cdot (I(i, \mathbf{s}) \mathbf{s}) + (a + \alpha_g) I(i, \mathbf{s}) = a n^2 \frac{\sigma T^4}{\pi} + \frac{\sigma}{4 \pi} \int_0^{4 \pi} I(i, \mathbf{s}) \phi(s) d\Omega \) |

In Table 19, \( \bar{\tau}_{\text{eff}} \) is the stress tensor, \( J \) is the diffusion flux of species \( i \) and \( j \), \( S_h \) is a source term including the heat of chemical reaction defined according to molecular weight of
species $M_{w,t}$ and Arrhenius molar rate of creation or destruction of species $\hat{R}_{i,r}$, $\alpha$ is the absorption coefficient, $\sigma_s$ is the scattering coefficient, $n$ is the refractive index, $I$ is the radiation intensity which depends on the position and direction vectors $\vec{r}$ and $\vec{s}$, $\phi$ is a phase function, $\vec{s}'$ is the scattering direction vector and $\Omega'$ is the solid angle [130]. The pressure loss across the porous medium is taken into account in the governing equations by adding a momentum source term (equation (72)), made up of a viscous loss term and an inertial loss term.

$$S_i = -\left(\frac{\mu}{\alpha} v_i + C_2 \frac{1}{2} \rho |v| v_i \right)$$  \hspace{1cm} (72)

In equation (72) $\mu$ is the molecular viscosity, $\alpha$ is the permeability, $C_2$ is the inertial resistance factor and $v$ is the velocity. Experimental data obtained by Turkeli-Ramadan [12] were used to obtain values for permeability and inertial resistance. Heat transfer through the porous medium is solved by Fluent [131] considering an effective thermal conductivity $k_{\text{eff}}$ which depends on the porosity $\gamma$ (kept at 0.5 in this study), as shown in equation (63), with $k_f$ and $k_s$ being the thermal conductivity of the fluid and the solid respectively. The porous material is assumed to be stainless steel with a thermal conductivity of 16 W/mK. The effect of varying the thermal conductivity and porosity on combustion and heat transfer has been previously investigated numerically [129] and it is not an objective of this study.

$$k_{\text{eff}} = \gamma k_f + (1 - \gamma) k_s$$  \hspace{1cm} (73)

The simulations are run using the pressure based solver with the SIMPLE algorithm for the pressure-velocity coupling and second order discretization for the governing equations. The Laminar model is enabled as well as the CHEMKIN volumetric reactions solver and to simulate radiation the Discrete Ordinates model was used, setting the walls external emissivity to 0.2 (typical of stainless steel). Ignition was achieved by patching a high temperature zone in the combustion chamber after running 200 iterations without enabling reactions to stabilize the flow. Residuals target was set at $1 \times 10^{-6}$ for continuity, velocity, energy and species concentrations. One of the parameters that will be changed in this study is the walls thermal conductivity and two values will be considered: $0.5$ W/mK and $16$ W/mK. In the first series of simulations the effect of changing the wall thermal conductivity will be shown; subsequently only the effect of varying the external wall thermal conductivity (maintaining the inner walls conductivities at $16$ W/mK) will be presented. Simulations are
performed in steady state. Comparison in terms of thermal characteristics will be made against the adiabatic case and in the final section the effects of introducing a 1 mm thick stainless steel porous medium as flame holder will also be presented. The porous holder is placed at 3 mm from the inlet in a way that the effects of axial conduction through the wall can be visualized. Equivalence ratio of the mixture will be kept equal to 1 and three different mass flow rates will be considered: 0.05 g/s, 0.1 g/s and 0.15 g/s. Operating pressure is kept at atmospheric conditions, thus 101,325 Pa, and the external temperature is set at 298 K with a convective heat transfer coefficient of 12 W/m²K, as suggested by proven experimental equations for natural convection from horizontal cylinders [64]. In order to transfer the average temperature from the outlet of the inner cylinder to the inlet of the combustor a user defined function was written and implemented. The same procedure has been adopted to transfer the average temperature at the combustor outlet to the outer cylinder inlet with the only difference that the temperature was decreased by 200 K (according to Poisson’s equation for gases expansion), in order to simulate the expansion in a hypothetical turbine. The models used in the simulations are shown from Figure 185 to Figure 188. Being an axially symmetric geometry, only half the domain was modelled and the axis boundary condition was applied.

![Figure 185 - Triple concentric tube heat exchanger model](image)
A mesh sensitivity analysis was performed on the simple triple concentric tube non adiabatic model at a mass flow rate of 0.05 g/s considering the following element sizes: 0.1 mm, 0.05 mm, 0.025 mm and 0.01 mm. Results in terms of heat released during combustion, maximum combustion temperature in the inner cylinder and mass-weighted temperatures at the inner and outer cylinder outlets (corresponding to reactants preheating temperature and exhaust gases temperature) are considered.
Table 20 – Sensitivity analysis

<table>
<thead>
<tr>
<th>Grid size</th>
<th>0.1 mm</th>
<th>0.05 mm</th>
<th>0.025 mm</th>
<th>0.01 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reactants preheating temperature</td>
<td>1082 K</td>
<td>1079 K</td>
<td>1077 K</td>
<td>1081 K</td>
</tr>
<tr>
<td>Exhaust gases temperature</td>
<td>865 K</td>
<td>873 K</td>
<td>871 K</td>
<td>867 K</td>
</tr>
<tr>
<td>Heat released during combustion</td>
<td>136.5 W</td>
<td>136.4 W</td>
<td>136.3 W</td>
<td>136.3 W</td>
</tr>
</tbody>
</table>

As it can be seen in Table 20, difference between the values with each mesh refinement is lower than 1 % in every case so the 0.1 mm element size is chosen for the simulations to reduce the demand on computational resources.

7.3 Model validation

As mentioned in the previous section, in the literature there are different experimental and numerical works on concentric tube heat exchangers, however no data has been found for small scale concentric tube heat exchangers incorporating a combustion chamber in the inner cylinder. Therefore the validation of the heat exchangers in this model is done against an analytical model which includes axial conduction [119] and which has been proven to closely match several experiments. To perform the analytical runs, EES (Engineering Equation Solver) [132] has been employed, which is a general equation-solving program able to solve coupled non-linear and differential equations. Combustion was not modelled in EES; instead the mass weighted average temperature at the section at 1 mm from the inlet was considered as a boundary condition, as it is the section at the highest temperature in the combustion chamber. The validity of the model is first tested on a single cylinder with 9 mm diameter, 80 mm length (to allow the flow and the reactions to be fully developed) and a 1 mm thick external wall and then it will be extended to the triple concentric tube exchanger configuration. The governing equation to be input in to EES describes the energy balance in the fluid domain (74):

\[ \dot{m}c_p \frac{dT}{dx} = -U_1 P_1 dx(T - T_\infty) \]

where \( U_1 \) is the overall heat transfer coefficient defined as:

\[ U_1 = \left( \frac{1}{h_{1k} r_{1i}} + \frac{r_{ext} \ln(r_{ext}/r_{1i})}{k_s} + \frac{1}{h_{ext}} \right)^{-1} \]
with \( h_{1k} \) and \( h_{\text{ext}} \) being the heat transfer coefficients related to the flow in the cylinder and the surroundings respectively and \( k_s \) the thermal conductivity of the wall. Other than the convective heat transfer coefficient, a radiative heat transfer coefficient equal to 7 W/m\(^2\)K was also included in \( h_{\text{ext}} \) to take into account radiation losses from the wall to the environment. The area averaged temperature of the external wall from the CFD simulation was considered a good approximation for the calculation. The heat transfer coefficient for the internal flow was estimated by combining expressions of Nu including thermal entry effects according to the equation developed by Stephan [133], and shown in (76),

\[
Nu = \frac{3.66 \tanh[2.264 Gz^{-1/3} + 1.7 Gz^{-2/3}] + 0.0499 Gz \tanh(Gz^{-1})}{\tanh(2.432 Pr^{1/6} Gz^{-1/6})}
\]

where \( Gz \) is the Graetz number defined as:

\[
Gz = \frac{D}{x} RePr
\]

Subsequently \( Nu_k \), which is the Nusselt number modified to include axial conduction, was calculated using equation (78) [119]:

\[
\frac{Nu_k}{Nu} = \frac{1}{1 + 4 \frac{k_s A_s Nu}{k_f A_f (RePr)^2}}
\]

In the CFD simulations the combustion reactions started occurring at 1 mm from the inlet, thus the mass-weighted average temperature of the fluid at that section was taken as inlet temperature to input in EES. Comparisons of temperature profiles and Nu are shown in Figure 189.
Nusselt number from the computational study was obtained directly from the temperature distribution using equation (79):

$$Nu_{CFD} = \frac{q}{k} \frac{D}{(T_{y^+}(x) - T_\infty(x))}$$

where $T_{y^+}$ and $T_\infty$ are the temperature distribution along the axial coordinate at a radial distance of 0.1 mm from the wall and the temperature at the centreline respectively, $k$ is the thermal conductivity of the fluid at $y^+$ (equal to 0.1 mm) and the difference $(y_{y^+} - y_\infty)$ represents the characteristic dimension which is the distance between the centreline of the considered tube or annulus and the nearest layer to the wall (at 0.1 mm from the wall itself).

It can be seen from Figure 189 that the temperature profile obtained from the CFD simulation matches very closely the analytical formulation with axial conduction. The fluid domain was divided in 30 different planes equally spaced along the axial coordinate and the mass-weighted average temperature on each plane was computed and plotted in Figure 189. The plot on the right shows Nu obtained from the numerical and analytical simulations. The thermal entry effect causes Nu to be extremely high at the inlet and then decreases asymptotically to reach a constant value, which is 4.381 at the outlet (green line).

When including axial conduction effects, the value of Nu is lowered (red line), as previously mentioned, and the plot in this case matches the results from the CFD simulation (blue line). The difference between the two profiles is in the thermal entry region, from the inlet to an axial distance of around 6 mm from it. This behaviour can be attributed to the
combustion reaction which suddenly increases the flow velocity, therefore the heat transfer conditions; however the discrepancy can be considered acceptable. Values of Nu at the outlet are 2.35 and 2.54 for the CFD simulation and analytical model respectively and the relative difference is never higher than 7.5 %. The plot also shows the inaccuracy of the model utilizing Nusselt number excluding axial conduction through the walls (green line in Figure 189).

The same approach has been adopted on the simple triple tube heat exchanger. In this case the set of coupled differential equations shown in (80), (81) and (82) has to be solved

\[ \frac{d}{dx} \left( \dot{m} c_{p, hot} T_h \right) = -U_1 P_1 (T_h - T_c) \]  
\[ \frac{d}{dx} \left( \dot{m} c_{p, cold} T_c \right) = -U_1 P_1 (T_h - T_c) - U_2 P_2 (T_ex - T_c) \]  
\[ \frac{d}{dx} \left( \dot{m} c_{p, ex} T_ex \right) = U_2 P_2 (T_ex - T_c) + U_3 P_3 (T_ex - T_\infty) \]

where the subscripts h, c and ex refer to the hot flow (in the inner tube), cold mixture (in the inner annulus) and exhaust gases flow (in the outer annulus) flow respectively. The overall heat transfer coefficients become in this case:

\[ U_1 = \frac{1}{\left( \frac{r_1}{h_{1k} r_{2i}} + \frac{r_{2i} \ln(\frac{r_{2i}}{r_1})}{k_s} + \frac{1}{h_{2k}} \right)} \]  
\[ U_2 = \frac{1}{\left( \frac{1}{h_{2k}} + \frac{r_{3i} \ln(\frac{r_{3i}}{r_{2o}})}{k_s} + \frac{r_{2o}}{h_{3k} r_{3i}} \right)} \]  
\[ U_3 = \frac{1}{\left( \frac{r_{3i}}{h_{\infty} r_{ext}} + \frac{r_{ext} \ln(\frac{r_{ext}}{r_{3i}})}{k_s} + \frac{1}{h_{3k}} \right)} \]

Subscripts 1k, 2k and 3k refer to the inner tube, middle tube and outer tube respectively. For the purpose of validating the triple tube heat exchanger model the temperatures at the combustor inlet and at the outer channel inlet were set at 800 K and 1100 K respectively. The comparison between the temperature profiles in the three channels is shown in Figure 190 and the contour maps of temperature are presented in Figure 191.
Figure 190 – Comparison of temperature profiles between analytical formulation and numerical study – 0.05 g/s total mass flow rate, equivalence ratio equal to 1

Figure 191 - Contours of temperature – 0.05 g/s total mass flow rate, 1 ER equivalence ratio
The temperatures obtained from the two methods of calculation are very similar and the maximum difference of temperature in any case is lower than 5%. The discrepancies can be attributed to the combustion occurring in the inner cylinder which might affect the heat exchanges and flow properties, however the CFD model can be considered accurate enough under this point of view. As mentioned above, the methane reaction mechanism is not validated in this paper as its check was performed by the same authors in a previous work [129].

7.4 Results and discussion

In this section the results of the Computational Fluid Dynamic simulations are presented. To have a reference to compare the simulations to, a series of adiabatic simulations is performed first, with two different thermal conductivity of the walls (16 W/mK which corresponds to a typical value for stainless steel and 0.5 W/mK which corresponds to a typical value for a ceramic material) and three total mass flow rates (0.05 g/s, 0.1 g/s and 0.15 g/s).

7.4.1 Adiabatic case

In this section the reference case is presented. Adiabatic conditions are imposed on the external walls of both the triple tube and the doubletube heat exchangers and simulations are run with varying thermal conductivity of the walls. Thermal conductivity, as can be seen from equation (78), has an important role in axial conduction through the walls, therefore it might affect heat transfer between the flows in the cylinder and annuli. In Figure 192 and Figure 193 contours of temperature for the adiabatic case are presented. Images are not to scale.
Figure 192 – Contours of temperature at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – adiabatic case
As can be seen from the contours in Figure 192, the temperature has a parabolic profile stretched towards the axial coordinate. As the mass flow rate is increased the area surrounding the centreline, where the temperature is maximum, becomes larger and longer because of the increased heat released during combustion and at 0.15 g/s total flow rate, the highest temperatures extend to the downstream boundary of the combustor tube. Also the maximum temperatures in the inner cylinder for any of the cases do not reach the adiabatic flame temperature, which in the worst case corresponds to around 2608 K (at reactants’ preheating temperature of 1164 K corresponding to 0.05 g/s flow rate in Figure 192). This behaviour is due to the heat transferred to the cold reactants, which is considerable especially
in the initial part of the cylinder, where the thermal entrance effect has a great influence. Nusselt number from the CFD simulations was also plotted and the graphs are shown in Figure 194.

The preheating temperatures of reactants in the adiabatic simulations of the triple tube system are shown in Table 21.
Table 21 – Preheating temperature of reactants and temperatures at the outer channel outlet at different mass flow rates and different conductivities in a triple tube system – adiabatic case

<table>
<thead>
<tr>
<th></th>
<th>Exhaust temperature at outlet</th>
<th>Reactants’ preheating temperatures</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.5 W/mK</td>
<td>16 W/mK</td>
</tr>
<tr>
<td>0.05 g/s</td>
<td>1031 K</td>
<td>945 K</td>
</tr>
<tr>
<td>0.1 g/s</td>
<td>1140 K</td>
<td>1113 K</td>
</tr>
<tr>
<td>0.15 g/s</td>
<td>1260 K</td>
<td>1225 K</td>
</tr>
</tbody>
</table>

In Figure 194 profiles of Nusselt number in the three-concentric tubes combustor are shown at both 0.5 W/mK and 16 W/mK. In the first row Nu in the inner cylinder is shown, in the second and third rows the Nusselt number profiles in the inner and outer annulus respectively are plotted. As it can be seen, in the combustion chamber Nu reaches values of around 80 at the initial section and rapidly decreases as the flow develops as a result of the entrance effect on the laminar flow. Along the tube, it asymptotically stabilizes to a constant value which is 2.6 when the total flow rate is 0.05 g/s and 4.1 with 0.15 g/s total mass flow rate. Theory shows that the asymptotic value is 4.36 when a constant heat flux is applied to the cylindrical combustion chamber and 3.66 when the wall is kept at a constant temperature [64]. However, these values do not take into account elevated temperature differences, axial conduction through the wall nor radiation. In the inner annulus the profiles show the same behaviour as of the combustor, where Nu is high at the inlet and then decreases to a constant value of around 1 towards the outlet. In the outer annulus the profiles show peaks along the axial coordinate and this could be due to the temperature at 0.1 mm radial distance from the wall \( T_y + (x) \) and the temperature of the fluid at the centreline being equal, therefore, according to the mathematical formulation used in equation (79), Nu shows a sudden increase. The peaks of Nu are located at different axial coordinates depending on the mass flow rate and it can be seen that as the flow rate increases the peak shifts towards the outlet. The plots of temperatures in Figure 195 show the profiles of temperature of the reactant mixture flowing in the inner annulus and the profiles of temperature of the exhaust gases flowing in the outer annulus in adiabatic conditions at mass low rates 0.05 g/s (left) and 0.1 g/s (right), with walls thermal conductivity of 0.5 W/mK.
Figure 195 – Profiles of temperatures of cold mixture in the inner annulus and exhaust gases in the outer annulus along the axial coordinate – 0.5 W/mK; 0.05 g/s (top), 0.1 g/s (center) and 0.15 g/s (bottom)
These graphs are useful to understand the relation between the peaks of Nusselt number shown in Figure 194 and the temperature trend in the channels. It can be seen that the temperature profiles meet at the same point where Nu shows a peak, as a result of the mathematical formulation used (79). Also, it can be noticed that Nu becomes very low (around 0.04) when considering the outer wall of the outer annulus as a result of the adiabatic conditions imposed on the boundary, therefore the only heat transfer occurring is because of axial conduction through the external wall.

Previous researches by Batmaz et al. [72,73] also showed that the cross over point can occur in a triple concentric tubes system and they developed an analytical solution for temperature profiles in each channel including the location of the cross over point. Knowing the exact location of this point can help in the design of the system, in particular deciding upon the length of the tubes.

The same set of simulations was performed on a double tube heat exchanger with internal combustion, where cold mixture flows in the annulus surrounding the inner tube and no recirculation of combustion products is present. From the contours of temperature in Figure 196 it can be seen that in this case as well, the adiabatic flame temperature is not reached in the combustion chamber for any of the cases due to heat transfer from the inner cylinder to the annulus via the tube wall. Velocity contours are shown in Figure 197 and it can be seen that at 0.15 g/s total mass flow rate the Reynolds number is around 70 at the annulus outlet and around 96 at the outlet of the combustion chamber, therefore the flow conditions are always laminar.

Preheating temperatures at the annulus outlet are shown in Table 22.

<table>
<thead>
<tr>
<th>Mass Flow Rate</th>
<th>0.5 W/mK</th>
<th>16 W/mK</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05 g/s</td>
<td>904 K</td>
<td>940 K</td>
</tr>
<tr>
<td>0.1 g/s</td>
<td>869 K</td>
<td>912 K</td>
</tr>
<tr>
<td>0.15 g/s</td>
<td>875 K</td>
<td>871 K</td>
</tr>
</tbody>
</table>
Figure 196 – Contours of temperature at different flow rates, equivalence ratio of 1 and different wall thermal conductivities

$k=0.5 \text{ W/mK}$

$k=16 \text{ W/mK}$

0.05 g/s

0.1 g/s

0.15 g/s
Figure 197 - Contours of velocity at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – adiabatic case

Nusselt number profiles along the axial coordinate are shown in Figure 198. The first row shows Nu corresponding to the closest layer to the cylindrical wall, at a distance of 0.1 mm from the wall. As explained previously, Nu is higher at the entrance of each channel and decreases gradually as the flow develops. Depending on the mass flow rate the asymptotic value of Nu is in the range from 2.6 to 3.75.
In this section the results of the simulations in non-adiabatic conditions are presented. The convective heat transfer coefficient at the outer-most surface where heat is exchanged with the surroundings is set at 12 W/m²K and the influence of wall thermal conductivities are studied and compared to the adiabatic case. Emissivity value is kept constant at 0.2.

7.4.2.1 Effect of walls thermal conductivity

This section presents the results of the numerical simulations of the triple tube system when non-adiabatic conditions are applied to the external walls. As it can be seen in Figure 199, the temperature profiles in the inner cylinder are not very dissimilar from the adiabatic case. One notable difference however is that the region of maximum temperature is less extended than the adiabatic case due to the heat losses to the surroundings from the external annulus. As expected, the greatest heat loss occurs from the outer annulus to the surroundings with the mass averaged temperatures at the exhaust outlet being 237 K and 206 K lower than
the adiabatic case when wall thermal conductivities are 0.5 W/mK and 16 W/mK respectively, at a mass flow rate for 0.15 g/s. At the same flow rate the preheating temperatures at the inner annulus outlet are 97 K and 87 K lower than the adiabatic case at the two wall thermal conductivities respectively. Table 23 summarizes the mass averaged temperatures at the outlets of the outer and inner annuli. Also the contour maps of velocity for the triple tube system in the non-adiabatic case are presented in Figure 200.

![Contour Maps](image)

**Figure 199** – Contours of temperature at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – non-adiabatic case

The velocity contours in the non-adiabatic case allow to evaluate the Reynolds number in the three channel which is never higher than 130, therefore the flow is always laminar. As it can be seen in the inner annulus the fuel-air mixture reaches the highest velocity due also to heating, therefore decrease in density. The opposite situation happens in the outer channel
where there is a decrease of velocity caused by the cooling of the fluid. In the combustion chamber the velocity increases suddenly just downstream the inlet where the combustion occurs, as a result of the expansion of the mixture.

Table 23 - Preheating temperature of reactants and temperatures at the outer channel outlet at different mass flow rates and different conductivities in a triple tube system – non-adiabatic case

<table>
<thead>
<tr>
<th></th>
<th>Exhaust temperature at outlet</th>
<th>Reactants’ preheating temperatures</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.5 W/mK</td>
<td>16 W/mK</td>
</tr>
<tr>
<td>0.05 g/s</td>
<td>758 K</td>
<td>833 K</td>
</tr>
<tr>
<td>0.1 g/s</td>
<td>893 K</td>
<td>908 K</td>
</tr>
<tr>
<td>0.15 g/s</td>
<td>1023 K</td>
<td>1019 K</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>0.5 W/mK</th>
<th>16 W/mK</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05 g/s</td>
<td>1032 K</td>
<td>1020 K</td>
</tr>
<tr>
<td>0.1 g/s</td>
<td>1093 K</td>
<td>1080 K</td>
</tr>
<tr>
<td>0.15 g/s</td>
<td>1103 K</td>
<td>1110 K</td>
</tr>
</tbody>
</table>

Figure 200 - Contours of velocity at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – non-adiabatic case
In Figure 201 plots of Nusselt number are shown. These values are extrapolated from the CFD simulations the same way as for the adiabatic case at 0.1 mm from the walls surfaces.

The graphs in Figure 201 show profiles of Nusselt number similar to the adiabatic case in Figure 194. In the inner cylinder Nu reaches values of around 80 in the entrance region, due to the phenomenon previously mentioned, and decreases to values in the range between 2.35 and 3.51, depending on the total flow rate. Nusselt number in the inner annulus has a similar trend to the adiabatic case.

![Figure 201 - Plots of Nu at 0.5 W/mK (left) and 16 W/mK (right) for the triple tube non-adiabatic case](image-url)
Figure 202 - Contours of temperature at different flow rates, equivalence ratio of 1 and different wall thermal conductivities – non-adiabatic case
Double tube system

In Figure 202 and Figure 203 contour maps of temperature and velocity of the double tube system at different flow rates and different wall thermal conductivities are shown.

\[
\begin{align*}
\text{Velocity} & \quad [\text{m s}^{-1}] \\
0.05 \text{ g/s} & \quad 0.1 \text{ g/s} & \quad 0.15 \text{ g/s} \\
\end{align*}
\]

In this case the contours show the same maximum temperature achieved in the inner cylinder as the adiabatic case and a lower mass averaged temperature at the inner annulus outlet. It can be seen how the preheating temperature decreases as the flow rate increases, contrasting the results obtained from the triple tube system. This can be explained considering the increased flow velocity which in turns causes an increase in the heat transfer coefficient, therefore in the heat lost to the external environment.
<table>
<thead>
<tr>
<th>Mass Flow Rate</th>
<th>Preheating Temperature (K)</th>
<th>0.5 W/mK</th>
<th>16 W/mK</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05 g/s</td>
<td></td>
<td>904 K</td>
<td>823 K</td>
</tr>
<tr>
<td>0.1 g/s</td>
<td></td>
<td>869 K</td>
<td>845 K</td>
</tr>
<tr>
<td>0.15 g/s</td>
<td></td>
<td>825 K</td>
<td>814 K</td>
</tr>
</tbody>
</table>

Nusselt number profiles related to the double tube system in non-adiabatic conditions are shown in Figure 204.

**7.4.2.2 Further insights into the heat transfer processes**

To have a deeper understanding of the heat transfer processes occurring in the combustion chamber, heat fluxes were averaged over the heat exchanging surfaces in order to obtain the total heat transmitted between the flows in the channels. Figure 205 presents a comparison between the heat transferred in the adiabatic case and the non-adiabatic case respectively, within a triple tube system. Obviously when the double tube is studied, the heat contained in the combustion products is not recovered, thus it is lost to the environment. Also as a reminder the combustion products are recirculated into the outer annulus after decreasing.

![Figure 204 - Plots of Nu at 0.5 W/mK (left) and 16 W/mK (right) for the double tube non-adiabatic case](image-url)
their temperature of 200 K to simulate a hypothetical turbine downstream the combustion chamber. Keeping this in mind, the present paragraph is only meant to show the heat fluxes within the combustion chamber and the amount of heat exchanged between the flows and lost through the walls. Figure 205 and Figure 206 show the heat transferred and percentages of the heat release from the combustion between the flows in both double tube and triple tube systems.

As it can be seen in the adiabatic case, the amount of heat transferred to the cold reactants flowing in the inner annulus reaches 43% and 44.6% of the heat generated during the combustion of the mixture, when the total flow rates are 0.1 g/s and 0.15 g/s respectively. When the flow rate is set at 0.05 g/s the percentage of heat transferred is 41%, which remains considerable. Applying non-adiabatic conditions at the external wall and at the ends of the
inner walls, the percentages decrease to 34.6% at 0.05 g/s, 37% at 0.1 g/s and 38.3% at 0.15 g/s. In these cases the amount of heat lost to the surroundings from the ends of the inner and outer walls range from 4.7 W (at 0.05 g/s) to 7.5 W (at 0.15 g/s). The same analysis was performed for the double tube model and the results are shown in Figure 206.

![Figure 206 - Heat transfer within the doubletube combustor at different mass flow rates – adiabatic case (left), non-adiabatic case (right)](image)

Compared to the triple tube model, the double tube shows a lower amount of heat transferred to the cold reactants in both cases and, in addition, the percentage of heat transferred decreases considerably as the mass flow rate increases. In fact, in adiabatic conditions, at 0.05 g/s 35.5% of heat is recovered to preheat the cold mixture, at 0.1 g/s the percentage is 30.3% and at 0.15 g/s it is only 25%. When non-adiabatic conditions are imposed at the external walls, the percentages (including heat lost through the wall ends)
decrease to 22.5% at 0.05 g/s, 24% at 0.1 g/s and 22.9% at 0.15 g/s. The heat losses through
the inner wall ends are in the range from 2 W to 3.2 W.

From the results and analysis presented in this section, it can be concluded that the
maximum amount of heat that can be transferred to the cold mixture in a triple tube system
for the case of perfect insulation i.e. the adiabatic case is up to 43% of the heat released
during combustion. The percentage decreases when the system exchanges heat with the
surroundings, however the amount of heat transferred can reach around 38% with the same
heat release rate from combustion, which still is considerable. In the double tube model heat
transferred to the reactants is much lower and it can reach 35.5% in the adiabatic condition
but only 24% in non-adiabatic conditions. However, the double tube model shows lower heat
losses from the external walls as the difference of temperature between the flow in the inner
annulus and the surroundings is considerably lower. As previously reminded, the heat
contained in the combustion products is not re-used in the double tube system so the
efficiency is lower than the triple tube case, as expected.

7.4.2.3 Effect of thermal conductivity of outer wall

In this section the thermal conductivity of the outer wall is changed, maintaining the
thermal conductivity of the inner wall at 16 W/mK. Results are compared in both the triple
tube and the double tube systems.

Figure 207 shows the contours of temperature (left) and velocity (right) for the three
concentric tubes model with outer wall thermal conductivity of 0.5 W/mK. Values of
preheating temperature of reactants in this case are very similar to the non-adiabatic case with
outer wall thermal conductivity at 16 W/mK. The only difference that can be noticed is a
slightly lower temperature (by 34 K) at the inner annulus outlet when the outer wall has a
conductivity of 0.5 W/mK. Nusselt number plots were not shown as the profiles are nearly
identical to the non-adiabatic case shown in the previous section.
Also the double tube model (Figure 208) shows minimal discrepancies in terms of preheating temperatures of unburnt mixture compared to the non-adiabatic case, as the differences range between 34 K and 19 K. Therefore it can be concluded that the influence of the outer wall thermal conductivity is not highly relevant in terms of heat exchanges in the two models considered. Although using fixed thermal conductivities may not be considered fully realistic, simulations showed how the preheating temperature at a fixed flow rate in a triple tube system is only slightly affected by this parameter. In a double tube system thermal conductivity has a more important influence on preheating temperature. Typically the thermal conductivity of stainless steel increases with the temperature and it reaches values of around 30 W/mK at about 1500 K [134]; therefore one could expect to have an increase of preheating temperature in the same order of magnitude of the one showed between the two

Figure 207 - Contours of temperature and velocity at different flow rates, equivalence ratio of 1 and outer wall thermal conductivity of 0.5 W/mK – non-adiabatic case, triple tube system
cases at 0.5 and 16 W/mK. This can be approximately quantified in around 70 K for the double tube case and about 15 K for the triple tube system.

Figure 208 - Contours of temperature and velocity at different flow rates, equivalence ratio of 1 and outer wall thermal conductivity of 0.5 W/mK – non-adiabatic case, double tube system

### 7.4.2.4 Effect of porous insert as flame holder in the inner cylinder

The last situation considered in this study includes a porous medium acting as a flame holder in the combustion chamber. As previously mentioned flat flame combustion is an employable practice in small scale devices such as UMGTs and several researches directed their efforts towards its development. The porous medium is modelled as a 1 mm thick stainless steel insert with a porosity of 0.5 and its physical characteristics are extrapolated from experimental data [12]. The walls in these simulations are modelled as stainless steel (with thermal conductivity of 16 W/mK), non-adiabatic conditions are applied and the total mass flow rate was varied as in the previous sections.
Figure 209 shows the contours of temperature and velocity in the triple tube model and Table 25 summarizes the preheating temperatures and the outer annulus outlet temperatures.

The introduction of the porous insert causes an increase in the maximum temperature in the inner cylinder as it reaches 2145 K at 0.05 g/s, 2597 K at 0.1 g/s and 2675 K at 0.15 g/s. This is due to the higher preheating temperature of the cold mixture which in the three cases considered reaches 1343 K, 1390 K and 1304 K respectively.

As explained in a previous research [129] where a CFD study was carried out on a cylindrical combustion chamber including a porous insert as flame holder, under certain conditions super-adiabatic temperature can be reached due to heat recirculation through the porous medium and the inner wall. In this case very high maximum temperatures are reached as shown in Table 25.
Table 25 - Preheating temperature of reactants and temperatures at the outer channel outlet at different mass flow rates and in the triple tube system with porous insert

<table>
<thead>
<tr>
<th>Mass Flow Rate</th>
<th>Exhaust Temperature at Outlet</th>
<th>Reactants’ Preheating Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05 g/s</td>
<td>957 K</td>
<td>1343 K</td>
</tr>
<tr>
<td>0.1 g/s</td>
<td>1085 K</td>
<td>1390 K</td>
</tr>
<tr>
<td>0.15 g/s</td>
<td>1169 K</td>
<td>1304 K</td>
</tr>
</tbody>
</table>

Figure 210 shows the temperature and velocity contours of the double tube model.

Also in this situation the maximum temperatures are much higher than the previous cases, as it reaches 2254 K at 0.05 g/s, 2456 K at 0.1 g/s and 2481 K at 0.15 g/s. As the mass flow rate increases, the preheating temperature decreases from 1070 K at 0.05 to 947 K at 0.15 g/s. This is due to the higher velocity in the outer tube which on one side enhances heat
transfer from the combustion chamber to the same outer tube but on the other side also increases heat losses to the surroundings.

In Figure 211 plots of Nusselt number are shown. As it can be noticed downstream of the porous insert Nu increases considerably and then asymptotically decreases to reach a constant value towards the outlet. This behaviour, as mentioned above, also happens in the entrance regions near the inlets and locally causes a higher heat transfer. The flame holder in this case seems to act like a new entrance, therefore the flow goes through a second development and in turns Nusselt number increases, as well as the heat transferred to the inner annulus. Also the presence of zones at the same temperature in both reactants channel and inner tube (at the same axial coordinate) causes the sudden increase of the Nusselt number. Another contribution to the increase of temperature of the cold mixture is given by the heat recirculated through the combustion chamber wall upstream the flame holder. As demonstrated in another numerical study [129], this produces benefits in terms of heat transferred to the unburnt gases increasing considerably the preheating temperature.

![Figure 211 – Nu plots at different flow rates for the inner tubes of triple tube and double tube cases with porous medium](image)

### 7.5 Summary

This chapter presented a parametric Computational Fluid Dynamic study of two different models of regenerative combustion chamber for micro scale applications with the purpose of investigating heat exchanges and performances of the system. The main advantage of using a regenerative combustion chamber is to increase the temperature of reactants before combustion in order to obtain higher burning velocity and lower fuel consumption. Therefore this study helps in the design of such systems and confirms the validity of the CFD in the design process. The models defined as “double tube” and “triple tube” systems were
simulated at different mass flow rates (0.05 g/s, 0.1 g/s and 0.15 g/s), different wall thermal conductivities (0.5 W/mK and 16 W/mK) and included a porous medium in the combustion chamber. A detailed skeletal mechanism was employed to better investigate the heat released during combustion and, after validating the model, results were presented in terms of temperature contours, Nusselt number profiles and total heat exchanged between flows. In conclusion the triple tube model, as expected, showed better thermal performances than the double tube, as the fraction of heat of combustion transferred to the cold mixture was always higher. On the other side in the double tube system heat losses to the external environment were contained, as the temperature difference between the cold flow and the surroundings was lower, however, since the combustion products were not recirculated, a greater amount of heat was lost to the ambient. Therefore a four concentric tubes regenerative combustion chamber with a second recirculation of cold reactants could be even more efficient but the radial dimension would increase as well as cost. Results show that wall thermal conductivity has only a small effect on reactants’ preheating temperature especially with increasing mass flow rate. Generally when the thermal conductivity is set at 0.5 W/mK the temperature at the inner annulus outlet of the triple tube model is only around 10 K higher than the case at 16 W/mK. The only exception is shown at 0.05 g/s in adiabatic conditions where the difference of temperature is 101 K. When only the thermal conductivity of the outer wall was changed no significant changes are present and its impact on heat exchanges is very limited. In the last set of simulations including a porous insert in the combustion chamber, a great increase in preheating temperature occurs due to heat recirculation through axial conduction in the flame holder and in the inner wall.
8 CONCLUSIONS

In this work experimental and numerical studies were carried out to investigate the interactions between combustion and heat transfer in small scale combustion chambers.

As highlighted in Chapter 1, the main scope of the research was to develop a regenerative combustion chamber for Ultra Micro Gas Turbines applications, designed to generate a power output included in the range from 1 W to 50 W. To increase thermal efficiency, the combustion chamber was designed in order to recover heat from the combustion products and transfer it to the reactants. Therefore the experimental part includes the measurement of thermal performances, combustion products composition, axial distribution of temperature in the combustion chamber and visual observation of the flame shape and characteristics. Also tests were run exclusively using vegetable oil as fuel and, although this created issues during operation, the feasibility of this configuration was demonstrated to be possible.

This work also includes two numerical studies. The first one is related to investigate the interactions between combustion and heat transfer in a cylindrical combustion chamber with flame stabilized on a porous medium acting as flame holder. Parameters such as equivalence ratio, mixture mass flow rate and porosity of the flame holder are varied and results are presented in terms of temperature and velocity contours and concentration of major species at the outlet. The second numerical study is focused on studying combustion and heat transfer processes in a three concentric tubes combustion chamber. This model is the simplification of the combustion chamber under experimental investigation and the main purpose was carrying out a parametric study varying equivalence ratio, mixture mass flow rate and thermal conductivity of the walls. Also the effects of including a flame holder in the combustion chamber were investigated.

The findings and conclusions from this research are summarized below.

Experimental work

- The combustion products analysis showed that the designed combustion chamber operating with flat flame was able to produce very efficient combustion. The combustion efficiency resulted nearly equal to 1 for a wide range of equivalence ratios on the lean side of combustion. At stoichiometric conditions the efficiency was
no lower than 0.97 in any case considered, and at the lowest flow rate (0.04 g/s) it resulted 0.997.

- Emissions of NO\textsubscript{x} were also detected and they showed a peak of concentration near stoichiometric conditions (about 0.95). This is in accordance with the expectations as the maximum production of NO\textsubscript{x} occurs at elevated temperatures and excess of oxygen.

- Equivalence ratio was calculated from the exhaust gases composition and the combustion efficiency at the chosen design point (equivalence ratio equal to 0.7) resulted no lower than 0.99 at every flow rate, demonstrating the achievement of an extremely clean combustion.

- The visual observation of the flame showed the characteristics, the shape and the colour of the flame. At every operating condition considered utilizing LPG as a fuel, the flame was stable and seemed attached to the porous holder. At low flow rates and low equivalence ratio the flame showed a dark blue colour as a result of the low heat released during combustion and of heat losses through the porous medium. As the heat released from the combustion was increased, the flame showed a glowing red area in the central part of the combustion chamber, which became larger as the flow rate was increased. This was due to the heat transferred to the flame holder which in turn became incandescent as a result of the increased surface temperature. At rich combustion the flame showed shades of green due to high concentration of fuel, matching experimental results of previous researches.

- Measurements of temperature in the combustion chamber along the axial coordinate showed that the highest temperature was always close to the porous medium (the closest measurement point was at 1 mm from the bottom surface of the flame holder), confirming the fact that the flame was attached and stabilized on the bottom surface of the flame holder. Temperature was dependant on the mass flow rate and equivalence ratio, thus heat released during combustion.

- The preheating temperature of the reactant mixture ranged between 600 K and 900 K in the cases considered. The value of temperature showed a peak at stoichiometric
conditions and it was experimentally demonstrate to be dependent on the mass flow rate, therefore on the heat released during combustion.

- The temperature of the exhaust gases at the combustion chamber outlet ranged between 555 K and 840 K at stoichiometric conditions in the situations considered. Also in this case this value was dependent on the heat released from the combustion (i.e. from equivalence ratio and mass flow rate). The heat contained in the exhaust gases could be therefore employed for other purposes (e.g. further preheating of reactant mixture).

- From the temperature measurements, the fraction of heat transferred to the cold mixture was calculated. This value resulted highest at low equivalence ratio in every case considered and it increased with the mass flow rate. The maximum amount of heat recovered from the combustion products was about 29 % at 0.18 g/s flow rate and 0.52 equivalence ratio. The reason for this effect was attributed to the decrease of laminar burning velocity when decreasing equivalence ratio. This, if the mass flow rate is kept constant, causes the incoming flow to “push” the flame towards the outlet of the combustion chamber lifting from the bottom surface of the porous medium. Therefore a lower heat rate is lost to the porous medium and, through conduction within the walls, to the surroundings. The worst situation occurred at 0.04 g/s and equivalence ratio equal to 1.15, as the fraction of heat recovered from the combustion products was only 9.1 %.

- Thermal imaging was also presented in order to show the temperature of the external surface of the combustion chamber. Only two cases were presented because of the limitations of the thermal camera temperature range. The external wall temperature was uniformly distributed on the surface and it resulted 311 °C when the mass flow rate was set at 0.04 g/s and 388 °C at 0.06 g/s, both at stoichiometric conditions. The temperature was detected in the central area of the combustion chamber and deviation from these values was contained in the ±5 °C.

- Tests were also run on the combustion chamber when insulating material was applied externally, encasing the device thus limiting heat losses. Images of the flame showed how the bottom surface of the porous medium becomes incandescent already at 0.06
g/s. Increasing the flow rate causes the flame holder to emit a bright orange colour. Tests were stopped at 0.1 g/s to avoid damages as the combustion chamber walls were incandescent.

The axial distribution of temperature in the insulated combustion chamber showed that the maximum temperature (which was close to the porous medium) was slightly lower than that in the non-insulated case at the same flow rate. As the distance from the flame holder increased the axial temperature in the insulated combustion chamber showed less decrease compared to the non-insulated combustor because of the contained heat losses.

The preheating temperature in the insulated case resulted higher than the non-insulated case and the highest temperature at every flow rate was recorded at stoichiometric conditions, as in the previous case. Comparing the reactants preheating temperatures at 0.1 g/s flow rate and stoichiometric conditions, it reached 792 K in the non-insulated configuration while with insulated combustor it resulted 1035 K.

The fraction of heat transferred to the cold reactants in the insulated combustion chamber reached 46 % of the heat released from the combustion at 0.1 g/s and 0.68 equivalence ratio against 22 % in the non-insulated combustion chamber. The decrease of enthalpy in a hypothetical turbine was not considered. As expected utilizing insulating materials externally and on the two ends of the combustion chamber has a positive effect in increasing the preheating temperature however the temperature at the combustion chamber outlet results also higher. This heat could be used for other purposes downstream of the combustion chamber outlet.

The combustion chamber was also tested using pure vegetable oil as fuel, through a flow blurring injector installed on the bottom flange. Operation only on vegetable oil was achieved and demonstrated to be feasible however clogging issues were encountered. After about thirty minutes of operation with vegetable oil, the combustion chamber had to be cleaned to remove deposits of coke and solid combustion products that caused the combustion chamber to be clogged. Condensation of vaporized oil on the top ceramic flange was hypothesized, with
consequent deposit of partial combustion products that blocked the pores of the flame holder.

- Emissions of CO₂, CO and NOₓ were measured at two different flow rates. A comprehensive set of measurements was not obtained because of clogging of the combustor chamber channels. The profiles of CO₂ as expected showed a peak at stoichiometric conditions and at the same point the O₂ concentration decreased to nearly zero. Concentration of CO resulted very low on the lean side of combustion (not higher than 0.1 % at equivalence ratio of 0.59) and increased slightly at stoichiometric conditions (0.5 %). In the present study the experiments were run at atmospheric pressure as the main objective was investigating how heat transfer and combustion interact together. In an actual configuration where the combustion chamber is coupled with a compressor/turbine group, the operating pressure will be above atmospheric pressure, ideally in the order from 2 to 4 bar. The pressure increase is expected to affect the combustion characteristics and the emissions concentration. NOₓ concentration, as documented in other researches [135,136], results higher as the operating pressure increases. This effect is expected to be more evident approaching the stoichiometric conditions, rather than on the lean side of combustion. Another research on flat-flame burners [137] showed that NOₓ emissions decrease with increasing pressure. On the other side CO and unburned hydrocarbons emissions resulted higher at increasing pressure.

- Flame images were acquired at mass flow rate 0.1 g/s at different equivalence ratio. The flame showed a blue colour at low equivalence ratio and, approaching the stoichiometric conditions, the heat released caused the temperature of the porous insert to increase, showing a red glowing area.

- Combustion efficiency at stoichiometric conditions reached 0.987 at 0.1 g/s and 0.99 at 0.12 g/s, confirming the achievement of clean and efficient combustion using vegetable oil.

- NOₓ emissions showed a peak at stoichiometric conditions reaching 88 ppm and 76 ppm at 0.12 g/s and 0.1 g/s respectively.
• Axial distributions of temperature in the combustion chamber showed similar profiles of those of LPG. The maximum temperature detected was located at 1 mm from the bottom surface of the porous insert and progressively decreased with the distance from it. Maximum measured temperatures were 1858 K at 0.1 g/s and 1926 K at 0.12 g/s.

Numerical work on cylindrical combustion chamber

• A numerical model of an 18 mm cylindrical combustion chamber with porous insert was developed. A 17 species, 73 reaction mechanism describing methane oxidation was included in the simulation with the aim of investigating interactions between combustion and heat transfers in the combustion chamber. Adiabatic conditions were simulated and compared with experimental results available in literature, providing a good validation in terms of concentration of major species in the combustion products and flame temperature.

• A parametric study simulating non-adiabatic conditions was carried out varying parameters as equivalence ratio, mass flow rate, initial mixture temperature, thermal conductivity of the flame holder, porosity of the flame holder and external temperature.

• The study showed that the location of the flame determines the mode of heat transfer to the surrounding walls. When the flame resulted attached to the porous medium heat was transferred to the walls through it. On the other side when the flame was lifted from the porous insert heat was transferred to the outer wall and recirculated upstream the flame, causing a preheating effect.

• Simulations showed how decreasing the flame holder thermal conductivity benefits the maximum temperature reached in the combustion products as it limits heat losses though the porous material. This resulted also in widened flammability limits.

• Similarly increasing the porosity of the flame holder, therefore the void fraction in its volume, provides an insulating effect thus a higher maximum temperature in the combustion chamber.
• Setting the external temperature at 800 K has also positive effects on limiting heat losses to the surroundings resulting in widened flammability limits.

Numerical work on triple tube combustor/heat exchanger

• Numerical simulations of a three concentric tube combustion chamber are carried out to investigate heat exchange processes and their interaction with combustion phenomena. Combustion of methane was simulated using a 17 species, 73 reactions mechanism and the model validation was obtained comparing a reference case with a developed analytical formulation. Profiles of temperatures showed good match between the two approaches. A parametric study was carried out varying equivalence ratio, mass flow rate, thermal conductivity of the walls, thermal conductivity of the outer wall only and including a 1 mm thick flame holder in the combustion chamber. Also a two concentric tube model was set up and simulated. Results were presented in terms of contours of temperature and velocity. Also Nusselt number profiles in the combustion chamber and in the two annuli were plotted.

• The triple tube model showed increasing preheating temperature with increasing flow rate and decreasing thermal conductivity of the walls in both adiabatic and non-adiabatic cases.

• Nusselt number plots showed how at the inlet of every tube the values of Nu were highest due to the entrance effect (Graetz effect). The curve rapidly decreased asymptotically along the axial coordinate to a constant value depending on local conditions. This value in every case was lower than 4.36, which corresponds to Nu when a constant heat flux is applied to the considered surface. This was explained with the presence of axial conduction through the wall which contributed to lowering the local Nu value.

• Plots of Nusselt number in the outer annulus also showed the occurrence of peaks at axial location where crossover of temperature happened. Profiles of temperature in the inner annulus and in the outer annulus confirmed this behaviour.

• Wall thermal conductivity was shown to have little effect on preheating temperature of reactants, especially at increasing mass flow rates. Also the impact of varying the
thermal conductivity of the outer wall only has minor effects on the preheating temperature of the reactant mixture.

- Including a flame holder in the combustion chamber causes a great increase in the preheating temperature due to heat recirculation through axial conduction within the walls of the combustion chamber. Nusselt number profiles showed a second peak downstream of the porous insert as if another entrance effect was present. As the flow develops downstream, Nu decreases asymptotically to a constant value.
Suggestion for future studies

- Although good performances in terms of heat recovery were obtained, the design of the regenerative combustion chamber could be improved to include an additional outer annulus where cold mixture can be inserted and preheated. This would further minimize heat losses to the surroundings.

- In this study the combustion chamber was tested at ambient pressure as the main focus was studying thermal exchanges within it. Investigating the interactions between combustion properties and heat transfer at design pressure (usually around 2.5-3 bar) is recommended.

- Numerical models can be developed to estimate heat losses and heat exchanges within the regenerative combustion chamber. This would also allow the optimization of the geometry of the channels and the fluid dynamic within them.

- Further improvements are required in order to use vegetable oils as fuel in small scale systems in a premixed combustion configuration. In particular oil vaporization can be improved and clogging issues need to be further investigated and solved. This could be achieved optimizing the geometry of channel to further enhance vaporization or adopting different combustion techniques such as the liquid film.

- Such a design could be adapted to suit applications, other than UMGTs. This may include for example coupling with thermophovoltaic cells to produce electric power.

- A complete small scale UMGT including a compressor/turbine group and a regenerative combustion chamber has to be developed and tested to test the performances.

- Numerical investigations on fluid dynamics and liquid breakup phenomena in flow blurring injectors need to be carried out to improve the design of the device, therefore atomization properties.
Appendices

A1. Flow controllers working principle

In the experimental work different mass flow controllers were used: the Sierra Smart Trak 2 Series 100 (C100L) [138] and a Vögtlin Instruments AG Red-y Smart Series controller (GSC up to 6000ml/min of air) [139]. Their working principle is based on the measurement of the heat transport by the gas flowing through the flowmeter. In Figure 212 the working principle of the Sierra flowmeter is shown as an example. The gas line is split into two paths, one connected to the sensor tube, the other one to a laminar flow bypass which, by generating a pressure drop across itself, forces a part of the flow to pass through the sensor tube. Two resistance temperature detector (RTD) coils are wrapped around the sensor tube and, through them, a constant amount of heat is directed into the gas stream. When gas flows through the sensor tube, it carries heat from the upstream coil to the downstream coil and the temperature difference is measured by the microprocessor. The output signal, which is linear with the flow rate, is then calculated. Figure 213 shows the block diagram of a red-y flow controller which incorporates a solenoid valve for setting the flow rate.

Figure 212 - Flow path through Sierra Smart Trak 2 (left) and measuring principle (right)

Figure 213 - Block diagram of red-y flow controllers
A2. Flow rate correction factor

As mentioned, the Sierra flow controller was used for the air flow and the red-y controllers were used for the gas flow. The first flow meter is programmed with different calibrations in order to adjust the output signal for 10 different gases (Air, Ar, CO₂, CO, He, H₂, CH₄, N₂, N₂O, O₂). The second one is calibrated only for operation with air so the mass flow rate needed to be adjusted for operation with the fuel gas mixture. A conversion method is reported in another Sierra flow meter manual [62].

Considering the thermal balance of the sensor tube, the relationship between heat transferred to the tube and the mass flow rate is:

$$H = \frac{\dot{m}c_p\Delta T}{N}$$  \hspace{1cm} (86)

In equation (86) H represent the constant amount of heat supplied by the coils, \(\dot{m}\) is the mass flow rate, \(c_p\) is the specific heat at 0 °C, \(\Delta T\) is the temperature difference between the downstream and the upstream coils and \(N\) is a coefficient that depends on the molecular structure of the gas and it is listed in Table 26.

<table>
<thead>
<tr>
<th>Number of Atoms in the Gas Molecule</th>
<th>(N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Monatomic</td>
<td>1.040</td>
</tr>
<tr>
<td>Diatomic</td>
<td>1.000</td>
</tr>
<tr>
<td>Triatomic</td>
<td>0.941</td>
</tr>
<tr>
<td>Polyatomic</td>
<td>0.880</td>
</tr>
</tbody>
</table>

The mass flow rate can be written as:

$$\dot{m} = \rho \dot{\gamma}$$  \hspace{1cm} (87)

where the gas density is at 0 °C and 760 mm Hg.

The temperature difference can be expressed as linear with the output voltage \(E\) of the flow meter, according to a constant \(a\):

$$\Delta T = aE$$  \hspace{1cm} (88)

Therefore equation (86) becomes:

$$\dot{\gamma} = \frac{bN}{\rho c_p}$$  \hspace{1cm} (89)

with \(b = \frac{H}{aE}\).
The correction factor between the flow rates is the ratio of the actual gas flow rate and the reference gas flow rate:

\[
\frac{\dot{V}_1}{\dot{V}_2} = \frac{\frac{N_1}{\rho_1 c_p_1}}{\frac{N_2}{\rho_2 c_p_2}} = \frac{K_1}{K_2}
\]

When operating with a multi component gas (with \(i=1,\ldots,N\) components), density, specific heat and \(N\) need to be adjusted using the weighted average:

\[
\rho = \sum_{i=1}^{N} \frac{\dot{m}_i}{\dot{m}_{TOT}} \rho_i
\]  

(91)

\[
c_p = \sum_{i=1}^{N} \frac{\dot{m}_i \rho_i}{\dot{m}_{TOT}} c_{pi}
\]  

(92)

\[
N = \sum_{i=1}^{N} \frac{\dot{m}_i}{\dot{m}_{TOT}} N_i
\]  

(93)

The validity of this procedure has been confirmed analysing the exhaust gases and calculating the equivalence ratio from their composition. The procedure to extrapolate the equivalence ratio from the emissions involves some chemistry of combustion and has been explained previously in this thesis. The Sierra SmartTrak Manual advises that the inaccuracy when utilizing K-factor corrections can range from ±5 % to 10 % of the accuracy. Comparisons of equivalence ratio calculated from emissions and equivalence ratio calculated using the K-factor are performed at several flow rates and the results are shown in Figure 214. Also a linear trendline is shown, described by its equation and the R-squared value.
Figure 214 – Comparisons between equivalence ratios calculated from emissions and equivalence ratios from the flow meter

- For a fuel flow rate of 0.04 g/s:
  \[ y = 1.0069x \quad R^2 = 0.9885 \]

- For a fuel flow rate of 0.06 g/s:
  \[ y = 1.0026x \quad R^2 = 0.9933 \]

- For a fuel flow rate of 0.08 g/s:
  \[ y = 0.9946x \quad R^2 = 0.9972 \]

- For a fuel flow rate of 0.1 g/s:
  \[ y = 1.0155x \quad R^2 = 0.9945 \]

- For a fuel flow rate of 0.12 g/s:
  \[ y = 0.9965x \quad R^2 = 0.9988 \]

- For a fuel flow rate of 0.15 g/s:
  \[ y = 1.0013x \quad R^2 = 0.9921 \]
A3. Thermocouple working principle

The working principle of a thermocouple is based on the Seebeck effect. Any two different materials can be used as a thermocouple if connected together in a junction, as in Figure 215. When the junction temperature is different from the reference temperature, a low level DC voltage will be generated at the terminals. The value of the voltage depends on the materials A and B, on the reference temperature and on the junction temperature. The governing equation is:

$$ E = \int_{0}^{L} \varepsilon_A \frac{dT}{dx} dx + \int_{L}^{0} \varepsilon_B \frac{dT}{dx} dx $$

(94)

It can be seen that the electromotive force (emf) is generated by the wires, not by the junction which is just an electrical connection between the wires.

Connecting a voltmeter to the terminals allows the reading of the voltage but also introduces two more junctions between different materials, which in turns introduce voltages in opposition to E. The solution to this problem is setting the new junction at a known reference temperature (e.g. an ice bath) so that it has no effects on the emf reading.
A4. Thermocouple correction

The temperature measured by the B type thermocouple in the combustion chamber is directly exposed to the flame therefore radiation losses from the bead have to be taken into account. Taking Figure 217 as reference a heat balance including radiation losses to the surroundings and axial conduction can be formulated [60].

\[ T_g = T_b - \frac{\varepsilon_b \sigma (T_b^4 - T_w^4) + \left(\frac{k_b A_{cross}}{L A_{surf}}\right) \Delta T}{h} \]  \hfill (95)

In equation (95) \( T_g \) is the gas temperature, \( T_b \) is the measured temperature at the bead, \( \varepsilon_b \) is the bead emissivity calculated using equation (96) for B type thermocouples [141], \( \sigma \) is the Stefan-Boltzmann constant \( (5.67 \times 10^{-8} \, \text{W m}^{-2} \text{K}^{-4}) \), \( T_w \) is the temperature of the surroundings, \( k_b \) is the thermal conductivity of the bead calculated using equation (97) for B type thermocouples [141], \( A_{cross} \) is the cross sectional area of the wire, \( A_{surf} \) is the surface of the bead, \( L \) is the length of the wire, \( \Delta T \) is the difference of temperature across the wire and \( h \) is the convection coefficient on the surface of the bead calculated using equations (98) and (99).

\[ \varepsilon_{Pt} = 0.136 \ln T - 0.8047 \]  \hfill (96)

\[ k_{Pt} = 0.0198 T + 64.141 \]  \hfill (97)

\[ Nu = 2 + 0.6 Re^{1/2} Pr^{1/3} \]  \hfill (98)

\[ Nu = \frac{h D}{k} \]  \hfill (99)
A5. Thermocouples uncertainty analysis

An uncertainty analysis has been performed in order to apply a correction to the B type and K type thermocouple measurements. The total uncertainty was calculated by using the following quadrature sum formula:

\[
U_{tot} = k \sqrt{\sum U_i^2}
\]

The single uncertainties \(U_i^2\) are classified in type A and type B. Type A uncertainties can be evaluated by statistical analysis of data as standard deviation of a series of measurements; type B uncertainties are dependent on additional information external to data, for example hysteresis, inhomogeneity or thermal perturbations in the thermocouple. \(k\) is the coverage factor, taken as 1.96. In this case few series of measurements were taken starting at a setpoint of 400 °C and increasing the temperature until 600 °C with steps of 50 °C. Also two more sets of measurements were taken at 500 and 400 °C, when decreasing the temperature from 600 °C. A Jupiter Dry Block 650 including a reference probe with ±0.15 °C uncertainty was used for the scope and eight measurements of temperature were taken for every setpoint. In Table 27 only the average values are shown.

### Table 27 - Average values of temperature measurements

<table>
<thead>
<tr>
<th>Setpoint °C</th>
<th>Ref. probe</th>
<th>B type</th>
<th>Standard dev. B type</th>
<th>B type uncertainty</th>
<th>K type</th>
<th>Standard dev. K type</th>
<th>K type uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>400</td>
<td>400.6823</td>
<td>409.7500</td>
<td>4.533875</td>
<td>9.414958</td>
<td>396.9425</td>
<td>1.869875</td>
<td>4.428968</td>
</tr>
<tr>
<td>450</td>
<td>449.8491</td>
<td>454.9375</td>
<td>2.544225</td>
<td>6.100501</td>
<td>445.9315</td>
<td>1.958775</td>
<td>4.861768</td>
</tr>
<tr>
<td>500</td>
<td>500.8507</td>
<td>504.3125</td>
<td>1.730906</td>
<td>5.264309</td>
<td>497.3484</td>
<td>1.751156</td>
<td>4.951331</td>
</tr>
<tr>
<td>550</td>
<td>550.4072</td>
<td>549.8750</td>
<td>0.296868</td>
<td>4.596722</td>
<td>546.8550</td>
<td>1.776081</td>
<td>5.450926</td>
</tr>
<tr>
<td>600</td>
<td>600.5435</td>
<td>596.2500</td>
<td>2.146762</td>
<td>6.660761</td>
<td>596.8915</td>
<td>1.826037</td>
<td>6.058280</td>
</tr>
<tr>
<td>600</td>
<td>500.6602</td>
<td>503.5625</td>
<td>1.451168</td>
<td>4.922052</td>
<td>497.2953</td>
<td>1.682450</td>
<td>4.858480</td>
</tr>
</tbody>
</table>

Inhomogeneity uncertainty was calculated with equation (101):

\[
U_{inh}(T) = 0.15 + 0.0003T + 0.000006T^2
\]

Hysteresis was evaluated through the difference between measurements of temperatures of an ice bath and the dry block at 600 °C. Hysteresis uncertainty was 0.938 °C for the B type
thermocouple and 0.367 °C for the K type thermocouple. Corrected values of temperature were reported in the experimental results.
Uncertainty due to wire and connector was quantified in 0.5 % of the reading.
**A6. 17 species, 73 reactions skeletal mechanism**

Elements: O, H, C, N, Ar  
Species: H, O, O₂, OH, H₂O, HO₂, CH₂, CH₂(S), CH₃, CH₄, CO, CO₂, HCO, CH₂O, H₂OH, N₂  
Units: moles, seconds, Kelvins, calories/mole – Reference [98]

\[ k_f = A T^\beta e^{\frac{E_0}{RT}} \]

<table>
<thead>
<tr>
<th>#</th>
<th>Reaction</th>
<th>( A )</th>
<th>( \beta )</th>
<th>( E_0 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>R1</td>
<td>2O+M⇄O₂+M</td>
<td>1.200E+17</td>
<td>-1.000</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>H₂/ 2.40/ H₂O/15.40/ CH₄/ 2.00/ CO/ 1.75/ CO₂/ 3.60/</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R2</td>
<td>O+H+M⇄OH+M</td>
<td>5.000E+17</td>
<td>-1.000</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>H₂/2.00/ H₂O/6.00/ CH₄/2.00/ CO/1.50/ CO₂/2.00/</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R3</td>
<td>O+H₂⇄H+OH</td>
<td>5.000E+04</td>
<td>2.670</td>
<td>6290.00</td>
</tr>
<tr>
<td>R4</td>
<td>O+HO₂⇄OH+O₂</td>
<td>2.000E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R5</td>
<td>O+CH₂⇄H+HCO</td>
<td>8.000E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R6</td>
<td>O+CH₂(S)⇄H₂+CO</td>
<td>1.500E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R7</td>
<td>O+CH₃⇄H+CH₂O</td>
<td>8.430E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R8</td>
<td>O+CH₄⇄OH+CH₃</td>
<td>1.020E+09</td>
<td>1.500</td>
<td>8600.00</td>
</tr>
<tr>
<td>R9</td>
<td>O+CO+M⇄CO₂+M</td>
<td>6.020E+14</td>
<td>0.000</td>
<td>3000.00</td>
</tr>
<tr>
<td></td>
<td>H₂/2.00/ O₂/6.00/ H₂O/6.00/ CH₄/2.00/ CO/1.50/ CO₂/3.50/</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R10</td>
<td>O+HCO⇄OH+CO</td>
<td>3.000E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R11</td>
<td>O+HCO⇄H+CO₂</td>
<td>3.000E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R12</td>
<td>O+CH₂O⇄OH+HCO</td>
<td>3.900E+13</td>
<td>0.000</td>
<td>3540.00</td>
</tr>
<tr>
<td>R13</td>
<td>O+CH₂OH⇄OH+CH₂O</td>
<td>1.000E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R14</td>
<td>O₂+CO⇄O+CO₂</td>
<td>2.500E+12</td>
<td>0.000</td>
<td>47800.00</td>
</tr>
<tr>
<td>R15</td>
<td>O₂+CH₂O⇄H₂O+HCO</td>
<td>1.000E+14</td>
<td>0.000</td>
<td>40000.00</td>
</tr>
<tr>
<td>R16</td>
<td>H+O₂+M⇄HO₂+M</td>
<td>2.800E+18</td>
<td>-0.860</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>O₂/0.00/ H₂O/0.00/ CO/0.75/ CO₂/1.50/ N₂/0.00/</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R17</td>
<td>H+O₂⇄HO₂+O₂</td>
<td>3.000E+20</td>
<td>-1.720</td>
<td>0.00</td>
</tr>
<tr>
<td>R18</td>
<td>H+O₂+H₂O⇄HO₂+H₂O</td>
<td>9.380E+18</td>
<td>-0.760</td>
<td>0.00</td>
</tr>
<tr>
<td>R19</td>
<td>H+O₂+N₂⇄HO₂+N₂</td>
<td>3.750E+20</td>
<td>-1.720</td>
<td>0.00</td>
</tr>
<tr>
<td>R20</td>
<td>H+O₂⇄O+OH</td>
<td>8.300E+13</td>
<td>0.000</td>
<td>14413.00</td>
</tr>
<tr>
<td>R21</td>
<td>2H+M⇄H₂+M</td>
<td>1.000E+18</td>
<td>-1.000</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>H₂/0.00/ H₂O/0.00/ CH₄/2.00/ CO₂/0.00/</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R22</td>
<td>2H+H₂⇄2H₂</td>
<td>9.000E+16</td>
<td>-0.600</td>
<td>0.00</td>
</tr>
<tr>
<td>R23</td>
<td>2H+H₂O⇄H₂+H₂O</td>
<td>6.000E+19</td>
<td>-1.250</td>
<td>0.00</td>
</tr>
<tr>
<td>R24</td>
<td>2H+CO₂⇄H₂+CO₂</td>
<td>5.500E+20</td>
<td>-2.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R25</td>
<td>H+OH+M⇄H₂O+M</td>
<td>2.200E+22</td>
<td>-2.000</td>
<td>0.00</td>
</tr>
<tr>
<td></td>
<td>H₂/0.73/ H₂O/3.65/ CH₄/2.00/</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R26</td>
<td>H+HO₂⇄O+H₂O</td>
<td>3.970E+12</td>
<td>0.000</td>
<td>671.00</td>
</tr>
</tbody>
</table>
R27  H+HO2<=>O2+H2  2.800E+13  0.000  1068.00
R28  H+HO2<=>2OH   1.340E+14  0.000  635.00
R29  H+CH2(+M)<=>CH3(+M)  2.500E+16  -0.800  0.00
LOW / 3.200E+27  -3.140  1230.00 /
TROE/ 0.6800  78.00  1995.00  5590.00 / H2/2.00/ H2O/6.00/ CH4/2.00/ CO/1.50/ CO2/2.00/

R30  H+CH3(+M)<=>CH4(+M)  1.270E+16  -0.630  383.00
LOW / 2.477E+33  -4.760  2440.00 /
TROE/ 0.7830  74.00  2941.00  6964.00 / H2/2.00/ H2O/6.00/ CH4/2.00/ CO/1.50/ CO2/2.00/

R31  H+CH4<=>CH3+H2  6.600E+08  1.620  10840.00
R32  H+HCO(+M)<=>CH2O(+M)  1.090E+12  0.480  -260.00
LOW / 1.350E+24  -2.570  1425.00 /
TROE/ 0.7824  271.00  2755.00  6570.00 / H2/2.00/ H2O/6.00/ CH4/2.00/ CO/1.50/ CO2/2.00/

R33  H+HCO<=>H2+CO  7.340E+13  0.000  0.00
R34  H+CH2O(+M)<=>CH2OH(+M)  5.400E+11  0.454  3600.00
LOW / 1.270E+32  -4.820  6530.00 /
TROE/ 0.7187  103.00  1291.00  4160.00 / H2/2.00/ H2O/6.00/ CH4/2.00/ CO/1.50/ CO2/2.00/

R35  H+CH2O<=>HCO+H2  2.300E+10  1.050  3275.00
R36  H+CH2OH<=>H2+CH2O  2.000E+13  0.000  0.00
R37  H+CH2OH<=>OH+CH3  1.200E+13  0.000  0.00
R38  H+CH2OH<=>CH2(S)+H2O  6.000E+12  0.000  0.00
R39  H2+CO(+M)<=>CH2O(+M)  4.300E+07  1.500  79600.00
LOW / 5.070E+27  -3.420  84350.00 /
TROE/ 0.9320  197.00  1540.00  10300.00 / H2/2.00/ H2O/6.00/ CH4/2.00/ CO/1.50/ CO2/2.00/

R40  OH+H2<=>H+H2O  2.160E+08  1.510  3430.00
R41  2OH<=>O+H2O  3.570E+04  2.400  -2110.00
R42  OH+HO2<=>O2+H2O  2.900E+13  0.000  -500.00
R43  OH+CH2<=>H+CH2O  2.000E+13  0.000  0.00
R44  OH+CH2(S)<=>H+CH2O  3.000E+13  0.000  0.00
R45  OH+CH3<=>CH2+H2O  5.600E+07  1.600  5420.00
R46  OH+CH3<=>CH2(S)+H2O  2.501E+13  0.000  0.00
R47  OH+CH4<=>CH3+H2O  1.000E+08  1.600  3120.00
R48  OH+CO<=>H+CO2  4.760E+07  1.228  70.00
R49  OH+HCO<=>H2O+CO  5.000E+13  0.000  0.00
R50  OH+CH2O<=>HCO+H2O  3.430E+09  1.180  -447.00
<table>
<thead>
<tr>
<th>Reaction</th>
<th>Equation</th>
<th>K1</th>
<th>K2</th>
<th>K3</th>
</tr>
</thead>
<tbody>
<tr>
<td>R51</td>
<td>OH+CH2OH &lt;=&gt; H2O+CH2O</td>
<td>5.000E+12</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R52</td>
<td>HO2+CH2 &lt;=&gt; OH+CH2O</td>
<td>2.000E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R53</td>
<td>HO2+CH3 &lt;=&gt; O2+CH4</td>
<td>1.000E+12</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R54</td>
<td>HO2+CO &lt;=&gt; OH+CO2</td>
<td>1.500E+14</td>
<td>0.000</td>
<td>23600.00</td>
</tr>
<tr>
<td>R55</td>
<td>CH2+O2 &lt;=&gt; OH+HCO</td>
<td>1.320E+13</td>
<td>0.000</td>
<td>1500.00</td>
</tr>
<tr>
<td>R56</td>
<td>CH2+H2 &lt;=&gt; H+CH3</td>
<td>5.000E+05</td>
<td>2.000</td>
<td>7230.00</td>
</tr>
<tr>
<td>R57</td>
<td>CH2+CH4 &lt;=&gt; 2CH3</td>
<td>2.460E+06</td>
<td>2.000</td>
<td>8270.00</td>
</tr>
<tr>
<td>R58</td>
<td>CH2(S)+N2 &lt;=&gt; CH2+N2</td>
<td>1.500E+13</td>
<td>0.000</td>
<td>600.00</td>
</tr>
<tr>
<td>R59</td>
<td>CH2(S)+O2 &lt;=&gt; H+OH+CO</td>
<td>2.800E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R60</td>
<td>CH2(S)+O2 &lt;=&gt; CO+H2O</td>
<td>1.200E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R61</td>
<td>CH2(S)+H2 &lt;=&gt; CH3+H</td>
<td>7.000E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R62</td>
<td>CH2(S)+H2O &lt;=&gt; CH2+H2O</td>
<td>3.000E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R63</td>
<td>CH2(S)+CH4 &lt;=&gt; 2CH3</td>
<td>1.600E+13</td>
<td>0.000</td>
<td>-570.00</td>
</tr>
<tr>
<td>R64</td>
<td>CH2(S)+CO &lt;=&gt; CH2+CO</td>
<td>9.000E+12</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R65</td>
<td>CH2(S)+CO2 &lt;=&gt; CH2+CO2</td>
<td>7.000E+12</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R66</td>
<td>CH2(S)+CO2 &lt;=&gt; CO+CH2O</td>
<td>1.400E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R67</td>
<td>CH3+O2 &lt;=&gt; OH+CH2O</td>
<td>3.600E+10</td>
<td>0.000</td>
<td>8940.00</td>
</tr>
<tr>
<td>R68</td>
<td>CH3+HCO &lt;=&gt; CH4+CO</td>
<td>2.648E+13</td>
<td>0.000</td>
<td>0.00</td>
</tr>
<tr>
<td>R69</td>
<td>CH3+CH2O &lt;=&gt; HCO+CH4</td>
<td>3.320E+03</td>
<td>2.810</td>
<td>5860.00</td>
</tr>
<tr>
<td>R70</td>
<td>HCO+H2O &lt;=&gt; H+CO+H2O</td>
<td>-1.000</td>
<td>17000.00</td>
<td></td>
</tr>
<tr>
<td>R71</td>
<td>HCO+M &lt;=&gt; H+CO+M</td>
<td>1.870E+17</td>
<td>-1.000</td>
<td>17000.00</td>
</tr>
<tr>
<td>R72</td>
<td>HCO+O2 &lt;=&gt; HO2+CO</td>
<td>7.600E+12</td>
<td>0.000</td>
<td>400.00</td>
</tr>
<tr>
<td>R73</td>
<td>CH2OH+O2 &lt;=&gt; HO2+CH2O</td>
<td>1.800E+13</td>
<td>0.000</td>
<td>900.00</td>
</tr>
</tbody>
</table>
A7. Technical drawings
SECTION 1-1

SECTION 2-2
Assembly injector - Views

251

Assembly injector - Sections

Mechanical Engineering
References


[52] Asimptote, Fluidprop thermodynamic property library/.


[58] National Institute of Standards and Technology (NIST), NIST B type thermocouple table.,


[99] New Zealand eScience Infrastructure, Pan Cluster.


[128] Ansys, ANSYS® Fluent Academic Research, Release 17.0.,


[140] Omega Engineering, Omega Thermocouple Theory.