Reducing Cold Start Fuel Consumption Through Improved Thermal Management

Faisal Samad Lodi

A Thesis submitted for the total fulfillment of the degree of Master of Engineering Science

Department of Mechanical and Manufacturing Engineering
The University of Melbourne

April 2008

Produced on archival quality paper
Abstract

The thesis presents research in achieving faster warm-up of an SI engine, thereby affecting the fuel economy penalty. The faster warm-up relates to faster heating of the cylinder head and engine block, targeting reducing viscous friction in the cold oil as the most likely candidate to improve. The strategy applied was to reduce the coolant flow circulation rate to achieve a faster warm-up of the engine. A lumped parameter model for engine heat transfer, coolant flow and heat capacities, in a single cylinder, based on engine operating points like spark advance, engine speed and MAP was built in Modelica.

The engine used for experimentation was a Ford in-line, 4 stroke, 6-cylinder engine, with a compression ratio of 10.3:1, in which 56 K-type thermocouples were installed at different locations to measure the temperature. The experiments were performed with varying coolant flow rate from normal down to zero, utilizing an electric water pump, over an approximation to the New European Drive Cycle (NEDC), at a speed of 1161 rev/min and load of 48 Nm. The selected speed and load were the average operating condition for 180 seconds of engine running over the urban part of a simulated NEDC. In addition, the coolant circuit was
modified to a split cooling supply and the sets of results analyzed to find the reduction in engine warm-up time and fuel consumption. It is shown from the results that the warm-up time of the engine and the fuel consumption were notably reduced, as the flow was reduced from maximum to minimum in steps. On average over an interval of engine running for 300 seconds from cold start, the cylinder head temperature was increased by about $2^{\circ}C$, the average engine block temperature was increased by about $6.5^{\circ}C$ and the average cylinder head coolant temperature was increased by about $4^{\circ}C$. However, the bulk temperature of the oil in the oil sump showed marginal improvement and remained consistent, even at the lowest coolant flow rate. Nonetheless, the improvements in block temperature had significant effects on reducing the friction between the piston and cylinder walls.

Analysis of the results show that the coolant flow pattern changed with the use of an electric water pump. The flow is less evenly distributed around the cylinders with the use of an electric water pump, whilst retaining the mechanical water pump body, compared to the mechanical water pump operation.

The model was applied to simulate for two engine operating points, i.e., 1161 rev/min, 48 Nm load and 700 rev/min and 0 Nm load. The model was calibrated at 1161 rev/min, 48 Nm load and validated at 700 rev/min, 0 Nm load. The modeling results were in fair agreement with the experimental results. The model can be employed to investigate electric water pump control.

The important finding is that around 3% fuel consumption savings are possible over the NEDC by management strategies that lead to faster cylinder block warm up, even though this may result in little or no change in oil temperature as measured in the sump.
Declaration

I hereby declare that this thesis comprises only my original work towards the fulfillment of the degree of Master of Engineering Science and contains no material previously written or published by another person, except where duly referenced and acknowledged in the text. I further certify that the thesis contains less than 30,000 words, exclusive of tables, figures, bibliographies and appendices.

Faisal Samad Lodi
Acknowledgements

I am thankful to God, who provided me with the opportunity of completing a Master's degree by research.
I thank my family with all my heart, which always supported me in this long quest of knowledge and strengthened me emotionally.
I am thankful to Professor Harry C Watson, who shared his vast knowledge, experience and wisdom, unselfishly with me and who always reminds me of my own father being very reasonable and fair.
I would always be indebted to Mr. Robert Dingli from Ford Motor Co., Australia, who was available to help me in this long project, even in odd hours, without which, the project would not have been a success.
I am thankful to Mr. Glen Voice from Ford Motor Co., Australia, who was not only looking after the progress of this project but was involved with all enthusiasm and motivation. His enthusiasm and timely response to any query regarding the project or arrangement of any parts/tools/components was not only motivating but also add to the unforgettable memories of this project.
My gratitude can not be expressed in words for Dr. Chris Manzie, who was not only available with his advices and experience but also arranged the scholarship for me, which helped me to successfully complete the Master’s project.
I wish to express my thanks to laboratory technical staff Don Halpins and Ted Grange for their important role in the machining and installation of the test engine, which was a great opportunity for me to have hands on experience and to learn from their vast experience.
I am thankful to the fellow research students and friends, Pouria Mehrani, Farzad Keynejad, Mohammad Ali Khan, William Attard, Elisa Toulson and Phoung Pham, who provided me with all the support and help that I needed.
Abstract ........................................................................................................ iii

Declaration .................................................................................................... v

Acknowledgements .................................................................................. vii

List of Figures ........................................................................................... xix

List of Tables ............................................................................................... xxix

Nomenclature ............................................................................................... xxxi

Chapter 1 - Introduction .............................................................................. 1
  1.1 The Fuel Consumption Problem .......................................................... 1
    1.1.1 Introduction ................................................................................... 1
    1.1.2 Background .................................................................................. 2
  1.2 Objective ............................................................................................... 6
  1.3 Outline of Thesis .................................................................................. 7

Chapter 2 – Review of the Engine Cold Start Problem ......................... 9
  2.1 Introduction .......................................................................................... 9
  2.2 Effects of Mixture Preparation on Cold Start ................................... 9
3.5 Combustion Heat Transfer ................................................................. 52
  3.5.1 Heat Transfer to the Combustion Chamber ................................ 52
  3.5.2 Effects of Wall Material on Heat Transfer ............................... 56
  3.5.3 Heat Transfer from the Combustion Chamber Walls ............... 56
3.6 Intake and Exhaust System Heat Transfer ........................................ 57
3.8 Approach to Modeling in Modelica .................................................. 59
  3.8.1 Introduction to Modelica ........................................................... 59
3.9 Model in the Current Study ............................................................... 61
  3.9.1 Heat Transfer Model ................................................................. 61
3.10 Summary ......................................................................................... 64

Chapter 4 – Experimental Set up ......................................................... 65
  4.1 Introduction .................................................................................... 65
  4.2 The Test Engine .............................................................................. 65
  4.3 Experimental Targets ..................................................................... 66
    4.3.1 Preliminary Test ...................................................................... 66
    4.3.2 Main Experiments ................................................................. 66
    4.3.3 Variable Coolant Flow Rate Test ............................................ 67
    4.3.4 Split Cooling Test .................................................................. 67
  4.4 Experimental Strategy ..................................................................... 67
    4.4.1 Calculation of Mean Torque and Mean Engine Speed ............ 68
  4.5 Summary ......................................................................................... 73

Chapter 5 – Modeling Results .............................................................. 74
  5.1 Introduction .................................................................................... 74
  5.2 Model Simulation ........................................................................... 75
  5.3 Initial Conditions for Simulation .................................................... 75
  5.4 Simulation Results for NEDC Test Point (1161 rev/min, 48 Nm Load)
.......................................................... 76
    5.4.1 Head, Block, Head Coolant and Oil Results .......................... 76
    5.4.2 Piston and Cylinder Walls Results ......................................... 77
  5.5 Simulation Results for Idle (700 rev/min, 0 Nm Load) ................. 79
6.9.8 Split Cooling (Flow in Head 3.17 L/min) ...................................120
6.9.9 Split Cooling (Flow in Block 3.17 L/min) ...................................121
6.10 Biasing of Flow with Electric Water Pump ..................................123
6.11 Summary ....................................................................................123

Chapter 7 – Comparison of Theory and Measurement ..................124
7.1 Introduction ..................................................................................124
7.2 Comparison of Results from Modeling and Experiment ..............124
  7.2.1 Cylinder Head Temperature ....................................................124
  7.2.2 Engine Block Temperature .......................................................127
  7.2.3 Engine Oil Temperature ..........................................................128
  7.2.4 Cylinder Head Coolant Temperature .......................................130
7.3 Summary ......................................................................................131

Chapter 8 - Conclusions .................................................................133
8.1 Introduction ..................................................................................133
8.2 Conclusions ................................................................................133
  8.2.1 Coolant Flow Rate .................................................................133
  8.2.2 Fuel Consumption ................................................................134
  8.2.3 Split Cooling Supply ...............................................................134
  8.2.4 Engine Oil Warm-up ..............................................................135
  8.2.5 Improvement in Warm-up Time .............................................135
  8.2.6 Engine Heat Transfer Modeling ..............................................135
8.3 Recommendation for Future Work ..............................................136
  8.3.1 Further Work in Modeling ......................................................136
  8.3.2 Trapping Hot Coolant on Exhaust side of the Head ..................136
  8.3.3 Investigate Thermal Shock .....................................................137
  8.3.4 Measuring In-cylinder Pressure ..............................................137
  8.3.5 Flow Diverter Valve ...............................................................137
  8.3.6 Modification .........................................................................138
  8.3.7 Design Improvement .............................................................139
Reference List ........................................................................................................... 140

Appendix A – Mixture Preparation .................................................................. 151
A.1 Mechanism of Fuel Transportation ............................................................ 151
A.2 Factors affecting Mixture Preparation .................................................... 158
   A.2.1 Cyclic Variability and Spark Retard ............................................. 158
   A.2.2 Effects of Swirl and Tumble ....................................................... 160

Appendix B – Faster Warm-up Strategies ...................................................... 162
B.1 Development and Advancement of Engines ............................................. 162
B.2 Downsizing the Engine for better Fuel Economy .................................... 163
B.3 Varying Engine Parameters .................................................................... 163
B.4 Effects on UHC Emissions with Varying Coolant Temperature ........... 165

Appendix C – Engine Modifications and Thermocouple Locations ................. 166
C.1 Modifications in the Engine ................................................................. 166
C.2 Thermocouple Location in Engine Block ............................................. 167
   C.2.1 Block Inlet and Exhaust Thermocouples .................................. 167
   C.2.2 Block Bore Thermocouples .................................................... 168
   C.2.3 Block Circumference Thermocouples .................................... 168
C.3 Thermocouple Location in the Cylinder Head ....................................... 169
   C.3.1 Head Inlet and Exhaust Thermocouples .................................. 169
   C.3.2 Head Bore Thermocouples .................................................... 170
   C.3.3 Head Exhaust Valve Bridge Thermocouples ........................... 170
C.4 Thermocouple Locations for Cylinder Head Coolant ............................. 171
C.5 Thermocouple Location on Coolant Inlet and Outlet ............................. 172
C.6 Thermocouple Extension Cable ........................................................... 174
C.7 Different Group of Thermocouples ...................................................... 174
C.8 Installation of Air Bleed ........................................................................ 174
C.9 Installation of Electric Water Pump ...................................................... 176
C.10 Experimental Set-up for Split Cooling Supply ..................................... 177
C.10.1 Modifications for First Part of Split cooling ........................................ 177
C.10.2 Modifications for Second Part of Split Cooling .................................... 179

Appendix D – Engine Wiring and Data Acquisition System .............. 181
D.1 Engine Wiring .................................................................................. 181
D.1.1 Engine PCM Wiring ..................................................................... 181
D.1.2 Mains through a 12 Volts Battery ............................................... 182
D.1.3 Ignition Relay ............................................................................. 182
D.1.4 Radiator and Fan Assembly ...................................................... 183
D.1.5 Fuel Tank .................................................................................. 183
D.2 Dynamometer ................................................................................. 183
D.3 Data Acquisition and Instrumentation ............................................. 184
D.3.1 Data Acquisition System ........................................................... 184
D.3.2 Instrumentation ......................................................................... 185

Appendix E – Pressure Transducer and Flow Meter ....................... 186
E.1 Pressure Transducer ....................................................................... 186
E.2 Flow Meter .................................................................................... 187
E.2.1 Flow Rate at different Speeds ................................................... 189
E.2.2 Flow Rate Data supplied by the Manufacturer ........................... 191
E.3 Experimental Error Estimates ......................................................... 191

Appendix F – Experimental Strategy and Procedure ....................... 193
F.1 Initial Plan for Experimentation ....................................................... 193
F.2 New European Drive Cycle (NEDC) [1] ........................................ 194
F.3 Preliminary Testing ......................................................................... 196
F.3.1 Thermocouples in the First Group ............................................. 197
F.3.2 Thermocouples in the Second Group ........................................ 198
F.3.3 Thermocouples in the Third Group .......................................... 199
F.3.4 Thermocouples in the Fourth Group ........................................... 200
F.4 Main Testing .................................................................................. 200
F.4.1 Thermocouples in the First Group ............................................. 201
F.4.2 Thermocouples in the Second Group ........................................ 202
F.4.3 Thermocouples in the Third Group ...................................... 203
F.4.4 Thermocouples in the Fourth Group .................................... 204
F.5 Installation of Flow meter and Thermocouple Modules .......... 205
  F.5.1 Thermocouple Module 1 ..................................................... 206
  F.5.2 Thermocouple Module 2 ..................................................... 207
  F.5.3 Thermocouple Module 3 ..................................................... 208
F.6 Electric Water Pump Testing .................................................. 208
F.7 Split Cooling System Testing ................................................ 209

Appendix G – Heat Transfer Model ............................................. 211
G.1 Model for Representative Cylinder ....................................... 211
  G.1.1 Combustion Chamber ...................................................... 211
  G.1.2 Crankshaft ..................................................................... 212
  G.1.3 Coolant Model ............................................................... 213
  G.1.4 Direction of Coolant Flow in the Model ......................... 213
  G.1.5 Upper Cylinder Wall Model ........................................... 213
  G.1.6 Middle Cylinder Wall Model ......................................... 214
  G.1.7 Lower Cylinder Wall Model .......................................... 216
  G.1.8 Oil Sump Model ............................................................. 216
  G.1.9 Cylinder Volume Model ................................................ 216
  G.1.10 Piston and Piston Crown Model ................................. 217
  G.1.11 Inlet and Exhaust Port Model ..................................... 217
  G.1.12 Head Model ................................................................. 219
  G.1.13 Combined Ports and Head Model .............................. 219
  G.1.14 Head Heat Exchanger Model ................................... 219
  G.1.15 Oil Heat Exchanger Model ....................................... 220
  G.1.16 Engine Block Model ................................................... 220

Appendix H – Results for MAP, Spark Advance and Engine Speed
........................................................................................................ 222
H.1 MAP Results ....................................................................... 222
H.1.1 MAP in Preliminary Test ........................................................... 222
H.1.2 MAP in Main Test ................................................................. 222
H.2 Spark Advance ........................................................................... 225
  H.2.1 General Understanding of the Controller Varying the Spark ... 225
  H.2.2 Comparison for Spark Advance ........................................... 225
  H.2.3 Spark Advance in Preliminary Idle Test ......................... 225
  H.2.4 Spark Advance in Main Test ............................................. 226
H.3 Engine Speed ........................................................................... 229
  H.3.1 Preliminary Idle Tests ....................................................... 229
  H.3.2 Main Tests ...................................................................... 229

Appendix – I : Biasing of Flow ......................................................... 232
  I.1 Introduction ........................................................................... 232
  I.2 Biasing of the Flow in 17.17 L/min EWP .......................... 232
  I.3 Comparison of MWP and EWP for 17.17 L/min .......... 233
    I.3.1 Cylinder 1 ...................................................................... 233
    I.3.2 Cylinder 4 ...................................................................... 234
    I.3.3 Cylinder 5 ...................................................................... 235
    I.3.4 Cylinder 6 ...................................................................... 235
  I.4 General discussion on Cylinder 1 Vs Cylinder 6 for 17.17 L/min .... 239
    I.4.1 Comparison between Coolant Temperatures ........... 239
    I.4.2 Comparison between BE1 and BE6, BI1 and BI6 ...... 240
    I.4.3 Comparison between HB1, HB6 and HB7 ................. 240
    I.4.4 Comparison between HE1 and HE6, HI1 and HI6 .... 241
    I.4.5 Higher Capacitance in EWP ........................................... 241
  I.5 Comparison between 17.17 L/min Vs 3.17 L/min EWP .......... 243
    I.5.1 Cylinder 1 ...................................................................... 243
    I.5.2 Cylinder 2 and Cylinder 3 ............................................. 243
    I.5.3 Cylinder 4 ...................................................................... 244
    I.5.4 Cylinder 5 ...................................................................... 244
    I.5.5 Cylinder 6 ...................................................................... 244
  I.6 Discussion on Metal Temperatures ........................................... 248
I.7 Summary...........................................................................................251
# List of Figures

1.1 Comparison of SAE-5W/30 (right) and SAE-15W/40 (left) at different Temperatures ..................................................5

2.1 Overview of an Engine Behavior during Start-up from Observations of Raw Data sets .........................................................11

2.2 Schematic diagram of factors causing excess fuel consumption in a cold engine ..................................................................12

2.3 The variation of Lubricating Oil and Water Temperatures for Ford CVH Engine with a constant Throttle Step Warm-up .........................16

2.4 Frictional Power Loss as a function of the Lubricating Oil Sump Temperature during a Step Warm-up at constant Throttle ..............................................................................................................18

2.5 Heat transported by the Lubricant During Engine Warm-up ........................................................................................................21

2.6 Lubricant and Coolant Temperatures during Engine Warm-up ........................................................................................................22

2.7 Influence of Engine Speed on Oil Warm-up ........................................23

2.8 Warm-up times for Mechanical Water Pump, Electric Water Pump and Electric Water Pump with No Heater Demand ..........................41

3.1 Thermal response of engine components ........................................50

3.2 Thermal response of exhaust gas .................................................51

3.3 Schematic heat flow chart of the Resistor-Capacitor model ........63
4.1 Prediction of (a) Vehicle Speed (b) Engine Speed and (c) Engine Torque from CARSIM model.............................................................................................................69

4.2 Engine Schematic diagram and Data Acquisition Board......................70

4.3 Engine Test Cell and Data Acquisition Boards.....................................71

5.1 Modeling results for engine block temperature, cylinder head temperature, head coolant temperature and engine oil temperature at NEDC test point (1161 rev/min and 48 Nm load)................................................................................................................77

5.2 Modeling results for piston crown temperature, piston skirt temperature, cylinder upper wall temperature, cylinder middle wall temperature and cylinder lower wall temperature at NEDC test point (1161 rev/min and 48 Nm load)................................................................................................................78

5.3 Modeling results for engine block temperature, cylinder head temperature, head coolant temperature and engine oil temperature at Idle 700 rev/min and 0 Nm load........................................................................................................80

5.4 Modeling results for piston crown temperature, piston skirt temperature, cylinder upper wall temperature, cylinder middle wall temperature and cylinder lower wall temperature at Idle 700 rev/min and 0 Nm load........81

6.1 Engine block schematic diagram with coolant transfer holes.............83

6.2 Schematic of coolant flow in the engine.................................................83

6.3 Average Fuel Consumption, Cylinder Head, Engine Block and Oil temperatures for Idle (9.17 L/min)................................................................................................................91

6.4 (a) Average Fuel Consumption, Cylinder Head, Engine Block and Oil temperatures for 17.17 L/min MWP.................................................................93
6.4 (b) Average Fuel Consumption, Cylinder Head, Engine Block and Oil temperatures for 17.17 L/min EWP ................................................................. 94

6.5 Average Fuel Consumption, Cylinder Head, Engine Block and Oil temperatures for 3.17 L/min EWP ................................................................. 95

6.6 Average Fuel Consumption, Cylinder Head, Engine Block and Oil temperatures for 0 L/min EWP ................................................................. 98

6.7 Average Fuel Consumption, Cylinder Head, Engine Block and Oil temperatures for Split (3.17 L/min, flow in head) ............................................. 99

6.8 Average Fuel Consumption, Cylinder Head, Engine Block and Oil temperatures for Split (3.17 L/min, flow in block) ....................................... 100

6.9 Comparison of total fuel consumption for repetitive tests for 17.17 L/min MWP ......................................................................................... 104

6.10 Comparison of total fuel consumption for repetitive tests for 17.17 L/min EWP ......................................................................................... 104

6.11 Comparison of total fuel consumption for repetitive tests for 3.17 L/min EWP ......................................................................................... 105

6.12 Comparison of total fuel consumption for repetitive tests for 0 L/min EWP ......................................................................................... 105

6.13 Comparison of total fuel consumption for repetitive tests for split (3.17 L/min, flow in head) ................................................................. 106

6.14 Comparison of total fuel consumption for repetitive tests for split (3.17 L/min, flow in block) ................................................................. 106

6.15 Fuel consumption summary over half minute interval .................. 108
6.16 Comparison of 3.17 L/min and 17.17 L/min (EWP) to 0 L/min EWP for %
reduction in fuel consumption.........................................................109

6.17 Comparison of Split (flow in block, 3.17 L/min) with 17.17 L/min EWP, 3.17
L/min EWP and 0 L/min EWP for % reduction in fuel consumption......110

6.18 Comparison for coolant thermocouples at Idle .................................116

6.19 Comparison of coolant thermocouples for 17.17 L/min MWP..............117

6.20 Comparison of coolant thermocouples for 17.17 L/min EWP.............118

6.21 Comparison of coolant thermocouples for 3.17 L/min EWP..............119

6.22 Comparison of coolant thermocouples for 0 L/min EWP...............120

6.23 Comparison of coolant thermocouples for split (3.17 L/min flow in
head)................................................................................................122

6.24 Comparison of coolant thermocouples for split (3.17 L/min flow in
block)................................................................................................122

7.1 (a) Comparison of predicted and experimental cylinder head temperature for
1161 rev/min and 48 Nm Load............................................................126

7.1 (b) Comparison of predicted and experimental cylinder head temperature for
700 rev/min and 0 Nm Load.................................................................126

7.2 (a) Comparison of predicted and experimental engine block temperature for
1161 rev/min and 48 Nm Load.............................................................127

7.2 (b) Comparison of predicted and experimental engine block temperature for
700 rev/min and 0 Nm Load.................................................................128
7.3 (a) Comparison of predicted and experimental engine oil temperature for 1161 rev/min and 48 Nm Load

7.3 (b) Comparison of predicted and experimental engine oil temperature for 700 rev/min and 0 Nm Load

7.4 (a) Comparison of predicted and experimental cylinder head coolant temperature for 1161 rev/min and 48 Nm Load

7.4 (b) Comparison of predicted and experimental cylinder head coolant temperature for 700 rev/min and 0 Nm Load

8.1 Schematic Diagram for External Oil Sump

A.1 Mass of Fuel Injected and Fuel Required for Stoichiometric Mixture for Cylinder 1 of 4.6 L V8 showing initial over-fueling and then under-fueling

A.2 Possible Intervals of Mixture Back Flow in the Inlet Port

A.3 Air Temperature Profile in the Inlet Port, where the position along the port is in inches

A.4 Variations Measured in the Exhaust and Inlet AFR at 2000 rev/min 0.6 Bar MAP, 11.1 AFR supply – 20°C from a motored start [14]

A.5 Cold Fluid Spark Retard Results at various A/F (2.0 L, 4 Valve Engine, 1200 rev/min, 1.0 Bar BMEP, 20°C Fluid Temperature

B.1 Engine Thermal Balance and Heat Used during Warm-up (Engine Speed = 2000 rev/min and Torque = 10 Nm)

C.1 Thermocouple locations on Engine Block

C.2 Thermocouple location on Cylinder Head
C.3 Thermocouple location for exhaust side coolant temperature…………..172

C.4 Thermocouple location before thermostat (coolant temperature at outlet from the engine)…………………………………………………………………………..173

C.5 Thermocouple location at water pump inlet………………………………..174

C.6 Installation of air bleed…………………………………………………………….175

C.7 Installation of Electric Water Pump………………………………………………177

C.8 Split cooling supply with 3.17 L/min flow in head and 0 L/min flow in block……………………………………………………………………………….179

C.9 Split cooling supply with 3.17 L/min flow in block and 0 L/min flow in head……………………………………………………………………………….180

D.1 Radiator and Fan Assembly and Fuel Tank……………………………………183

D.2 ATI Vision Hub and Thermocouples Modules………………………………..184

E.1 Pressure Transducer installation…………………………………………………187

E.2 Variable area flow meter installed on the engine……………………………..189

E.3 Coolant Flow rate for different speed and load (thermostat closed)….190

E.4 Coolant Flow rate data supplied by Ford Motor Co. (Thermostat open) …………………………………………………………………………………..191

F.1 ECE 15 emissions cycle .................................................................195

F.2 EUDC Cycle .....................................................................................195
H.1 Comparison of MAP for Idle .............................................................. 223
H.2 Comparison of MAP for 17.17 L/min MWP ........................................... 223
H.3 Comparison of MAP for 17.17 L/min EWP ............................................. 223
H.4 Comparison of MAP for 3.17 L/min EWP .............................................. 223
H.5 Comparison of MAP for 0 L/min EWP ..................................................... 224
H.6 Comparison of MAP for split (3.17 L/min, flow in head) ...................... 224
H.7 Comparison of MAP for split (3.17 L/min, flow in block) ..................... 224
H.8 Comparison of Total Spark Advance for Idle tests .............................. 227
H.9 Comparison of Total Spark Advance for 17.17 L/min MWP ................. 227
H.10 Comparison of Total Spark Advance for 17.17 L/min EWP ................. 228
H.11 Comparison of Total Spark Advance for 3.17 L/min EWP ................. 228
H.12 Comparison of Total Spark Advance for 0 L/min EWP ......................... 228
H.13 Comparison of Total Spark Advance for split (3.17 L/min flow in head) .............................................................................................................. 228
H.14 Comparison of Total Spark Advance for split (3.17 L/min flow in block) .............................................................................................................. 229
H.15 Comparison of engine speed for 17.17 L/min MWP .............................. 230
H.16 Comparison of engine speed for 17.17 L/min EWP .............................. 230
H.17 Comparison of engine speed for 3.17 L/min EWP .............................. 230
H.18 Comparison of engine speed for 0 L/min EWP..............................230

H.19 Comparison of engine speed for split (3.17 L/min flow in head)........231

H.20 Comparison of engine speed for split (3.17 L/min flow in block)...........231

I.1 (a) Comparison for cylinder 1 (MWP and EWP 17.17 L/min).....................237

I.1 (b) Comparison for BE1 and BI1 (MWP and EWP 17.17 L/min).................237

I.2 (a) Comparison for cylinder 3 (MWP and EWP 17.17 L/min).....................237

I.2 (b) Comparison for HI3 (MWP and EWP 17.17 L/min)..............................237

I.3 (a) Comparison for cylinder 4 (MWP and EWP 17.17 L/min).....................238

I.3 (b) Comparison for BE4 and BI4 (MWP and EWP 17.17 L/min).................238

I.4 Comparison for cylinder 5 (MWP and EWP 17.17 L/min).........................238

I.5 (a) Comparison for cylinder 6 (MWP and EWP 17.17 L/min).....................239

I.5 (b) Comparison for BE6 and BI6 (MWP and EWP 17.17 L/min)...............239

I.6 Comparison for BI1 and BI6 (MWP and EWP 17.17 L/min)......................242

I.7 Comparison for HI1 and HI6 (MWP and EWP 17.17 L/min)......................242

I.8 Comparison for HB1, HB6 and HB7 (MWP and EWP 17.17 L/min).........242

I.9 Comparison for HE1 and HE6 (MWP and EWP 17.17 L/min).................242

I.10 (a) Comparison for cylinder 1 (17.17 L/min and 3.17 L/min EWP)..........245
I.10 (b) Comparison for BE1 and BI1 (17.17 L/min and 3.17 L/min EWP) ....... 245
I.11 Comparison for cylinder 2 (17.17 L/min and 3.17 L/min EWP) .......... 246
I.12 Comparison for cylinder 3 (17.17 L/min and 3.17 L/min EWP) ........... 246
I.13 (a) Comparison for cylinder 4 (17.17 L/min and 3.17 L/min EWP) ........ 246
I.13 (b) Comparison for BE4 and BI4 (17.17 L/min and 3.17 L/min EWP) .... 246
I.14 (a) Comparison for cylinder 5 (17.17 L/min and 3.17 L/min EWP) ........ 247
I.14 (b) Comparison for BI5 (17.17 L/min and 3.17 L/min EWP) .............. 247
I.15 (a) Comparison for cylinder 6 (17.17 L/min and 3.17 L/min EWP) ........ 247
I.15 (b) Comparison for BE6 and BI6 (17.17 L/min and 3.17 L/min EWP) .... 247
I.16 Comparison for block bore temperatures for split cooling (flow in head and block) ............................................................................................................ 250
I.17 Comparison for block bore temperatures for 3.17 L/min and 0 L/min EWP ........................................................................................................... 250
I.18 Comparison for BE and HV thermocouples for 3.17 L/min and 0 L/min EWP ........................................................................................................... 250
I.19 Metal temperatures for 0 L/min EWP .................................................. 250
List of Tables

2.1 Percentage of Power Losses ......................................................... 14

2.2 Comparison between an Electrical Water Pump and a Mechanical Water Pump ................................................................. 42

4.1 BF Ford Falcon Engine Specifications ............................................. 72

6.1 Average temperature of Cylinder Head ............................................ 89
6.2 Average temperature of Engine Block ........................................... 89
6.3 Average temperature of Engine Oil ............................................... 90
6.4 Average temperature of Cylinder Head Coolant .............................. 90
6.5 Average Fuel Consumption Rate .................................................... 101
6.6 Temperature profile for Cylinder 1 ................................................ 111
6.7 Temperature profile for Cylinder 2 ................................................ 112
6.8 Temperature profile for Cylinder 3 ................................................ 112
6.9 Temperature profile for Cylinder 4 ................................................ 113
6.10 Temperature profile for Cylinder 5 ............................................... 113
6.11 Temperature profile for Cylinder 6 ............................................... 114
6.12 Temperature profile for Water Pump Inlet ...................................... 114
6.13 Temperature profile for Thermostat .............................................. 115
C.1 Split cooling strategy ................................................................. 180
E.1 Flow meter reading on different speed and load.........................190

E.2 Experimental Error Estimates..............................................192

F.1 First Group Thermocouples in Preliminary Tests......................197

F.2 Second Group Thermocouples in Preliminary Tests...................198

F.3 Third Group Thermocouples in Preliminary Tests.....................199

F.4 Fourth Group Thermocouples in Preliminary Tests...................200

F.5 First Group Thermocouples in Main Tests...............................201

F.6 Second Group Thermocouples in Main Tests.............................202

F.7 Third Group Thermocouples in Main Tests...............................203

F.8 Fourth Group Thermocouples in Main Tests.............................204

F.9 Thermocouple Module 1 in Main Tests with Electric Water Pump...206

F.10 Thermocouple Module 2 in Main Tests with Electric Water Pump...207

F.11 Thermocouple Module 3 in Main Tests with Electric Water Pump...208

F.12 Series of Tests undertaken with EWP....................................209

F.13 Series of Tests undertaken for Split Cooling System...............210
# Nomenclature

## Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>ABDC</td>
<td>After Bottom Dead Center</td>
</tr>
<tr>
<td>ADR</td>
<td>Australian Design Rules</td>
</tr>
<tr>
<td>AFR</td>
<td>Air Fuel Ratio</td>
</tr>
<tr>
<td>ATDC</td>
<td>After Top Dead Center</td>
</tr>
<tr>
<td>ATI</td>
<td>Accurate Technologies Inc. (Data Acquisition System)</td>
</tr>
<tr>
<td>BB</td>
<td>Block Bore Thermocouple</td>
</tr>
<tr>
<td>BC</td>
<td>Block Circumference Thermocouple</td>
</tr>
<tr>
<td>BBDC</td>
<td>Before Bottom Dead Center</td>
</tr>
<tr>
<td>BDC</td>
<td>Bottom Dead Center</td>
</tr>
<tr>
<td>BE</td>
<td>Block Exhaust Thermocouple</td>
</tr>
<tr>
<td>BI</td>
<td>Block Inlet Thermocouple</td>
</tr>
<tr>
<td>BMEP</td>
<td>Brake Mean Effective Pressure</td>
</tr>
<tr>
<td>BTDC</td>
<td>Before Top Dead Center</td>
</tr>
<tr>
<td>BSFC</td>
<td>Brake Specific Fuel Consumption</td>
</tr>
<tr>
<td>CA</td>
<td>Crank Angle</td>
</tr>
<tr>
<td>CCC</td>
<td>Close Coupled Catalyst</td>
</tr>
<tr>
<td>CAD</td>
<td>Computer Aided Design</td>
</tr>
<tr>
<td>CAE</td>
<td>Computer Aided Engineering</td>
</tr>
<tr>
<td>CR</td>
<td>Compression Ratio</td>
</tr>
<tr>
<td>CSSRE</td>
<td>Cold Start Spark Retard for Reduced Emissions</td>
</tr>
<tr>
<td>DAQ</td>
<td>Data Acquisition</td>
</tr>
<tr>
<td>DIHC</td>
<td>Direct Injection Homogeneous Charge</td>
</tr>
<tr>
<td>DISC</td>
<td>Direct Injection Stratified Charge</td>
</tr>
<tr>
<td>DOHC</td>
<td>Double Overhead Camshaft</td>
</tr>
<tr>
<td>ECU</td>
<td>Engine Control Unit</td>
</tr>
<tr>
<td>EGR</td>
<td>Exhaust Gas Recirculation</td>
</tr>
<tr>
<td>EU</td>
<td>European Union</td>
</tr>
<tr>
<td>EVC</td>
<td>Exhaust Valve Close</td>
</tr>
<tr>
<td>EVO</td>
<td>Exhaust Valve Open</td>
</tr>
<tr>
<td>EWP</td>
<td>Electric Water Pump</td>
</tr>
<tr>
<td>Acronym</td>
<td>Description</td>
</tr>
<tr>
<td>---------</td>
<td>-------------</td>
</tr>
<tr>
<td>FMEP</td>
<td>Frictional Mean Effective Pressure</td>
</tr>
<tr>
<td>GDI</td>
<td>Gasoline Direct Injection</td>
</tr>
<tr>
<td>HB</td>
<td>Head Bore Thermocouple</td>
</tr>
<tr>
<td>HC</td>
<td>Hydrocarbons</td>
</tr>
<tr>
<td>HE</td>
<td>Head Exhaust Thermocouple</td>
</tr>
<tr>
<td>HI</td>
<td>Head Inlet Thermocouple</td>
</tr>
<tr>
<td>HV</td>
<td>Exhaust Valve Bridge Thermocouple</td>
</tr>
<tr>
<td>IC</td>
<td>Internal Combustion</td>
</tr>
<tr>
<td>IMEP</td>
<td>Indicated Mean Effective Pressure</td>
</tr>
<tr>
<td>IVC</td>
<td>Inlet Valve Close</td>
</tr>
<tr>
<td>IVO</td>
<td>Inlet Valve Open</td>
</tr>
<tr>
<td>L</td>
<td>Liter</td>
</tr>
<tr>
<td>MAP</td>
<td>Manifold Absolute Pressure</td>
</tr>
<tr>
<td>MWP</td>
<td>Mechanical Water Pump</td>
</tr>
<tr>
<td>NEDC</td>
<td>New European Drive Cycle</td>
</tr>
<tr>
<td>PCM</td>
<td>Ford Power-Train Control Module</td>
</tr>
<tr>
<td>SI</td>
<td>Spark Ignition</td>
</tr>
<tr>
<td>SOHC</td>
<td>Single Overhead Camshaft</td>
</tr>
<tr>
<td>ST</td>
<td>Spark Timing</td>
</tr>
<tr>
<td>TDC</td>
<td>Top Dead Center</td>
</tr>
<tr>
<td>THC</td>
<td>Total Hydro-Carbon</td>
</tr>
<tr>
<td>TWC</td>
<td>Three way Catalyst</td>
</tr>
<tr>
<td>UHC</td>
<td>Unburnt Hydro-Carbon</td>
</tr>
<tr>
<td>VVT</td>
<td>Variable Valve Timing</td>
</tr>
<tr>
<td>WOT</td>
<td>Wide Open Throttle</td>
</tr>
</tbody>
</table>

**DIMENSIONLESS GROUPS**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nu</td>
<td>Nusselt Number</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl Number</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds Number</td>
</tr>
</tbody>
</table>

**SYMBOLS**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>g</td>
<td>Acceleration due to Gravity</td>
</tr>
<tr>
<td>D</td>
<td>Bore</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>------------------------------</td>
</tr>
<tr>
<td>$P$</td>
<td>Combustion Pressure</td>
</tr>
<tr>
<td>$T$</td>
<td>Combustion Temperature</td>
</tr>
<tr>
<td>$h$</td>
<td>Convective Heat Transfer Coefficient</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Crank Angle</td>
</tr>
<tr>
<td>$Vol$</td>
<td>Cylinder Volume</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density of the gas</td>
</tr>
<tr>
<td>$C_d$</td>
<td>Discharge Coefficient</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Dynamic Viscosity</td>
</tr>
<tr>
<td>$N$</td>
<td>Engine Speed</td>
</tr>
<tr>
<td>$\dot{Q}$</td>
<td>Heat Flux</td>
</tr>
<tr>
<td>$h_w$</td>
<td>Length of the Cylinder Wall</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>Mass flow rate</td>
</tr>
<tr>
<td>$U_m$</td>
<td>Mean Piston Speed</td>
</tr>
<tr>
<td>mod</td>
<td>Modulus of Engine Speed</td>
</tr>
<tr>
<td>$M$</td>
<td>Molecular Weight of the gas</td>
</tr>
<tr>
<td>$A_R$</td>
<td>Reference Area</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Specific Heat of the gas</td>
</tr>
<tr>
<td>$P_o$</td>
<td>Stagnation Pressure</td>
</tr>
<tr>
<td>$T_o$</td>
<td>Stagnation Temperature</td>
</tr>
<tr>
<td>$\Delta T/\Delta x$</td>
<td>Temperature Gradient</td>
</tr>
<tr>
<td>$k$</td>
<td>Thermal Conductivity of the Material</td>
</tr>
<tr>
<td>$P_t$</td>
<td>Throat Pressure</td>
</tr>
<tr>
<td>$R$</td>
<td>Universal Gas Coefficient</td>
</tr>
<tr>
<td>$A_v$</td>
<td>Valve Curtain Area</td>
</tr>
<tr>
<td>$D_v$</td>
<td>Valve Diameter</td>
</tr>
<tr>
<td>$L_v$</td>
<td>Valve Lift</td>
</tr>
<tr>
<td>$t$</td>
<td>Wall Thickness</td>
</tr>
</tbody>
</table>
Chapter 1 - Introduction

1.1 The Fuel Consumption Problem

1.1.1 Introduction

There have been developments in SI engine performance at a very rapid pace and to some extent, a threshold has been reached on the performance side. Therefore, researchers and automotive manufacturers have now shifted their concern on reducing cost of the vehicles in use. There are new smaller automotive vehicles, which are in greater demand than the traditional larger six cylinder engine cars, which have hitherto dominated the Australian market. Car manufacturers are now focusing on smaller sized engines, with cost and fuel economy being the primary concerns. Downsizing the engine not only reduces the amount of materials needed and to some extent cost, but it also enhances the performance of the engine, mainly affecting the fuel economy. There has been a lot of work done in reducing fuel consumption of an automotive vehicle under steady state operating conditions, not only to cope with the everyday increasing operating costs of the fuel, but also to meet the stringent regulations on emissions. There has also been a lot of work directed towards improving
the transient operating conditions of the engine, mainly when the engine is cold started, because this is the operating condition where the highest fuel consumption penalty occurs accompanied by highest emissions. Nonetheless, the problem persists, and means to get faster warm-up of the engine still remains an important goal.

1.1.2 Background

When an engine is cold started, fuel enters the combustion chamber in the form of vapor and droplets. In order to start an engine, a combustible mixture of fuel vapor and air, rich enough to ignite easily at starting temperatures must be supplied near the spark plug. The approximate limits of flammability of air gasoline vapor mixture are about 8:1 for fuel rich mixture and 20:1 for lean mixture. A mixture of 12:1, which is the fastest burning mixture, is most suitable for starting. From the standpoint of fuel, the problem of cold starting is largely one of getting sufficient fuel evaporation, a more volatile fuel is desirable. However, with current gasoline, there are limits on volatility, to limit evaporative emissions from the fuel tank and inlet manifold when the vehicle is standing. Cold starting is also affected by front-end volatility of the gasoline used. Cold starting can be improved if front-end volatility is higher, but it may lead to increased problem of hot starting and vapor lock [15]. Hence, the air fuel ratio supplied at the cold start is far richer than the air fuel ratio actually being burnt and the tendency of the combustion chamber to allow liquid or vapor to pass to the exhaust occurs, because of the cold conditions. All these factors lead to higher emissions and poor fuel economy.

Furthermore, because the coolant fluid is cold, the hot air demister and the cabin heater do not function. To make the situation worse, the exhaust system catalyst is inactive at low temperatures. It becomes active only when it reaches the light off temperature, in the region of $150^\circ C$, and hence most of the exhaust, which leaves the exhaust pipe, is left untreated, as discussed by Sandoval and Heywood [16]. Some multifunction catalytic
converters, which use a three way catalyst and an upstream absorber, work on the principle that all the raw unburned hydrocarbons are absorbed in the front catalyst, which acts as a sponge and they are desorbed, as it warms up and treated later, by the second catalyst, once it has ‘lit-off’. The situation only improves once the engine is fully warmed up. However, it has been shown in previous studies that many car journeys are too small for the engine to fully warm up, as discussed by Cole et al [17]. Researchers have extensively studied the influence of cold start on fuel economy and have found that it is a function of cold start temperature as well as the trip length. Correlations developed by researchers at General Motors show that at $25^\circ C$, the relationship between fuel consumption from cold start to a warmed up engine fuel consumption is given by:

$$\frac{FC_{cold\ start}}{FC_{warmed\ up}} = 1 + L^{-0.8}$$

(1.1)

where $L$ is the trip length in miles [18].

Moreover, there are many environmental and human factors involved, which have their own effects on the fuel consumption, as discussed by Watson et al [19].

Once a vehicle is running, the best way to warm it up is to deliver power or consume energy. With computer-controlled, fuel-injected engines, it needs no more than 30 seconds of idling on winter days before driving away (for severe cold conditions, for example in Northern U.S). For less cold conditions, this time can be even shorter, down to a few seconds. Anything more simply wastes fuel and increases emissions. Besides, not only does the engine need to be warmed up but also do the wheel bearings, power steering, suspension, transmission and tires, to reduce friction. All of which can be done only when the vehicle is moving. A typical vehicle must be driven for at least five kilometers to warm up these parts [20]. Although, it is important to drive away as soon as possible after a cold start (but not before the windows are defrosted), high speeds and rapid acceleration for the first five kilometers or so, should be avoided (in order to keep the fuel consumption down). High speeds and acceleration bring a faster warm-up,
but it increases the fuel consumption. The goal is to bring the whole vehicle up to peak operating temperature as quickly as possible while maximizing the fuel economy. Therefore, there is always a trade off between rich mixtures needed for combustion in cold engines and the power to warm up the components.

Due to the increasing demand for both environmentally friendly and cost-effective vehicles, heat management of the power-train system is becoming more important. One aspect of this is the optimization of cold-start and warm-up performance and their interaction must be considered. To realize an advantage in fuel consumption for vehicles, suitable cold-start and warm-up strategies are required. The examination of a 6-cylinder gasoline engine and an automatic transmission shows how important the development of a suitable heat transfer concept is for the automotive fluids such as coolant, engine and transmission oil, in order to obtain further reduction in fuel consumption. By using a bypass integrated into the cooling circuit of the automatic transmission or transmission oil-water heat exchanger that takes advantage of the heat of the engine coolant the transmission reaches its working temperature more quickly in the EU-3-test [21].

In a cold started spark ignition engine, more fuel is consumed and subsequently more emissions are produced. Under urban drive conditions, 35% - 45% of the fuel goes into overcoming friction in the engine. During cold start, this figure can be up to 20% higher, which indicates that reducing friction during cold start through enhanced thermal management is a significant area of fuel economy improvement.

Prominently, the fuel consumption can be decreased by reducing the engine friction, without having any loss on the performance side because a large amount of energy goes into overcoming engine friction, out of which a small portion is transferred to the oil and coolant and a large part is lost to the surroundings. It signifies that engine warm up does not use all the energy produced by combustion and frictional losses.
The losses are much higher at low temperature due to the losses that occur in the hydrodynamically lubricated components, such as the bearings and the piston assembly. On the other hand, power losses in the valve train may decrease with lower temperatures, when the boundary layer lubrication thickness in sliding components may be thicker. A 40% increase in friction for the thicker oil corresponds to an increase in fuel consumption of about 12%. Changing to a lower-viscosity-grade lubricant would lead to a decrease in engine friction that could correspond to an improvement of 4% in fuel consumption.

Figure 1.1 shows the comparison of SAE-5W/30 and SAE-15W/40 grades of oil at different temperatures [2]. The comparison has been done for 100°C (upper figure) and 40°C (lower figure) for a 2L European gasoline engine operating under medium load.

![Figure 1.1: Comparison of SAE-5W/30 (right) and SAE-15W/40 (left) at different Temperatures [2]](image)
Fuel-economy benefits, however, are difficult to demonstrate in practice because many factors, such as weather conditions, engine conditions and driver behavior can have a much more significant effect on fuel consumption than the lubricant viscosity. The Bureau of Transport and Regional Economics has recently published figures on both fuel consumption and engine performance of Australia’s passenger vehicle fleet from the mid 1970’s and the trend indicates that both are decreasing [22]. Viewing all these problems collectively, it can be easily concluded that there is a significant need for faster engine warm up.

1.2 Objective
The importance of engine performance during cold start has now been well recognized, not only in terms of fuel consumption penalty but also in emission regulation and the emission laws getting tougher year by year. The current background highlights the problem of the cold start fuel consumption penalty. The question motivating this research is how to achieve faster warm-up of the engine to reduce the fuel economy penalty. The options for reducing engine friction relate to faster heating of the interfacing surfaces between the lubricating oil and the rotating and reciprocating parts in the engine. The goal of this project is to validate the hardware improved cold start fuel economy. The strategies to be investigated include the existing hardware but utilizing improved engine control for minimum fuel usage, as well as new hardware related strategies. With the goal of achieving a faster warm-up of the engine, the objectives of this study may be described as follows:

- To investigate the effects of changes to component temperature on heat transfer between the various engine components, especially from engine block to the lubricating oil.

- To develop a lumped parameter model for engine coolant flow, heat capacities and model heat transfer to the cylinder walls based on
engine operating points (manifold pressure, engine speed, spark advance, etc.) and to calibrate the model with experimental data.

- To investigate simple strategies utilizing an electric water pump, with variable flow and speed control to vary the coolant flow rate from maximum to zero flow, to bring a faster warm-up of the engine.

- To investigate the effect of faster coolant warm-up on oil warm-up.

- To investigate the strategy of split fluid supply to cylinder head and engine block, which can be categorized in two parts. They are:
  1. To run the cylinder head with no flow rate and to run low flow in the block.
  2. To run the engine block with no flow rate and to run low flow in the cylinder head.

1.3 Outline of Thesis

The present chapter begins with the background of the fuel consumption problem, the basic factors leading to more fuel consumption, when the engine is started. Moreover, in this chapter, the effects of friction on cold starting and the effects of cold oil and its viscosity are also discussed. Finally, it covers the research objectives and the motivation behind taking up this problem as a research project.

Chapter 2 presents the relevant literature behind the research topic, mainly covering the factors leading to more fuel consumption during cold start, like poor mixture preparation, friction etc. Moreover, a detailed review of engine heat transfer and the engine cooling system is also covered in this chapter.

Chapter 3 includes some literature on the background of modeling, different approaches to modeling engine heat transfer and the advantages and disadvantages associated with each approach. Moreover, the model in the current study is also discussed in this chapter.
Chapter 4 covers an overview of the experimental strategy and targets to be achieved. However, complete details about the engine installation, engine modifications, engine wiring and testing procedure and remarks are covered in the appendices.

Chapter 5 includes a discussion on the modeling results. Results for cylinder head temperature, engine block temperature, engine oil temperature and cylinder head coolant temperature are discussed in this chapter.

Chapter 6 focuses on the experimental results, which includes results of fuel consumption, MAP, engine speed, spark advance and coolant thermocouple (coolant temperature) behavior. Further discussion on the flow pattern of the coolant, metal temperature of the cylinder head and engine block and the flow biasing with the utilization of the electric water pump and its effects on component temperature is covered in Appendix I.

Chapter 7 includes the comparison between experimental and modeling results and some illustrative applications.

Chapter 8 finally concludes the whole work with future scope of research and recommendations.
Chapter 2
Review of the Engine Cold Start Problem

2.1 Introduction
The current chapter presents a discussion on the background of the problem of cold starting. A brief review on the effects of mixture preparation in cold start conditions and the complexities of engine heat transfer during transient operating conditions is done. Extra friction, which is one of the major contributors to the fuel economy penalty, during the start of the engine, is also discussed in this chapter. Moreover, the role of coolant in the performance of the engine is discussed, which includes the effects of varying the coolant flow rate on engine heat transfer, warm-up time, temperature of components and fuel consumption.

2.2 Effects of Mixture Preparation on Cold Start
The first reason for the fuel consumption penalty in a cold start engine, which has been well analyzed in the literature [7,8,14,23-27] as one of the major initial contributors to the excess fuel consumption, is improper mixture preparation. Fuel enters the combustion chamber in the form of vapor and droplets and the air fuel mixture at the start of an engine needs
to be far richer than that actually burnt under normal conditions\(^1\). Moreover, with the tendency of the combustion chamber to be cold, leading to insufficient evaporation of fuel, there is formation of a significant fuel film on the combustion chamber walls, causing more emissions and more fuel consumption.

In recent years, optical access to the engine and other laser diagnostic techniques have enabled researchers to visualize the process of mixture preparation.

The most common method of injecting the fuel to date is to inject it in the inlet port. The fuel is injected in such a way that the spray of fuel is aimed at the back of the inlet valve head. This is done in order to get the most efficient vaporization of fuel because the valve head is the area or component, which is normally the hottest in the inlet port, but is a problem when an engine is started cold. To meet the requirement for vaporized fuel, enough to form a combustible mixture in the combustion chamber, richer fuel is supplied at the start of the engine. Figure 2.1 represents an overview of the engine behavior during start up and figure 2.2 shows a schematic diagram of the factors causing excess fuel consumption in a cold start engine. Further discussion on the effects of mixture preparation during cold start, the mechanism of fuel transportation and backflow of gases during the valve overlap period has been included in appendix A.

\(^1\) In carbureted engine, initial fuel flow rates of six times that of the warmed up engines have been reported by [19].
Figure 2.1: Overview of an Engine Behavior during Start-up from Observations of Raw Data sets [7].
2.3 Spark Retard and Cyclic Variability

The major factors which affect mixture preparation are cycle to cycle variations in the air fuel ratio and the spark timing, consequently affecting fuel consumption and emissions during cold start. Spark retard is introduced as a cold start strategy to heat up the catalyst faster. With the ignition and hence, combustion occurring late, the work done on the piston during the expansion stroke is less and the combustion may even continue when the exhaust valve opens. This raises the blow-down gas temperature and elevates the temperature of the exhaust port and exhaust manifold, which helps the catalyst heat up faster and reach light off temperature more quickly during cold start engine operation. However, spark retard introduces some cyclic variations and a small fuel economy penalty. Further
discussion on spark retard and improving cyclic variations through swirl and tumble has been covered in the Appendix A.

2.4 Engine Friction

2.4.1 Introduction

Engine friction is the difference between the amount of energy delivered to the piston as a result of fuel combustion and the amount of energy available for the transmission input shaft.

Engine friction can be divided into three main categories:

- Mechanical losses,
- Pumping losses and
- Auxiliary components losses.

Mechanical losses are the losses that occur due to the movement of mechanical components across each other, regardless of the fact that they are in close contact with each other. In most of the engine running parts, the surfaces are separated by a film of oil, either as a hydrodynamic film or boundary lubricant.

Pumping losses are the work done, when the mixture of fuel and air and exhaust are induced into and forced out of the cylinder, through the inlet and exhaust valves respectively.

Auxiliary losses are the losses that occur due to the running of other accessories like the water pump, oil pump, etc.

Wakuri et al [13] studied the characteristics of total frictional loss, the friction of piston assembly and the friction of cam. The authors calculated the value of the total friction mean effective pressure on a single cylinder engine with four piston rings and two valves. However, the experiment measured the net power losses due to the pumping losses of gas exchange, the mechanical frictional losses of piston, bearings, transmission and valve train system and the power consumed to run the auxiliary. However, on the indicator diagram, it has been found that the gas pumping
losses change with speed and load, and so any generalization will be an approximation. The percentage of the power loss is broken up as:

Table 2.1: Percentage of Power Losses [13].

<table>
<thead>
<tr>
<th>Component</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas pumping loss</td>
<td>10 to 20%</td>
</tr>
<tr>
<td>Piston assembly</td>
<td>30 to 40%</td>
</tr>
<tr>
<td>Cams and followers</td>
<td>5 to 10%</td>
</tr>
<tr>
<td>Other auxiliary</td>
<td>35 to 50%</td>
</tr>
</tbody>
</table>

2.4.2 Effects of Friction on Engine Warm-up

An engine friction model, discussed by Sandoval and Heywood [16], gives an estimation of individual friction components and total spark ignition engine mean effective pressure. The total frictional MEP is predicted as a function of temperature and the temperature used is the coolant temperature. In this study [16], the total mechanical friction at 20°CA is compared to friction at 90°CA on the firing stroke and it has been found that the cold engine friction power loss is approximately 2.1 times higher than the hot engine friction power loss. The oil viscosity rapidly reduces with the increase in temperature. Validated predictions of total engine friction and the comparison between cold and hot engine friction indicates that friction inside the engine can be greatly reduced if a faster warm-up of the engine is achieved.

However, a major part of the frictional heat is used to heat up the oil and other auxiliaries. The effects of heat transfer and frictional dissipation on the oil warm-up process can be understood from measurements done by Shayler et al and Gyan et al [28-30]. Lechner et al [31] further compare the different grades of oils and discusses their effects on friction and engine out emissions during cold start. The effect of frictional dissipation is a dominant factor in governing the oil warm-up process under most operating conditions, mainly during cold start. The authors used the Ford 1.1L, four
cylinder Valencia engine mounted on an eddy current dynamometer and the study shows that out of the total power produced, a large proportion is used up in pumping and heating the oil and the remaining part is used to heat up the auxiliary part such as coolant pump and alternator. The authors argue that at a given engine speed, the average value of heat transfer to the oil increases when the total heat transfer increases. The results show that the ratio of the total heat transfer (calculated from Taylor and Toong correlation) to the heat transferred to the oil, increases approximately by a factor of 2, when frictional heat is considered and taken into account. The results also show that the frictional heat increases approximately with the engine speed.

Harigaya et al [32], used the unsteady state 2-D energy equation considering viscous heating to analyze the heat transfer in an oil film between the piston ring and cylinder liner and to predict the viscosity and oil film thickness. It has been noted in this study that the temperature distributions in the oil film are affected by the viscosity, the wall velocity and the oil film thickness and also the ring and liner surface temperature. The temperature of the oil film increases with increase in engine speed and the viscosity decreases due to viscous energy dissipations.

### 2.4.3 Calculation of Engine Friction

There are two experimental techniques to determine the total frictional power:

- The Morse test.
- Measurement of indicated and brake power.

The Morse test is an experimental method used to measure engine friction. The spark plug to one of the cylinders is disconnected and the loss in the brake power or fuel supply is related to the frictional power.

Another method is to use a pressure transducer to measure the cyclic pressure fluctuations inside the cylinders. This gives a pressure Vs time (P-T) or directly a pressure Vs volume (P-V) diagram, from which indicated
work/cycle and hence power can be calculated. The difference between indicated and brake power is frictional power. Ghazakhini et al [4] showed this in their study. Their results show that the water and oil temperature increases as a function of time from start-up. The oil heat-up rate is slower than for the water and shows an initial dead period, followed by a temperature rise, which is linear with the engine speed and does not depend much upon engine load, which indicates that the lubricating oil heat-up depends more on the frictional effects and not on the heat rejected from the engine. Figure 2.3 shows the results for the rise in temperature of oil and water with time.

Figure 2.3: The variation of Lubricating Oil and Water Temperatures for Ford CVH Engine with a constant Throttle Step Warm-up [4].
The authors also discussed results, which show that there is little variation in the peak cylinder pressure and the rise is gradual for fixed speed and throttle, during the warm-up process, when the observed increase was approximately 2 bar.

The authors further explain that initial IMEP during warm-up is higher than the steady state and there are no abnormal cycle to cycle variations during warm-up. This shows that mixture preparation has little impact on cycle to cycle variations because during warm-up, the inlet manifold is not warm enough and the mixture preparation is not good. This is because the coolant flowing through the inlet manifold is cold and it does not heat up the air flowing into it. The indicated power and the brake power are calculated and the difference between the two is the frictional power. The results show that there is a very sharp reduction in the frictional power with time. This study has also shown the results of the Morse test as a function of lubricating oil temperature. The Morse test also shows a reduction in frictional power as the temperature of the lubricating oil rises. Figure 2.4 shows that the frictional power losses come down with time.
2.4.4 Different ways to Reduce Friction

Some ways of reducing friction have been discussed by Insight-Central [33]. They are:

- Offset Cylinder Design: This is a design change in which the cylinder block has an offset design which means the bore centre has an offset 14 mm from the crank centre. Since the maximum combustion pressure occurs close to when the connecting rod and crank are straight up and down in the cylinder, the friction and piston slap are both reduced by having a non-vertical crank pin position at peak cylinder pressure.

Figure 2.4: Frictional Power Loss as a function of the Lubricating Oil Sump Temperature during a Step Warm-up at constant Throttle [4].
• Process of Shot Peening: In order to reduce friction, Honda engineers have specified a new lightweight aluminium alloy piston with minimum skirt area and the surface of the skirt has been shot peened. Shot peening is a process in which a metal part, such as a piston or connecting rod, is blasted with shot-like particles, creating uniform, microscopic dimples on the surface. This dimpled surface is better able to retain a lubricating oil film. Shot-peening the IMA engine's pistons accounts for another 1.5- to 2.0-percent reduction in internal friction.

• Electric Power Steering: Unlike the hydraulic power steering, the electric power steering draws electric power and no extra energy is needed while cruising, which reduces the load on the engine and saves fuel consumption.

There are many other examples of design changes, which reduce sliding forces and rubbing areas and other examples of reducing auxiliary power demands are beyond the scope of this thesis.

2.5 The Engine Heat Transfer Problem

2.5.1 Introduction
The peak temperature inside the combustion chamber is of the order of 2500°C, but taking into consideration the limits of maximum sustainable temperature of the metal components, this has to be kept as low as 300°C for aluminium and 400°C for cast iron (e.g. piston crown). The cylinder wall, which is in contact with the hot burning gases, should not go beyond 180°C to avoid deterioration of the oil film in contact with the piston. In order to achieve the required cooling of all metal components, heat is transferred from the piston, cylinder head and combustion chamber wall.
These conditions lead to generation of high heat fluxes to the combustion chamber walls and these heat fluxes can go up to a limit of 10MW/m\(^2\) during the combustion process, as explained by Heywood [34]. Moreover, the spark plugs and the valves must be cooled to prevent knocking, because overheating of the spark plug electrodes and the exhaust valves lead to knocking.

The heat transfer takes place under conditions of varying temperature, pressure and local velocities. These local velocities vary more or less depending on the inlet port and combustion chamber configuration. The surface area of the combustion chamber varies through the cycle and the heat flux into the combustion chamber walls starts from a small negative value during the intake process and go as high as an order of several MW/m\(^2\) during the combustion process and in the early expansion process. The heat transfer process occurs from the hot gases to the combustion chamber wall and then from the wall to the coolant. The gas heat flux into the wall has both convective and radiative terms. Heat conducted through the wall is then convected from the wall by the coolant. The heat flux distribution over the combustion chamber wall is highly non-uniform. Heat dissipation in the engine as explained by Heywood [34] is as follows: A large part is dissipated through the piston and piston rings to the cylinder walls and transferred to the cooling medium as thermal energy. The remainder is dissipated through the cylinder head and cylinder walls whilst a considerable component is transferred through the exhaust valve and port. Smaller amounts are transferred to the bearings, valves mechanism and other moving auxiliaries and as thermal energy to the oil or surrounding environment. Thus the heat carried by the coolant consists of heat transferred from the gases to the cylinder, heat transfer to the exhaust valve and port and a fraction of the friction work. With cold surfaces, the heat transfer can be double the value as when hot.
2.5.2 Effects on Fuel Economy and Engine Warm-up time

It becomes important to study the various heat transfer processes occurring in the engine to improve the cold start fuel consumption and reduce the warm-up time. There has been a thorough study about the areas, modes and amount of heat transferred in the engine when an engine is run under transient cold start conditions and is discussed in an investigation done by Trapy and Damiral [5] on a Renault F2N engine, fuelled with propane and mounted on a test bed coupled to Schenck W-150 dynamometer. The study shows that major part of heat generation in the lubricant is through friction in the bearings, piston assembly and other moving parts; i.e. about 61% and remaining 39% is generated through other sources like the combustion chamber and coolant exchanges. Analysis shows that oil is the main heat carrier inside the engine and it does not retain most of the heat generated from friction of bearings and other moving parts and the combustion chamber. A major part is lost to the ambient or goes to the combustion chamber walls, oil filter and oil sump and it has been found in this study that only 40% of the heat transferred to the oil is used for engine warm-up. Figure 2.5 shows the energy break-up from the authors' analysis.

Figure 2.5: Heat transported by the Lubricant During Engine Warm-up [5].
Trapy and Damiral [5] showed that the engine speed and load have significant effect on oil and coolant warm-up. The experimental results show that there is an evident disparity between the oil and coolant thermal responses. Figure 2.6 and figure 2.7 explain these responses. The author argues that the coolant reaches the peak operating temperature in 7 minutes while the oil reaches the same temperature 4 minutes later for a specified running condition. Nonetheless, lubricant is the last component of the engine to get warmed up and the coolant heats up much faster than the oil.

Figure 2.6: Lubricant and Coolant Temperatures during Engine Warm-up [5].
2.5.3 Factors affecting Engine Heat Transfer

There are many factors that affect the heat transfer in an SI engine and consequently affecting the performance, fuel consumption, emissions, drivability and the life of engine components at large. It has been discussed by many authors such as [11, 34-44].

In an ideal theoretical IC engine cycle, heat transfer is neither necessary nor desirable. It ideally includes an isentropic compression, isochoric combustion, isentropic expansion and gas exchange. However, in practical IC engine processes, this heat transfer cannot be avoided and infact, it plays a very important role in designing the engine [38, 45]. The cyclic nature of the IC engine hinder efforts towards building an adiabatic engine and the ideal situation of an adiabatic engine is technically unachievable. Thus heat transfer plays a very important role in influencing an engine’s performance. It is also a significant issue when designing the engine in regards to its operational performance and durability. Heat transfer also becomes important to design the cooling system of the engine.

Gilaber et al [36] measured the heat flux at different positions in the engine and then validated a multidimensional model against the experimental data.

Figure 2.7: Influence of Engine Speed on Oil Warm-up [5].
Some of the factors influencing engine heat transfer that [34, 36, 37] discussed are:

- **Change in Heat Flux due to Variation in Spark Advance**: When the spark timing is retarded, spark occurs late. Subsequently, combustion occurs late and the burned gas temperature is reduced. Therefore, there is a decrease in the combustion chamber heat flux when the spark is retarded. However, the exhaust side burned gas temperature is increased, as the combustion is slower during the expansion stroke and even incomplete when the exhaust valve opens, and there is an increase in temperature of both the exhaust valve and port. Gilaber et al [36] states that within a range of spark advance from $0^\circ$ to $40^\circ$ BTDC, results showed a reduction of flame velocity, as the spark is retarded, and this is due to the decreasing pressure conditions when the flame reaches its fully developed size. The heat flux measured at certain location has a peak value at a spark advance of $40^\circ$ and when the spark is retarded from $40^\circ$ to $0^\circ$ BTDC, the peak value decreases from $22000 W/m^2$ to $12000 W/m^2$.

- **Change in Heat Flux due to Variation in Volumetric Efficiency**: It has been observed that highest volumetric efficiencies, corresponding to highest cylinder pressure conditions, provide the highest heat release rates. Alkidas and Myers [35] found that an increase in volumetric efficiency of about 40-60% resulted in an increase in heat flux of around 30%.

- **Change in Heat Flux due to Variation in Engine Speed**: The average heat transfer per unit time increases with engine speed, load and A/F ratio. The heat flux is found to be maximum at stoichiometric or near stoichiometric and making the mixture leaner or richer to any extent results in a decrease in heat flux [34]. Gilaber et al [36] showed that
when the speed is increased from 500 rev/min to 2500 rev/min, the heat flux increases from $2000 \text{ kW/m}^2$ to $3100 \text{ kW/m}^2$.

- Change in Heat Flux due to Variation in Equivalence Ratio: Gilaber et al [36] also showed that when the equivalence ratio is increased from 0.7 to 0.9, the heat flux increases from $800 \text{ kW/m}^2$ to $2200 \text{ kW/m}^2$ and then becomes a maximum of $2400 \text{ kW/m}^2$ at an equivalence ratio of 1.0 or 1.1. On the contrary, when the equivalence ratio is decreased from 1.0 to 0.7, the heat released is slowed down and the burnt gas temperature decreases because of dilution.

The magnitude of heat flux depending on the engine variables has been further discussed by [34, 37]. Some other factors are:

- Compression Ratio: As discussed by Heywood [34], increasing the compression ratio decreases the total heat flux to the coolant until $r_c \approx 10$, thereafter, heat flux increases slightly as $r_c$ increases.

- Swirl and Tumble and Mixture Inlet Temperature: Introducing swirl, tumble and squish motion improves the mixing process and lead to increased heat fluxes. Increasing the inlet temperature of the mixture helps in better evaporation of the injected fuel and consequently leads to higher heat flux.

- Coolant Temperature and Composition: Increasing liquid coolant temperature increases the temperature of the components cooled directly by the coolant. When the coolant temperature is raised, there is smaller response of the metal temperature to the coolant temperature change in regions of higher heat flux (e.g. regions where nucleate boiling occurs; i.e. exhaust valve bridge. The response is greater in the regions of lower heat flux, where heat
transfer is largely by forced convection (e.g. the cylinder liner). The composition of the coolant also has effects on the heat transfer because different compositions have different thermodynamic properties. For example, when ethylene glycol is added to water, it changes its thermodynamic properties. It raises the boiling point of mixture and this helps in the prevention of the occurrence of boiling. When nucleate boiling occurs, there is a sudden increase in the heat flux. Although, the bulk temperature of the coolant remains below the saturation temperature, but the metal temperature becomes almost independent of the coolant temperature and velocity of the coolant. However, if the film boiling occurs, the heat flux can reduce by two order of magnitude and the metal temperatures can rise significantly to cause damage.

- Wall Material: The most common cylinder wall materials are cast iron and aluminium and they both operate in the temperature zone of $200K – 400K$ [34], which is relatively low compared to the temperature of the hot gases inside the combustion chamber. Some wall materials that can operate at much higher temperatures and have lower thermal conductivity are silicon nitride and zirconia. When gas enters the chamber during intake stroke, the heat transfer takes place from the cylinder walls to the gas during the intake and compression strokes i.e. $T_w > T_g$. This heat transfer from the wall to the gas in the intake stroke decreases the volumetric efficiency and in the compression stroke, it increases the compression work. The heat transfer from the gas to the wall takes place during combustion and expansion strokes, when $T_g > T_w$.

- Knock: During knocking, the cylinder gas temperature and pressure are increased above the normal combustion levels. Knock results in increased local heat fluxes in the regions of the cylinder head, liner and piston in contact with the end-gas.
2.5.4 Combustion Heat Transfer

The basics of the combustion heat transfer are founded on the Nusselt number, Reynolds number and Prandtl number correlations found for turbulent flow in pipes and flow over flat plates.

\[ Nu = a \, Re_m \, Pr_n \]  \hspace{1cm} (2.1)

For calculating heat fluxes from these correlations, the velocity used in the Reynolds number calculation is to be determined either from the experimental data or calculated from the flow rate of the charge going into the combustion chamber, which can never be accurate. The other two variables are the gas temperature and pressure, at which the gas properties are evaluated to calculate the correlation numbers. These heat transfer correlations differ for different operating conditions depending upon the velocity of the gas used for Reynolds number calculations, the gas temperature and pressure at which the gas properties are evaluated.

2.5.5 Components Temperature Distribution

Normally, the greatest amount of heat is generated in the centre of the piston, centre of the cylinder head and the exhaust valve bridge region. It is lowest in the cylinder walls. Cast iron pistons run about 40°C to 80°C hotter than aluminium pistons. The highest temperatures occur when the heat flux is high and access for cooling is difficult. Such locations are regions between the exhaust valve and the adjacent cylinder and the bridge between the valves. The lower regions of the cylinder liner are exposed to the combustion products only for a small part of the cycle after significant gas expansion has occurred. The heat flux and temperature decrease significantly with distance from the cylinder head. Moreover, there is also a significant contribution to heat flux due to friction. Choi et al [46] investigated the cylinder head and piston temperatures and heat fluxes
on two different engines (SOHC$^2$ and DOHC$^3$ engines). These temperature measurements are done at various engine speeds and load conditions. In the experimental work, the authors have used twenty one instantaneous temperature probes totally embedded in the cylinder head and the piston. Because of the increased valve area, the charging efficiency of the DOHC engine is increased and the engine can achieve high performance. Their engine ran lean and low on emissions due to the tumble flow through the two intake valves. Moreover, due to the increased valve area, more energy is supplied to the combustion chamber and hence, it results in higher temperatures in the combustion chamber which results in higher stresses on the cylinder head, cylinder liner and the piston. These high temperatures especially occur at the exhaust valve bridge because of the complex design and least access of cooling. The results analyzed by the authors are at wide open throttle at an engine speed of 3000 rev/min. The thermal load also damages the piston because it reciprocates fast under high pressure, temperature and inertial load. Furthermore, the increased temperature in the ring groove and land causes the sticking and lubricant carbonation. The authors' primary focus was on the temperatures of exhaust valve bridge, piston crown and ring land areas. The probe located at the valve bridge measured the highest temperature. The temperatures decreased as the location is moved from the spark plug location to outer edge of the combustion chamber. The temperature of the spark plug probe was the highest in the SOHC engine, whereas in the DOHC engine, the exhaust valve bridge probe was the highest and remarkably higher than the SOHC one.

---

$^2$ SOHC is single overhead camshaft engine, and there is only one camshaft per header. Inline engines normally contain one camshaft. V-type and/or flat engines contain 2 camshafts. For a SOHC engine there are usually 2 valves per cylinder but there can be more with the addition of cams for each valve [47].

$^3$ DOHC is double overhead camshaft. In a DOHC engine, there are 2 camshafts per header. These DOHC engines usually have 4 valves, one camshaft for the exhaust valves and the other one for the intake valves [47].
The temperature swings increased, with the increase in engine load. This can be explained as more energy is supplied to the engine with increased engine load as compared to the shortened combustion duration with increased engine speed and it affects the combustion temperature to a great extent.

The temperature of the exhaust valve increased with the increase in engine speed and load. The temperature of the exhaust valve is higher than the temperature of the inlet valve because the exhaust valve is interacting with the hot combustion products while the inlet valve is cooled through its interaction with the fresh charge. The irregularity and non-uniformity in the heat flux is also discussed by Cipolla [48].

2.6 The Role of Coolant in a Cold Start Engine

Automotive engine coolant technology began in 1885, when Karl Benz invented the first automotive radiator and the application of ethylene glycol was first suggested in 1916 in England for high performance military aircraft engines, as discussed by Beal [49].

2.6.1 Phases of Engine Cooling

There are four phases of engine cooling which have been discussed by Porot et al [50]. They are:

- The local convective cooling, which occurs in the engine under normal conditions.
- Local nucleate boiling.
- Stable film boiling.
- Unstable and unsafe film boiling.

This study analyses all the cooling regimes with respect to the coolant flow rate. Under normal conditions, the heat transfer occurs through forced convection. However, when the metal surface temperature increases or the coolant flow is reduced, it leads to boiling or nucleate boiling. Nucleate boiling starts to occur, after the local convective heat transfer is insufficient
to suppress the local boiling and leads to the entrainment of bubbles in the coolant flow, which is indeed profitable. However, when the flow is further reduced to take advantage of boiling heat transfer, a thin film of vapor starts to develop, which reduces the heat transfer and can be a dangerous and unsafe regime. Nucleate boiling is also discussed by Porot et al [50], where the authors also talk about geometric modifications in the engine cooling circuit and other improvements like balancing the flow around every cylinder.

The main advantage of boiling heat transfer is the power-exchange rate, which is higher than the normal convective heat transfer. This implies that one can get a higher heat transfer at the same mass flow rate of the coolant. However, boiling heat transfer can only occur at a higher temperature than the convective heat transfer, which can obviously lead to failure or wear of metal parts of the engine at very high temperatures. Significantly, best utility of boiling heat transfer is only possible at certain running points of high loads and high speeds, where maximum heat release occurs.

### 2.6.2 Effects of Engine Speed on Coolant Temperature

As the engine speed is increased, heat fluxes inside the engine increase, which increases the heat transfer to the coolant. Coolant pressure also plays an important role in heat transfer to the coolant. As the coolant pressure is decreased, the flow rate of the coolant drops and consequently it increases the heat transfer to the coolant. However, decreasing the coolant below a certain threshold leads to cavitation⁴, which may further lead to unstable film boiling. This has been shown by Lee et al [51] in experiments performed on a 1.6L, four cylinder engine. Lee et al [51] used 10 thermocouples on the exhaust valve bridge position, where heat flux is highest. These tests were performed on various engine loads and two different coolant pressures. Both the inlet and outlet coolant temperature

---

⁴ Cavitation is normally boiling because of low pressure without heat transfer as in pump blade passages.
and pressure were monitored with the help of thermocouples and two pressure transducers respectively.

The results in this study show that the temperature in the exhaust valve bridge increases proportionally to the engine speed and as we move closer to the spark plug, the temperature tends to increase.

The results show linear slopes, which indicate pure convection and no nucleate boiling at 3000 rev/min. The effect of coolant pressure drop is also negligible at this speed range, but the effect is noticed at the speed of 5600 rev/min, when the coolant flow rate drops drastically with the decrease in pressure. Therefore, the boiling phenomena in the coolant only occurs at a speed of 5600 rev/min at WOT and not below it.

Although, the heat rate is highly time dependent inside the cylinder, but it is very steady on the coolant side. The boiling temperature of the coolant, which tends to be fairly uniform throughout the engine block, effectively controls the surface temperature. If the engine is desired to operate at higher temperature, then the boiling point of the coolant must be increased. Either using a coolant with a higher boiling point or pressurizing the coolant or both can do this.

There have been substantial efforts to investigate the heat from the combustion chamber to the cylinder walls and from the walls to the coolant. The heat transfer from the combustion chamber to the cylinder walls is highly complex because it varies greatly in different parts of the cycle and in different parts of the cylinder. For example the maximum amount of heat is transferred in or near the exhaust port during the exhaust stroke. Due to the complex geometry involved, it is highly unreliable to depend on the empirical correlations to make the heat transfer calculations correctly and precisely.

The maximum surface temperature is mainly controlled by the boiling temperature of the coolant. Boiling occurs at the surface and the vapor condenses as the vapor bubbles mix with cooler water, so there is no net boiling. Most of the heat transfer occurs through this mechanism, at higher loads. Since the surface temperature is controlled by the boiling point of the
fluid, this boiling point has to be increased if we want to increase the surface temperature, which can be done either by pressurizing the liquid or adding something to the coolant which has a higher boiling point. Having such discussion about the boiling of the coolant at the surface, it becomes important to provide more coolant flow near the ‘hot spots’ to carry away the heat. Although we can have more heat transfer when the coolant boils in regions of these hot spots, but it can lead to formation of vapor pockets, so boiling of the coolant is indeed helpful but it is not desired.

2.6.3 Effectiveness of the Radiator

Kays [52] describes the usage of various after-coolers and superchargers in the cooling loop of the engine. The heat load in the combustion chamber temperature has to be maintained at around $400^\circ K$, so a common rule of thumb that applies is that the necessary cooling rate is about equal to the mechanical power output of the engine, but studies show that a typical heat for full load is 0.7 times the power and a larger factor for part load.

The ratio of the actual heat transfer rate to the maximum possible heat transfer rate is termed as “Effectiveness” of a heat exchanger.

$$E_{ff} = \frac{q}{q_{max}} \quad (2.2)$$

$$E_{ff} = C_c (T_{c, out} - T_{c, in})/C_{min} (T_{h, in} - T_{c, in}) \quad (2.3)$$

$$E_{ff} = C_h (T_{h, in} - T_{h, out})/C_{min} (T_{h, in} - T_{c, in}) \quad (2.4)$$

Where $q_{max} = C_{min} (T_{h, in} - T_{c, in})$, the lesser value of $C_c$ or $C_h$ is treated as $C_{min}$ and the other as $C_{max}$.

$$q = C_c (T_{c, out} - T_{c, in}) = C_h (T_{h, in} - T_{h, out}) \quad (2.5)$$

$C_c$ & $C_h$ are the capacity rates for cold and hot fluid respectively.

$C_c = M_c C_p$ for colder fluid and $C_h = M_h C_p$ for hotter fluid.

The effectiveness of a typical automotive radiator ranges from 0.2 to 0.5 at full load. To get a better heat transfer rate from a heat exchanger, compact heat exchanger arrangements can be used. Compact heat exchanger
surfaces are arrangements for which the surface area per unit of the heat exchanger volume is relatively large. Large surface area arrangements can be obtained by either using small diameter tubes closely packed or by using extended surfaces “fins”. As the heat transfer coefficient in an automotive radiator is fairly large on the coolant side as compared to the air side, the fins are only employed on the air side.

Muto et al [53], reports effective prevention of conduction from the radiator to the air conditioning condenser by combining the two units into one, developing a new cooling module.

### 2.6.4 Parameters affecting Engine Cooling

There are four parameters in an engine cooling system directly affecting the heat transfer rate. They are:

- Coolant temperature,
- Coolant flow rate,
- Coolant properties and
- Coolant system pressure.

Analysis shows that when the temperature of the coolant rises, nucleate cooling starts to occur after local convective cooling. Nucleate boiling relates to the entrainment of bubbles in the coolant flow. To increase the heat transfer coefficient by reducing the coolant pressure at WOT to promote nucleate boiling may result in film boiling which reduces the heat transfer and may sometimes completely stop the heat transfer as discussed earlier.

Furthermore, low system pressure may also lead to pump erosion. Coolant properties are also difficult to adjust and therefore coolant temperature and coolant flow rate are the only two parameters that can be changed to get a better heat transfer rate. This has been very well discussed by Yang [54]. The author realized the importance of reducing the flow of coolant during engine’s start-up to achieve a faster warm-up because there is a need for the engine to be thermally isolated rather than to be cooled. The strategy
discussed in this research is to implement a throttle at the pump’s outlet and this throttle can be a function of the engine performance. It can be actuated by the engine management system, so that when the engine load increases at WOT, the coolant flow can be increased but under part load conditions, it can be restricted and consequently, this reduced flow can bring a faster warm-up.

An electric water pump is an excellent alternative to vary or reduce the coolant flow rate and its importance together with an electric thermostat has been praised by Yang [54], but the author tried to avoid these solutions because of the cost involved in their application. With an electric pump, some sensors, flow meters, actuators and electric motors would be involved. There is no doubt that the cost of these items was high at the time when this work was carried out, but it has come down to a considerable amount in recent years as the electronic market has seen new developments and horizons. As for the power consumed by the actuators, motors and sensors, it can be compensated by the reduction in power consumption with the use of an electric water pump because under part load conditions the mechanical water pump consumes more power than is needed as its power use is only speed dependent.

2.7 Faster Warm-up Strategies

The current section covers faster warm-up strategies related to engine heat transfer management and engine coolant system. However, further discussion on faster warm-up, related to the technological developments is included in Appendix B and the reader is advised to refer to it for more details on the topic.

2.7.1 Coolant Flow Rate and Thermal Capacity of the Engine

A rapid warm-up of the engine is very critical in attaining low fuel consumption and emissions because the fuel consumption and engine out emissions are highest during cold start and improve as the engine warms-
up. Typically in the engine considered here in the 'cold' first 505 seconds of the US FTP city test (ADR37/02) about 16% more fuel is used than in the repeat 'hot' start first 505 seconds, as discussed by Watson [55]. The demand for cabin heater to improve the cabin conditions and to perform demisting and deicing of the windscreen further prolongs the engine warm-up. The engine's thermal management system should provide an optimum solution by maintaining a balance between engine warm-up, cabin heating, catalyst light off and emission performance. The engine management system controls the heat distribution in the engine and in the vehicle by compensating engine controls such as spark timing and air-fuel ratio to regulate the engine power output as well as heat production and distribution to each part of the engine.

The heat rejection problem, which is more prominent at wide open throttle, is tackled by optimizing the coolant gallery design for optimum heat transfer effectiveness by targeting this region with high coolant velocities. At the same time, there are hydraulic losses at part load conditions, because of the mechanical water pump (engine driven pump), which supplies more than the required coolant flow in the system. If critical areas such as exhaust valve-bridge are subjected to high coolant flow velocities, better heat transfer, without steep temperature gradients and higher heat flux can be achieved. The idea is to decrease the temperature around these areas and this can be achieved by decreasing the cross sectional area of the coolant passage in these regions to attain higher coolant velocity without high bulk flow rates. The design of such a cooling system involves sizing of the coolant gallery and selection of the coolant pump to ensure that the heat removal rate of the system can satisfy the constraints of an engine operating temperature in these vulnerable regions even at low speed at WOT conditions. Use of finer coolant galleries brings the flow rate down and increases the coolant velocity in those areas. Pang and Brace [56] attained a flow rate of coolant of 4m/s from 1.4m/s, with the use of finer coolant galleries and consequently this gain led to the
increase of heat transfer in the cylinder head and bringing the head temperature down by up to $60^\circ C$.

Reducing the coolant volume in the cylinder head and engine block in order to decrease the overall warm-up time of the engine, Clough [57] designed a modified water jacket. In the engine block, the author not only reduced the thermal mass of the cylinder liner by about 30% by using aluminium only (no cast iron), but also the length of the water jacket, reducing it to 40% of the piston stroke. In the design of this new cooling jacket, the total volume of the coolant was reduced by 37%.

Clough [57] reported a decrease in warm-up time by 18%, when precision cooling system was used as compared to the conventional cooling system. Metal temperatures have been reported to come down by $80^\circ C$. Also the cylinder block temperature was reported to be in the range of 105$^\circ C$ – 143$^\circ C$ and the temperature distribution was also considerably less as compared to the standard engine. Bulk coolant flow was also reported to be 35% as compared to the standard engine. For all these data, the engine was running at 5000 rev/min and WOT.

Furthermore, Clough [57] also discussed two different types of flow pattern or flow directions; transverse and longitudinal. Transverse cooling is the cooling in which the coolant typically exits the cylinder head from a single location, normally at the front of the engine. So, the cylinder nearest to this exit point will experience more flow of the coolant than the other cylinders. The other flow direction is longitudinal, in which there is equal distribution of coolant in every cylinder. Nonetheless, there are problems associated with keeping the same flow rate around every cylinder, firstly, the variation of temperature between the entry and exit points and secondly, the pressure drop in the coolant flow. However, the high pressure drop can be reduced by reducing a high bulk flow rate.
2.7.2 Advanced Cooling System

The engine cooling system plays a vital role in dictating the performance of the engine. With increasingly compact engine design and higher specific power, the heat fluxes have increased significantly. Removing heat from an increasingly restricted space is a particular concern at vulnerable regions, such as the exhaust valve bridge area. There is a risk of complete failure in these regions, even when there is a minor fault or major failure of the cooling system.

Advanced cooling system refers to the changes and innovations in the conventional cooling system considering the limitations of the cooling system, when the engine is running in transient conditions, especially during cold start. Unlike the conventional cooling system, the advanced cooling system not only helps in retaining the heat during engine cold start but also protects the engine from excessive heat during extreme running conditions of part load as well as in typical city driving. Consequently, it serves the purpose of improving engine fuel economy and emissions. This goal can be achieved by maintaining a fine balance between the fuel efficiency, emissions and the factors stated below:

- Frictional losses within the engine.
- Auxiliary power required to operate the cooling system.
- Combustion system boundary conditions like combustion chamber temperature, charge density and charge temperature.

There are so many constraints when improving the engine’s performance and it is absolutely not possible to improve all aspects of an engine’s performance by just manipulating one variable, which is the engine coolant system. Therefore, a number of features are being incorporated into the advanced cooling system to meet most of the constraints of the engine performance. These features increase the operating flexibility of the cooling system. The changes are neither purely hardware nor purely software but they exist in combination to compliment each other’s effects [56].
Moreover, in the advancement and development of cooling system, Yoo [58] developed a coolant temperature model designed for the diagnosis of the engine cooling system faults.

2.7.3 Controlling Coolant/Metal Temperature

The most common approach for handling the conventional engine coolant system was to control the coolant temperature. The coolant temperature influences the engine metal temperature, but this relationship only holds true for steady state condition, due to the complexity of the actual temperature distribution in the engine structure. Consequently, there are significant variations in the metal and coolant temperatures from one point to another giving a non-uniform temperature distribution.

As the operating limit of the engine depends on the peak operating temperature of the vulnerable regions such as the exhaust valve bridge or the piston crown, because these are the areas where the temperatures are highest, it is more desirable to design a cooling system which is based on metal temperature rather than coolant temperature. As the cooling system is designed to sustain peak heat rejection rate at WOT, which is based on the concept of controlling the coolant temperature and not the metal temperature, the engine and its cooling system does not perform ideally under part load conditions, which are mainly city driving conditions and slow cruising. Hence, the engine may become less efficient in this scenario leading to higher fuel consumption.

This is the area where an improvement can be done by varying the set coolant temperature to improve the engine’s performance at part load conditions. If the peak operating temperature of the exhaust valve bridge area is known, the coolant or metal temperature can be shifted upward or downward to improve the performance of the engine and the cooling system.

Finlay et al [59] reported benefits in knock protection by lowering the inlet temperature to the cylinder head. Lowering the coolant operating
temperature has benefits in improving the volumetric efficiency of the engine and also in lowering the charge temperature. With varying the coolant or metal temperature, the effects on engine out emissions have been included in Appendix B and the reader is advised to refer to the Appendix for further explanations.

2.7.4 Controlling the Engine Operating Temperature

Raising the operating temperature of the engine has clear benefits because it is directly related to minimizing the losses in the engine, the effectiveness of the cooling system and the formations of emissions in the engine. The increase in the operating temperature of the engine raises the engine oil temperature, which reduces the frictional losses and improves the fuel efficiency of the engine. Finlay et al [59] argues that when the block temperature is increased to about $150^\circ C$, there was a decrease in fuel consumption by about 4-6%. Increasing the operating temperature of the engine has an effective influence on the engine. It improves the effectiveness of the heat transfer process in the engine and the radiator by allowing a lower flow rate for a given power and thus lowering the pumping requirement. Consequently, it lowers the power consumption by the auxiliaries, particularly at part load conditions.

2.7.5 Split Cooling System

In a split cooling system, the cylinder head and the engine block are cooled as two independent cooling circuits, so that the two sections of the engine can now have their independent optimum temperature set point, which maximizes the overall effect of the cooling system and engine performance. A desired temperature set point can be obtained by defining the flow rate of the coolant in either of the two sections, according to the relative cooling needs. A strategy reported by Pang and Brace [56] is to run the head cooler while having the block running warmer relative to the standard condition. By running the head cooler, volumetric efficiency improves, increasing the
mass of trapped air even at lower temperature. The increase in the mass of trapped air at lower temperature allows a more rapid and complete combustion, reducing the amount of NOx, CO and HC and increasing the output power at the same time. In an another study Finlay et al [60] also investigated the effects on BSFC, emissions, detonation – borderline ignition timing by running a dual cooling system, with the head and block running at different temperatures. Running the block warmer than the head reduces frictional losses, especially in the piston, subsequently, heating up the oil faster and contributing to the fuel efficiency improvement. It indirectly lowers the peak in-cylinder pressure and temperature, which has strong association with NOx formation. As also reported in the literature by Finlay et al [59] that when the temperature of the head is brought down to \(50^\circ C\) and the temperature of the block is raised to \(150^\circ C\), there are reduced frictional losses, which subsequently provide reduction in fuel consumption of about 4-6% and a reduction in unburned HC to about 25-35%.

2.7.6 Electric Water Pump
Whenever an engine starts, the mechanically driven water pump starts to work with the movement of the crankshaft and it is a function of engine speed. The greater is the engine speed, the faster the pump runs and significantly, greater is the flow rate. A higher flow rate of coolant is not desirable, when the engine is cold because the engine does not need to be cooled but thermally isolated. Controlling the coolant flow rate, when the engine is cold can be beneficial in all respects. The above outline describes a clear advantage of an electric water pump over the conventional mechanical pump. Thousands of miles of mixed road driving with normal passenger load and chassis dynamometer data has been collected and in the light of these results, the performance of an electric water pump is compared to a mechanically driven pump by [6, 61-64].
Brace et al [6] reports through the simulation results that the electric water pump worked well under this cooling system and control strategy. The pump increased its speed for short periods whenever it was required, in times of extreme hill climbs or extended idle in hot conditions or on “heat soak” on motorway driving. The simulation results show good signs of fuel economy, except for a few drawbacks in the system. Firstly, the system pressure was not always as high as desired, which definitely leads to bigger problems of cavitation. Secondly, the thermostat responded slowly and under certain open conditions, the heater circuit was not receiving sufficient flow. The simulation results of the system also show a considerable reduction in pump power consumption and the increased coolant temperatures lead to faster warm-up and fuel economy benefits. Figure 2.8 explains the difference between the warm-up times for an electric water pump, mechanical water pump and electric water pump with no heater in demand.

![Figure 2.8](image.png)

Figure 2.8: Warm-up times for Mechanical Water Pump, Electric Water Pump and Electric Water Pump with No Heater Demand [6].

Table 2.2 shows the comparison between electric water pump and mechanical water pump [6].
### Electric Water Pump with a Flow Diverter Valve

- Flow is Independent of Engine Speed
- Diverter valve can prevent sudden rushes of cold water from the radiator
- Smooth Radiator Warm-up Reduces Thermal Shocks
- Provides a Precise engine Outlet Temperature
- Protects Engine, avoiding Heat Soak
- Reduces Noise Nuisance
- Maintains Cabin Temperature even when the Engine is Stopped, as the valve keeps the hot coolant in the Heater
- Accelerate Engine Warm-up due to Reduced Flow

### Mechanical Water Pump with a Conventional Thermostat

- Flow Depends on Engine Speed
- There is a sudden rush of cold water
- There are more Thermal Shocks due to Mixing of Hot and Cold Water
- Engine Outlet Temperature depends on the flow.
- Engine is not protected against Heat Soak
- Noise is there when Thermostat opens
- Does not Maintain Cabin Temperature when the Engine is Stopped
- More Heat Loss because of more Flow of the Pump

#### Table 2.2: Comparison between an Electrical Water Pump and a Mechanical Water Pump [6].

**2.7.7 Electrical Water Valve or Diverter Valve**

There is a clear need to optimize thermal management, to improve the cooling strategies, systems that could help reduce the fuel consumption and satisfy the Euro 4 and 5 standards. An improved and efficient cooling system is important because it not only brings a faster warm-up but also directly affects the emissions and fuel consumption.

The advantage of a new electrical water valve over the conventional wax thermostat has been discussed by [63, 64]. The results show a significant reduction in emissions and gain in fuel consumption by 2% to 5% with the use of this valve.
Chanfreau [64] argues that the most important property of this valve is its ability to completely stop the flow during engine warm-up, when no cooling is required. Since there is no waste of heat energy, the engine’s warm-up process is accelerated.

Working towards a better cooling system, Couetoux et al [63] developed a system, which is primarily concerned with an electronically controlled cooling system consisting of an electric water pump and a controlled valve. This system, unlike the conventional system was independent of the engine running conditions. The electric pump and control valve regulate and optimize the coolant flow rate, coolant temperature and air speed in all running conditions. With an electric pump, the system is no longer dependent on engine speed and reduction in heat transfer while warming up can be achieved to meet the challenge of the fuel economy penalty.

The literature proves that reducing the coolant flow rate does not affect engine’s reliability and the engine’s internal temperatures largely depend on coolant flow rate. Dugas et al [62] also showed that the rise in coolant temperature has effects on fuel consumption.

The results of the new control system discussed by Couetoux et al [63] are good in terms of fuel consumption and emissions but there was not much heating of the passenger compartment until about 20 minutes after the engine is started. The results evaluated at different driving conditions show a reduction in fuel consumption from 3% up to a maximum of 10% under different vehicle speed conditions. There has been a reduction of about 10% of hydrocarbons but $NO_x$ was increased by 10%.

Moreover, the reduction in fuel consumption is at the expense of electric power consumption, which makes the gain negligible. This new cooling system is a big step in the area of faster engine warm-up, but to make the system perfect in itself, the power consumed by the electronic circuits, actuators, sensors and valves etc. cannot be overlooked. Nonetheless, there exists an opportunity to obtain this power by regenerative electric braking using a more advanced alternator battery charging strategy.
2.8 Summary

The internal combustion engine heat transfer process is a very complex phenomenon due to the cyclic nature of the thermal source in the process. Design and control problems increase many times in magnitude, when the engine is operating under transient conditions, including cold starting conditions. The problem of insufficient fuel evaporation, inefficient mixing, and cold oil friction are a few of the major issues associated with cold starting. As discussed in the literature, there is a need to thermally isolate the engine rather than cooling it, when it is started.

The ideas of improvement on the engine warm-up concepts have been well established now, but the main point of concern is to develop a strategy to implement some or all these ideas of improvement collectively to get a positive outcome. If the full potential of an engine cooling system is to be realized, efforts have to be made towards lowering the engine coolant flow rate, through the application of an electric water pump and a flow diverter valve and exploring the merits and demerits of split cooling supply.

The following chapters detail the research objectives and the experimental strategy of achieving the goal of faster warm-up of the engine and of reducing the fuel consumption penalty.
Chapter 3 – Modeling of Engine Heat Transfer

3.1 Introduction

As discussed in previous chapters, internal combustion engines still have potential for substantial improvements, particularly with regard to fuel efficiency and environmental compatibility. To obtain a better understanding of the system, one of the most frequently used tools is modeling.

Modeling is a mathematical representation of a system, which has similar system responses but usually includes simplification of the system. The mathematical model can be employed as a problem solving approach, which not only reduces the system development time but also helps in developing a robust system, as discussed by Maria [65]. Modeling engine processes enables engineers to make design changes without incurring the cost of making a prototype. It not only saves times but also money and resources.

3.2 Approaches to Modeling

The model here is a set of differential or algebraic equations which introduces constraints among the variables that describe a system and
recent advances in the numerical analysis have given algorithms for integrating such equations.

In most instances, a model has to be calibrated and validated against experimental data for a range of data set points. The approaches to modeling can be divided into a hierarchy according to the level of complexity and hence, accuracy involved.

The very basic heat transfer modeling involves the dimensional analysis of the Nu, Re and Pr number, following a simple pipe flow analogy. Moving to the next step in this direction, there are empirical models and laws, which are based on experimental results. These models are partially physically based models, either joined together as lumped parameters or dispersed parameters, or CFD modeling. The level of complexity increases with the accuracy in the results and finally there are models, which do not require any experimental data for their validation. Such an approach in modeling is called Direct Numerical Simulation (DNS). There is no doubt that these models are more powerful and accurate than the previously described categories of models, as discussed by Jennings and Morel [66].

The main problems encountered during modeling heat transfer of an internal combustion engine, as described by Jennings and Morel [66] are:

- **Turbulence:** The flow fluid in all the engine processes is highly turbulent.
- **Flow unsteadiness:** All the flows, in the intake system, exhaust system and in-cylinder are highly unsteady.
- **Separated and reattaching boundaries:** There are some geometrical features of solid boundaries, like movement of the piston, which cause the piston roll-up vortex.
- **Combustion:** The flame is highly turbulent and produces high spatial and temporal variations in the temperature and species concentration.
3.3 Lumped Parameter Approach to Modeling

The group of components, in which the flow of heat is unsteady and keeps on switching between convection and conduction, resistor-capacitor analogy is the best suited strategy in such situations. Being the simplest approach to model the heat transfer in an engine, the lumped parameter approach is preferred over other approaches. A lumped parameter approach to modeling the thermal behaviour of an internal combustion engine is described in various previous studies [67-72].

Bohac et al [69] discussed a two-zone quasi-dimensional spark ignition engine simulation to determine in-cylinder gas temperature and convective coefficients, based on a resistor-capacitor thermal network model. Engine heat fluxes can subsequently be calculated from the engine’s specifications, dimensions, operating conditions and material.

Although, heat transfer occurs even during compression, when there is no combustion, but it increases with the onset of combustion, when the flame traverses the charge. The flame originates from the spark plug and travels across the combustion chamber up to the walls of the combustion chamber where it finally subsides or quenches. From the heat generated in the combustion chamber, it is important to accurately estimate the average effective gas temperature and resistances (or conductance) from the gas to the combustion chamber to get an R-C model working. However, the spatially varying temperature may be mass weighted to find a time varying average, and the walls lumped into several discreet sinks, such as piston, head, liner, etc., as will be explained later.

3.4 Fundamental Equations for Modeling the Engine Heat Transfer

Modeling the gas thermodynamics is a crucial part of the engine modeling, which is based on the fundamental equations of energy and mass transfer into the cylinder.

\[ \frac{dU}{dt} = \dot{Q} - \dot{W} \]  

(3.1)
\[
\frac{dM}{dt} = m_+ - m_-
\]  
(3.2)

A typical engine model includes one or several control volumes, which could include inlet and exhaust system with exchange of mass and energy between them. Flow through the valves is modeled as flow through an orifice.

As explained by Heywood [34], the mass flow rate through a poppet valve is usually described by the equation for compressible flow through a flow restriction. The equation is derived from a one dimensional isentropic flow analysis, and real gas flow effects are included by means of an experimentally determined discharge coefficient \( C_d \).

The air flow rate is related to the upstream stagnation pressure \( P_o \) and stagnation temperature \( T_o \), static pressure just downstream of the flow restriction (assumed to be equal to the pressure at the restriction, \( P_r \)) and a reference area \( A_R \), which is a characteristic of the valve design.

\[
m = \frac{C_d A_R P_o}{(RT_o)^{1/2}} \left( \frac{P_r}{P_o} \right)^{1/\gamma} \left[ \frac{2\gamma}{\gamma-1} - \frac{\gamma+1}{\gamma-1} \right]^{1/2}
\]  
(3.3)

When the flow is choked, i.e. \( \left( \frac{P_r}{P_o} \right) \leq [2/(\gamma + 1)]^{1/2} \), the appropriate equation is

\[
m = \frac{C_d A_R P_o}{(RT_o)^{1/2}} \left( \frac{2}{\gamma+1} \right)^{(\gamma+1)/(2(\gamma-1))}
\]  
(3.4)

For flow into the cylinder through an intake valve, \( P_o \) is the intake system pressure and \( P_r \) is the cylinder pressure. For flow out of a cylinder through an exhaust valve, \( P_o \) is the cylinder pressure and \( P_r \) is the exhaust system pressure.

\[ R = \frac{\hat{R}}{M} \], where \( \hat{R} = 8314.3 J/kmol K \), is the universal gas constant and \( M \) is the molecular weight of the gas. \( \gamma = 1.3 \) is the best value for the entire cycle for a stoichiometric mixture.
The product of the discharge coefficient, \( C_d \), and the reference area, \( A_r \), gives the effective flow area of the valve assembly.

\[
A_e = C_d \cdot A_r
\]  

(3.5)

The most commonly used reference area in practice is the valve curtain area, which is as follows:

\[
A_c = \pi D_v L_v
\]  

(3.6)

where \( D_v \) is the valve diameter and \( L_v \) is the valve lift. The above mentioned curtain area is easy to determine and varies linearly with the valve lift.

Heat transfer calculations in the engine systems are primarily based on the convection and conduction equations. For many applications, the accepted form of energy equation for 2-dimensional, incompressible, laminar flow of a viscous fluid is as following [73]:

\[
\begin{align*}
    k \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) + \mu \left( 2 \left( \frac{\partial v_y}{\partial x} \right)^2 + 2 \left( \frac{\partial v_y}{\partial y} \right)^2 + \left( \frac{\partial v_x}{\partial x} + \frac{\partial v_y}{\partial y} \right)^2 \right) \\
    = c_p \rho \left( v_x \frac{\partial T}{\partial x} + v_y \frac{\partial T}{\partial y} + \frac{\partial T}{\partial t} \right)
\end{align*}
\]  

(3.7)

This may be reduced to a 1-dimensional equation for a steady state as following:

\[
\dot{Q} = hA(T_1 - T_2) \quad (3.8a)
\]

\[
Q = kA \left( \frac{\Delta T}{\Delta x} \right) \quad (3.8b)
\]

Where \( \dot{Q} \) is the heat flux in an engine component, \( h \) is the convective heat transfer coefficient based on experimental correlations, \( A \) is the area of the component taken into account, \( k \) is the thermal conductivity of the metal considered, \( T_1 \) is the gas temperature, \( T_2 \) is the wall temperature considered and \( \frac{\Delta T}{\Delta x} \) is the temperature gradient through the material considered. In unsteady heat transfer, \( \frac{\rho C_p dT}{dt} \) must be included in the heat capacitance of the system.
In Figure 3.1, Batteh et al [9] shows the modeling results of the engine simulation under cold start conditions for 20 seconds. The engine start with a cold cranking and idle speed and runs for 20 seconds before the first acceleration occurs. The above figure shows the thermal response of the engine components during a cold start simulation. The components which receive heat directly from the gas warm up faster. These components are piston, head and liner. The heat capacitance of the piston is the lowest, so it has got the highest temperature rise. The capacitance of the liner and the head lay next to the piston, so the liner and head profile is lower than that of the piston.

Figure 3.1: Thermal response of engine components [9].
Figure 3.2 shows the temperature response of the exhaust system with an engine with enhanced thermal management on start-up [9]. The maximum temperature occurs in the exhaust port just near the exit of the gas from the combustion chamber through the valve. The figure also suggests that the temperature of the exhaust port is a function of the engine speed, since the peak of the exhaust port temperature occurs at the maximum engine speed. The maximum temperature occurs due to the availability of maximum amount of mass of mixture in the combustion chamber due to high manifold pressure. The temperature inside the exhaust port decreases, as the gas advances further towards the manifold because of its contact with the cold exhaust pipe due to the flow of coolant. The amount of energy lost through the exhaust manifold and piping losses have to be minimized because this energy is crucial for the catalyst warm-up, so that the time to the catalyst light off temperature is minimized.
3.5 Combustion Heat Transfer

The problem of engine heat transfer is very complicated because of the temperature and velocity of the gas inside the combustion chamber is far from uniform and it changes at a very rapid pace. The problem of heat transfer from the walls to the coolant is not as complicated because the temperature fluctuations do not penetrate far into the cylinder walls. However, there are spatial variations in convection from cylinder to cylinder because of the sequential flow of water through the engine and variation in the local passage sizes, leading to variations in coolant side temperature.

3.5.1 Heat Transfer to the Combustion Chamber

A number of studies have been done to analyze the heat transfer inside the engine [38, 46, 74-87]. For analyzing the in-cylinder heat transfer, there are three general approaches. They are:

- Approach based on steady convective heat transfer approach, using the correlations for Nusselt, Reynolds and Prandtl number.
- Approach based on solving the unsteady heat conduction equation together with the equation for conservation of energy in the boundary layer, taking into account the effect of periodic combustion around ports, valves, plugs etc.
- Approach based on the conservation of energy of the whole mass contained inside the cylinder.

In one of the studies performed to find the heat transfer correlation, Maciejewski and Anderson [83] presents algebraic empirical relations used for the calculation of heat transfer based purely on the Nusselt, Reynolds and Prandtl numbers. These correlations vary according to the geometry and specifications of the component used as the mode of heat transfer. The authors have come up with a general correlation, assuming that the wall heat transfer rate rises with turbulent velocity fluctuations and also that the adiabatic temperature rise is one of the major driving forces of heat transfer. According to this correlation, the wall heat flux is a function of
turbulent velocity fluctuations, the adiabatic temperature rise and fluid properties like density, viscosity, thermal conductivity and specific heat. Martorano [84] discussed the second approach of heat transfer in the wall as an approach to calculate the convection from the gas to the wall, joined with the conservation of energy equation in the boundary layer. The boundary layer region in this case is very important because the flame quenches in close proximity to the wall and a steep temperature gradient exists in this boundary layer. So, it becomes very important to have a correct assessment of the boundary layer thickness in this region. The boundary layer thickness is more important in smaller engines because the volume to surface area ratio is small and indicates a higher effective heat transfer coefficient. The boundary layer also changes with the crank angle position in the engine cycle. It is the thinnest when combustion occurs and thicker at later times during the engine cycle. Martorano [84] assumed a correlation between the boundary layer thickness with the length of the flame quenching thickness; i.e., the heat transfer inside the cylinder from the gas to the wall depends very largely on the quenching thickness. This quenching thickness is different for different position of the crank, depending upon the Peclet number. Thus, angular pressure fluctuations inside the cylinder increase the heat transfer and introduce oscillations in the heat transfer regime.

The heat transfer in a steady state from a fluid to the surface is given by the equation (3.8a). However, the situation changes in a reciprocating engine where the conditions are not steady but vary cyclically. If the process is considered quasi-steady, then the instantaneous heat transfer rate is proportional to the temperature difference existing at that instant. Because of the thermal capacity of the fluid, there is a lag between the driving temperature difference and the heat transfer rate. This lag is called phase lag, as discussed by Annand [75]. He also discussed that when the engine is motored, the heat transfer rate largely depends upon the gas properties such as the thermal conductivity $k$, dynamic viscosity $\mu$, specific heat $C_p$ and density $\rho$ of the gas, but when firing, the rate of heat release per unit
volume and the oscillatory components depends not only on the engine rotational speed \( N \), but also on the work or IMEP.

For quantifying convective heat transfer, it is necessary to know the temperature of the gas. As this temperature cannot be measured directly, an instantaneous mean value is usually calculated from the instantaneous pressure. It is quite unrealistic to use the same assumption during the combustion phase of the cycle as the temperature may not be so uniform across the chamber in this stroke and, making use of a mean value at a given crank angle position in this stroke, may not yield satisfactory results. However, as no other better alternative is available to measure the temperature in the combustion stroke, a mean value of temperature is taken for this stroke as well.

Following the basic correlation between the Nusselt and Reynolds numbers for the investigation, which is

\[
Nu = a \, Re^b
\]  

for convection heat transfer and

\[
q_{\text{radiant}} = Ac(T^4 - T_w^4)
\]  

for the radiation heat transfer, where \( c \equiv 0 \) for the compression stroke and \( c = \text{constt.} \) for the combustion and expansion stroke. Therefore, the combined equation becomes as follows:

\[
\frac{q}{A} = a \frac{k}{D} (Re)^b (T - T_w) + c(T^4 - T_w^4)
\]

The values of \( a, b, \) & \( c \) were calculated from the analysis experiments in a two-stroke engine by Annand [75]:

\[
\begin{align*}
a &= 0.76 \\
b &= 0.64 \pm 0.10 \\
c &= (1.48 \pm 0.52) \times 10^{-12}
\end{align*}
\]

Woschni [76] found that during the combustion stroke and expansion stroke, heat is being transferred from the hot combustion gases to the combustion chamber walls, but on the contrary, in the induction and early compression strokes, the flow of heat is opposite; i.e. from the walls to the fresh charge arriving through the intake port. Due to this fact, the process in
the cylinder not only changes locally but also with respect to time and it is very difficult to measure it.

Woschni [76] found a universally applicable equation designed for the evaluation of heat transfer coefficients, which is as follows:

$$h = 3.26 B^{-0.3} p^{0.8} T^{-0.5} w^{0.8}$$

(3.12)

where $B$ is the bore in meters, $T$ is the temperature in Kelvin, $p$ is the pressure in kPa and $w$ is the velocity in m/s, but the average cylinder velocity is calculated from motored (suffix m) and reference (suffix r) conditions and the mean piston speed is $\bar{S}_r$. So, the equation for velocity becomes

$$w = \left[ c_1 \bar{S}_r + c_2 \frac{V_d T_r}{p_r v_r} (p - p_m) \right]$$

(3.13)

The value of constants $c_1$ and $c_2$ are taken as follows:

$c_1 = 6.18$ and $c_2 = 0$ for gas exchange period
$c_1 = 2.28$ and $c_2 = 0$ during compression
$c_1 = 2.28$ and $c_2 = 3.24 \times 10^{-3}$ during combustion and expansion

Woschni [76] laid stress on calculating the heat transfer throughout the engine cycle, because the cyclic gas temperature. This approach whilst correct, integrates to the overall heat transfer to the cylinder wall at the completion of the engine cycle. So calculating the heat transfer in different engine cycles can be ignored as the present work only relates to the overall heat transferred to the cylinder walls.

According to Woschni [76], the heat transferred to the throat of the inlet and exhaust valves are investigated as a function of valve lift and Reynold's number. Taking instantaneous values of the valve lift and mass flow rate, the authors have calculated the heat transfer to the valves. Subtracting this heat transfer from the total heat transfer, they found the value of heat transfer to the walls during scavenging period.

Harigaya et al [79] concluded that there is a strong correlation between the beginning of the initial high rate of increase of heat flux and the flame arrival time, which has been measured by the authors at positions they selected in
the combustion chamber. The maximum heat flux decreases with the increase in the flame arrival time and the factors that affect the maximum heat flux are gas flow, gas temperature and pressure at flame arrival time. The authors stated that at a position where the flame front arrival is earlier, the coefficient of heat transfer is higher but at a position where the flame arrival is late, the coefficient of heat transfer is lower.

Shayler et al [77] stated that there is a linear dependence of heat transfer on peak pressure and it varies on a cycle to cycle basis. The variation in heat transfer is slightly dependent on the temperature variations. The heat transfer fluctuations are directly proportional to the combustion stability and as the combustion approaches the lean limit, the heat transfer fluctuations increase.

3.5.2 Effects of Wall Material on Heat Transfer

[80-82] investigated the effects of wall material on heat transfer and they found that heat transfer coefficient has great dependence on the material of the cylinder wall and the heat transfer coefficient, consequently, affecting the surface temperature. Woschni [82] reported a gain in fuel consumption of approximately 30% with the use of ceramic material.

3.5.3 Heat Transfer from the Combustion Chamber Walls

The net heat transfer from the combustion chamber can be deduced together with the variation of mass fraction burnt as a function of crank angle. According to the first law of thermodynamics, the net heat transfer is calculated during the closed part of the cycle, particularly during the interval when the combustion rate is high. This interval contributes a substantial amount to the total.

Shayler et al [88] found that this approach did not agree with the experimental data, so the authors explored an alternative approach. This approach relates to reasonable assumptions made by the authors that the total net heat transfer over a cycle is the difference between the heat
rejected to the coolant and the gas heat transfer to the exhaust port surfaces. The authors have calibrated Woschni’s correlation with Taylor and Toong correlation for finding the heat transfer to the coolant, together with a correlation for finding the heat transfer to the exhaust port surfaces. Of all the results obtained using Annand, Woschni and Eichelberg correlations, the most promising results, which are in good agreement with the Taylor and Toong results are the results obtained by using Woschni’s correlation. However, both Annand’s and Woschni’s results agree reasonably with the cycle averaged heat transfer rates determined by using the Taylor and Toong correlation used for calculating heat transfer to the coolant.

Shayler et al [88] also found that the heat transfer rates from the cylinder using Woschni’s correlation are a maximum at near stoichiometric values and consistent with combustion temperature reaching a maximum value and combustion phasing being optimum. The exhaust port heat transfer has very little effect with Air Fuel Ratio changes, due to small changes in gas flow and gas to surface temperature difference.

### 3.6 Intake and Exhaust System Heat Transfer

The heat transfer in the intake and exhaust system is not in any way simpler than the in-cylinder heat transfer, because it is governed mainly by the flow velocities, which are even higher than in-cylinder velocities. Intake system heat transfer is usually described by steady, turbulent pipe flow correlations. Exhaust port is the area where maximum heat transfer takes place, because of the high gas velocities during the exhaust blow-down process and high gas temperature. Highest heat transfer rates occur during blow-down to the exhaust valve and port. Detailed exhaust port heat transfer correlations have been developed and they are based on the Nusselt-Reynolds number correlations. As discussed by Heywood [34], for the valve open period, relations of the form

\[ Nu = K \, Re^{n} \]  

(3.14)
for $L_v/D_v \leq 0.2$, the flow exits the valve as a jet and $Re_j = \nu_j D_v/\nu$, where $\nu$ is the kinematic viscosity, $D_v$ is the valve diameter and $\nu_j$ is the velocity of the exhaust gases through the valve opening. For $L_v/D_v \geq 0.2$, the port is a limiting area and a pipe flow model with $Re = \nu_p D_p/\nu$ is more appropriate; where $\nu_p$ is the velocity in the port and $D_p$ is the port diameter. For a closed valve period, the correlations that was developed was

$$Nu = 0.022Re_D^{0.8}$$  \hspace{1cm} (3.15)

where $Re_D = \overline{\nu_p D_p}/\nu$, such that $\overline{\nu_p}$ is the time averaged exhaust port gas velocity and $D_p$ is the port diameter. For the section extending downstream of the exhaust port, in the exhaust manifold, an empirical correlation based on the measurement of average heat transfer rates considering straight section pipe has been developed, which is

$$Nu = 0.0483Re_D^{0.783}$$ \hspace{1cm} (3.16)

Concluding the above discussion, it can be noticed that the gas side heat transfer correlations used in the intake and exhaust system have always been of the form of equation (3.7).

Depoik and Assanis [89] tried to find out a general correlation for both the intake and exhaust flow systems, depending upon the assumptions that both the systems have turbulent pipe flows and only the mean flow rate is important. The authors worked on the strategy to fix one of the coefficients and changed the other through experimental data. The authors approach is based on the turbulent micro-scale, in which they have fixed the exponential factor of the Reynolds number and determined the other constant by means of experiments. Using the experimental data for exhaust and intake side, the authors came up with a universal correlation as

$$Nu = 0.07 Re_D^{3/4}$$ \hspace{1cm} (3.17)

The authors state that it can be used for manifold flows. The authors found through experimental data and their proposed heat transfer model that the correlation constant for exhaust side is 0.800 and for intake side it is 0.845. Another correlation used by Sleicher and Rouse [70] is as follows:
\[ \text{Nu}_D = 5 + 0.015 \text{Re}^{a_{D,f}} \text{Pr}^{b_{w}} \]  \hspace{1cm} (3.18)

where \( a \) and \( b \) are functions of Prandtl number (typical values are 0.83 and 0.66).

Since, all the engines have different geometries and design, the heat transfer is found to be different for every engine, even if the magnitude of flow of gases is the same. Bohac et al [69] tried to take the mean flow into account and the fluctuating components have been entered through experimental data.

Zeng and Assanis [90] not only discussed the governing equation for modeling the flow but also the correction factor needed. The Nusselt and Reynold’s correlations can predict steady flow heat transfer, but the other part of the flow of heat, which is unsteady, cannot be ignored, as there are large velocity variations in an engine. These variations can produce errors in both phase and magnitude. The authors have introduced a correction factor for unsteady heat transfer in the Nusselt and Reynold’s correlation and the final equation that the authors have derived becomes:

\[
\text{Nu}_{dy} = 0.023 \text{Re}_{\text{steady solution}}^{0.8} \text{Pr}^{0.4} \left[1 - 0.75 \frac{D}{U^2} \frac{dU}{dt} \right]^{0.8} \]  \hspace{1cm} (3.19)

where \( \text{Nu}_{dy} \) is the dynamic Nusselt number derived from the dynamic Reynolds number and dynamic velocity.

### 3.8 Approach to Modeling in Modelica

#### 3.8.1 Introduction to Modelica

Modelica is a product of Dymola (Dynamic Modeling Laboratory). Modelica is a new object-oriented multi-domain language for modeling large, complex and heterogeneous physical systems. The first version of the language came into existence in 1996, which was later improved to a newer version 1.1 in 1998, version 1.3 in 1999 and version 1.4 in 2000. Models in Modelica are mathematically described by differential, algebraic and
discrete equations. Modelica is designed to handle a system of more than a hundred thousand equations with the help of specialized algorithm [91, 92]. In Modelica, like all other modeling or computer programming languages, the concept of composition and decomposition plays a vital role in developing an understanding of the entire system. Decomposition means that the system can be broken into several subsystems to make it easy to understand. On the contrary, several subsystems can be combined together in a particular hierarchy to make a single large system, this is termed as composition.

Modelica’s concept of class parameters extends the notion of parameters to a higher level. This concept states that complete sub models can be replaced with other sub models if they fulfill certain criteria or restrictions. The concept of object oriented approach is a convenient tool for system decomposition as all subsystems are represented as classes. Modelica support multiple inheritance, which plays an important role in object oriented languages. A class can be defined as a subclass to another class which is the base class or super class and all the attributes of the super class can be made available in the subclass and new attributes can be added to the subclass, if required. This is termed as inheritance.

Modelica is a high quality simulation tool, with industry strength model libraries for several application domains, which are capable of supporting large and complex systems. The model libraries are helpful in assembling complex plant topologies from basic building blocks of the model so that the resulting model is physically correct and possible to simulate efficiently [93, 94].

The importance of modeling engine heat transfer has been discussed by [9, 34]. It provides a convenient and efficient means to evaluate different start-up strategies and hardware design and their effects on the exhaust system thermal response. Modelica is an open source which compiles faster than other low level languages and provides a robust platform for running heavy applications like multi-domain engine system modeling. These multi-domain models are best suited for the evaluation and optimization of hardware
design and control strategies, especially in the early stage of the design process.

The Modelica standard mechanical, rotational, multibody and thermal libraries contain the connector definitions, interface and basic models that provide the framework for modeling of engine systems, as discussed by Batteh [9].

3.9 Model in the Current Study

3.9.1 Heat Transfer Model

A single cylinder model is built in Dymola based on the resistor-capacitor network as the model described by [69, 71]. In this model, the engine intake mixture is tracked from the port entrance. It goes from the port entrance to the combustion chamber. When the engine is operating under steady state conditions, convective heat transfer takes place from the fluid to the port walls, intake valve stem and intake valve seat, but during start-up the heat transfer is reversed and it takes place from the valve stem, valve fillet and port walls to the fuel air mixture.

Furthermore, inside the combustion chamber, the heat produced by the hot burning gases is convected to the piston crown, the upper and middle layer of the cylinder wall and the region of the head, which are directly exposed to the hot gases. The lower part of the cylinder wall receives heat from the piston because it does not communicate directly with the combustion gases. The coolant draws the heat out of the engine block, cylinder walls, cylinder head, and inlet and exhaust port. It moves further into the head heat exchanger to complete the cycle and then fed back into the engine block for the next cycle.

The oil sump receives heat from the piston and the lower cylinder wall. Some heat is transferred to the engine block and some to the ambient surroundings. Moreover, the single cylinder model also has an oil heat
exchanger. The model uses 3 types of thermal resistances, which are in-built in Modelica thermal library. They are:

- Conduction resistance
  \[ R = \frac{kA}{L} \]  
  (3.20)

- Convection resistance
  \[ R = hA \]  
  (3.21)

- Flow resistor / Capacitance
  \[ R = \dot{m}C_p \]  
  (3.22)

The conduction and convection resistors are inbuilt resistors in Modelica thermal-heat transfer library, whereas the flow resistor transports coolant into the engine where heat is exchanged between the coolant and engine components. A schematic diagram of the heat flow is given in figure 3.3. To find the complete details about the development of the model, the reader is advised to refer to appendix G.
Figure 3.3: Schematic heat flow chart of the Resistor-Capacitor model
3.10 Summary
Modeling of engine heat transfer is an important tool to assess design changes with least prototype construction. The process is therefore cost effective and time saving. In the present chapter, an effort has been made to develop a model for engine heat transfer and coolant flow.
Chapter 4 – Experimental Set up

4.1 Introduction
This chapter presents a discussion on the experimental set up, the experimental targets and testing strategy. It only provides a brief overview and complete details of the engine set up, modifications, instrumentation and experimental strategy can be found in appendices C - F.

4.2 The Test Engine
The test engine used was a Ford I-6, BF Falcon Engine installed on the Hennan and Froude Dynamic Dynamometer. The test engine is the latest model of Ford Falcon, which is an extremely popular vehicle in Australia. This new model of Ford Falcon, called BF Falcon has been launched in Australia in October 2005. For further research and development on the engine, an experimental test rig was built up in the University of Melbourne thermal research laboratory with all the required assistance provided by Ford Motor Co., Australia. The installation of the engine on the dynamometer test bench has been referred to the work done by Baker [95]. Figure 4.2 and figure 4.3 show a schematic diagram and a picture of the engine test cell respectively, whereas, engine specifications are presented in table 4.1.
The test engine was modified to install K-type thermocouples on it. These thermocouples were installed to measure components temperatures at various locations on the engine. There were 56 thermocouples installed on the engine and a brief overview about the locations of thermocouples on the engine can be as follows:

- 27 thermocouples installed on the engine block.
- 21 thermocouples installed on the cylinder head.
- 6 thermocouples in the cylinder head for measuring coolant temperature (1 in each cylinder).
- 2 thermocouples for coolant entry and exit (on thermostat and water pump)

Complete details about the locations of the thermocouples can be found in appendix C.

4.3 Experimental Targets

The experiment was designed to investigate the options of reduction in warm up time of the engine and thereby, leading to fuel economy benefits. The investigation can be summarized as follows:

- Preliminary tests.
- Main experiments.
- Variable coolant flow rate test utilizing an electric water pump.
- Split cooling system tests.

4.3.1 Preliminary Test

The idle tests were done to develop an understanding of the default Ford PCM and the working condition of the thermocouples in the engine.

4.3.2 Main Experiments

It was decided to run the engine on a simulated drive cycle called NEDC (New European Drive Cycle), as though it was installed in the Ford Falcon
An average of speed and load for 180 seconds of engine running on this drive cycle was calculated, which was 1161 rev/min and 48 Nm load.

4.3.3 Variable Coolant Flow Rate Test
The coolant flow rate was found at the speed of 1161 rev/min test point and then by varying the speed of the electric water pump, to investigate the effects of reduction in coolant flow rate on engine warm-up time and fuel consumption. The purpose of running the engine with variable coolant flow rate was not only to attempt to reduce the engine warm-up time and fuel consumption but also to investigate the behaviour of engine components when subjected to different temperature conditions. The experiments for variable coolant flow rate were done from maximum to minimum flow rate and also for no flow condition.

4.3.4 Split Cooling Test
A split cooling exercise was undertaken to further investigate any fuel consumption benefits and reduction in engine warm-up time. The investigation was divided into two parts. They are

- To run the engine block with 0 L/min and cylinder head with minimum possible flow.
- To run the cylinder head with 0 L/min and engine block with minimum flow possible.

4.4 Experimental Strategy
The experimental strategy was to run the engine on a simulated New European Drive Cycle (NEDC). As discussed in appendix F, it involved running the engine on fixed speed and load conditions. The fixed speed and load conditions were averaged over 180 seconds of engine running, which was repeated 4 times in the first part of the drive cycle and calculated to be 1161 rev/min engine speed and 48 Nm load. According to the
experimental strategy, the engine was run at idle for the first 20 seconds and then subjected to the fixed speed and load operating point in every test. The reader is advised to refer to appendix F for a detailed discussion on NEDC.

4.4.1 Calculation of Mean Torque and Mean Engine Speed
The mean test speed and torque were determined as follows:
The department’s CARSIM model [10] was configured with the values of inertia, rolling resistance and aerodynamic drag, engine maps and other data for the Ford Falcon AU sedan. The model was then driven over the Euro 2/3 or New European Drive Cycle (NEDC) seen in figure 4.1 (a). The model predicted engine speed and torque values, which are plotted in figure 4.1 (b) and figure 4.1 (c). The values were averaged over 196 seconds of the 4 times repeated part of the urban cycle in the NEDC to produce an average speed of 1161 rev/min and average torque of 48 Nm.
Figure 4.1: Prediction of (a) Vehicle Speed (b) Engine Speed and (c) Engine Torque from CARSIM mode[10].
A schematic diagram of the engine test cell is presented in figure 4.2. Figure 4.3 shows a picture of the actual engine test cell and data acquisition board and Table 4.1 shows the engine specifications.

Figure 4.2: Engine Schematic Diagram and Data Acquisition Board.
Figure 4.3: Engine Test Cell and Data Acquisition Boards
<table>
<thead>
<tr>
<th>Engine Configuration</th>
<th>Vertical Inline</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Type</td>
<td>6 cylinders, 4 stroke, water cooled</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>$R_c = 10.3$</td>
</tr>
<tr>
<td>Length of connecting rod</td>
<td>153.85 mm</td>
</tr>
<tr>
<td>Capacity/cylinder</td>
<td>664 cc (0.664L)</td>
</tr>
<tr>
<td>Bore x Stroke</td>
<td>92.25 x 99.4 mm</td>
</tr>
<tr>
<td>Valve Train</td>
<td>Double OHC, 4 valves per cylinder</td>
</tr>
<tr>
<td>Firing Order</td>
<td>1, 5, 3, 6, 2, 4</td>
</tr>
<tr>
<td>Maximum Brake Torque</td>
<td>393 Nm at 2500 rev/min</td>
</tr>
<tr>
<td>Maximum Power</td>
<td>190 kW at 5200 rev/min</td>
</tr>
</tbody>
</table>
| Valve Timing         | IVO = 5.5 BTDC at maximum power  
                        | EVO = 486.5 ATDC at maximum power |
| Inlet valve diameter (x2) | 35 mm                       |
| Inlet valve lift     | 11 mm                          |
| Exhaust valve diameter (x2) | 32 mm                     |
| Exhaust valve lift   | 11 mm                          |
4.5 Summary

In this chapter, a brief overview of the experimental set-up, experimental strategy, engine modification for thermocouple locations has been provided. However, the details are presented in appendix C, D, E and F. The results of these experiments are discussed in chapter 6.
Chapter 5 – Modeling Results

5.1 Introduction
In chapter 3, a detailed heat transfer model has been discussed. The present chapter covers the further work of simulating the model and includes the modeling results. The modeling results give an understanding of heat transfer in different engine components under transient cold start operating conditions. The temperatures of the components included in the simulation are:

- Cylinder head
- Engine block
- Engine oil
- Cylinder head coolant
- Piston crown
- Piston skirt
- Upper cylinder wall
- Middle cylinder wall
- Lower cylinder wall
5.2 Model Simulation

The simulation is carried out with the fixed step size of 0.0001 and the maximum integration is by a 4th order Runge-Kutta method. While compiling, the Modelica code is converted into a C code and the compiler used is the GNU compiler collection often called GCC compiler, commonly used to compile C programs. The present model has been calibrated against 1161 rev/min and 48 Nm of load and validated against 700 rev/min and 0 Nm load. The model simulation is being run for 250 seconds. The results of the model are plotted as a time-temperature graph, the time being in seconds and the temperature being in Kelvin.

5.3 Initial Conditions for Simulation

As discussed in chapter 3, the combustion pressure and temperature data with respect to time and crank angle swept is used from the model of Keynejad [96], as the heat input for the present model under study. Therefore, the heat produced in the combustion chamber varies as a function of time, crank angle, engine speed, load, manifold pressure, bore, stroke, cylinder wall thickness, air charge temperature and inlet mass flow rate. The flow of heat from the combustion chamber to the engine components is also a factor of oil and coolant flow rates.

The initial conditions at the start of the simulation are as follows:

<table>
<thead>
<tr>
<th>Variable</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed</td>
<td>1161 rev/min</td>
</tr>
<tr>
<td>Load</td>
<td>48 Nm</td>
</tr>
<tr>
<td>Temperature (inlet)</td>
<td>293K = 20°C</td>
</tr>
<tr>
<td>Bore</td>
<td>0.092 m</td>
</tr>
<tr>
<td>Stroke</td>
<td>0.099 m</td>
</tr>
<tr>
<td>Wall thickness</td>
<td>0.075 m</td>
</tr>
<tr>
<td>( mCp_{water} )</td>
<td>( 1045 J/°C )</td>
</tr>
<tr>
<td>( mCp_{oil} )</td>
<td>( 1530 J/°C )</td>
</tr>
</tbody>
</table>

\( mCp_{water} \) = Capacitance for coolant

\( mCp_{oil} \) = Capacitance for oil
5.4 Simulation Results for NEDC Test Point (1161 rev/min, 48 Nm Load)

The model has been calibrated at the NEDC test point of 1161 rev/min and 48 Nm load. The results obtained are discussed below.

5.4.1 Head, Block, Head Coolant and Oil Results

As discussed earlier, in the present model, the cylinder wall is divided into 3 equal parts. The region below the cylinders in the block, which receives heat through conduction from the oil sump and engine coolant is referred to as engine block in the model. It has to be noted that only the cylinder head, upper portion of the cylinder wall and the piston crown receive heat directly from combustion and not the engine block. The results are lumped to represent the average of all the head and the block.

Figure 5.1 shows the modeling results for the engine component temperatures that will be compared against measurement in Chapter 7. These temperatures are the block temperature, cylinder head temperature, head coolant and engine oil temperature. It can be noticed that there is a significant difference between the cylinder head and engine block temperature. This is due to the fact that in the current model, the block (excluding the cylinders) is modeled with the heat being received by conduction and through the engine coolant. However, the cylinder head receives heat directly from the combustion chamber.

As shown in the figure, at the end of 250 seconds, the approximate temperatures of the cylinder head reaches $353\,\text{K}$, the cylinder head coolant is $336\,\text{K}$, the engine block is $332\,\text{K}$ and the engine oil temperature is $316\,\text{K}$.
5.4.2 Piston and Cylinder Walls Results

Figure 5.2 includes the heat flow pattern in the piston crown, skirt, and the cylinder walls. As the piston crown receives the largest amount of heat from the combustion chamber, it has the highest temperature rise. The piston crown and cylinder upper wall are the two components, directly in contact with the flame, when it is generated, but when the piston starts moving down from the TDC, in the expansion stroke, heat from combustion is further shared by the middle cylinder wall. Moreover, particularly after combustion, the piston skirt receives heat from the piston crown by conduction.
This is shown in the results in figure 5.2, where piston crown is running at the highest temperature. At the end of 250 seconds, the temperature of the piston crown is $382\,K$ approximately. The temperature of the cylinder walls and piston skirt are close to each other without much difference. However, as discussed earlier, the cylinder upper wall receives heat directly from combustion and its temperature of $339\,K$ approximately is slightly higher than the other cylinder walls and the piston skirt at the end of 250 seconds. The temperatures of piston skirt, middle cylinder wall and lower cylinder wall are very close to each other. At the end of 250 seconds, their temperatures are between $336-338\,K$ approximately, with the middle wall highest at $338\,K$, piston skirt in the middle as $337\,K$ and the lower wall at the bottom with the temperature of $336\,K$.

![Figure 5.2: Modeling results for piston crown temperature, piston skirt temperature, cylinder upper wall temperature, cylinder middle wall temperature and cylinder lower wall temperature at NEDC test point (1161 rev/min and 48 Nm load).](image-url)
5.5 Simulation Results for Idle (700 rev/min, 0 Nm Load)
The calibrated model is validated against a second set of data points, which is 700 rev/min and 0 Nm load. The coolant flow rate has been reduced in line, with the coolant flow rate, measured in the experiments and found to be reduced by about 50%, according to the graph in Appendix E.3, when the engine speed reduces from 1161 rev/min to 700 rev/min. The convective heat transfer coefficient from the cylinder head to the coolant and from the walls to the coolant also reduces with engine speed.

5.5.1 Head, Block, Head Coolant and Oil Results
Figure 5.3 shows the modeling results for engine block temperature, cylinder head temperature, cylinder head coolant temperature and engine oil temperature. The results show a decline in temperatures as the engine speed, load, coolant flow rate and heat transfer coefficients have been reduced. At the end of 250 seconds, the cylinder head temperature is $334 \, K$ approximately and engine oil temperature is $305 \, K$ approximately. The engine block temperature and the cylinder head coolant temperatures are nearly coinciding and they are $318 \, K$ approximately after 250 seconds.
5.5.2 Piston and Cylinder Walls Results

Figure 5.4 shows the results for piston and cylinder wall temperatures. At the end of 250 seconds, the temperature of the piston crown is 336 K approximately. The temperatures of the piston skirt, cylinder middle wall and cylinder lower wall are nearly coinciding at 319 K approximately. The temperature of the upper cylinder wall is running slightly higher, with a difference of 1 K and is 320 K approximately.

Figure 5.3: Modeling results for engine block temperature, cylinder head temperature, head coolant temperature and engine oil temperature at Idle 700 rev/min and 0 Nm load.
5.6 Summary

The heat transfer model discussed in chapter 3 has been applied and the results are discussed in this chapter. The simulation has been carried out for conditions of NEDC test point of 1161 rev/min, 48 Nm load and Idle 700 rev/min and 0 Nm load. These results are further compared to the experimental results in chapter 7. The comparison shows that the results are in reasonable agreement with each other.
Chapter 6 – Experimental Results and Discussion

6.1 Introduction

The research strategy was to achieve faster warm-up through reducing the coolant flow rate, with an electric water pump, while running the engine on a simulated drive cycle. The present chapter includes the results obtained by implementing this strategy. The chapter gives a detailed understanding of the Ford Power-Train control module (PCM). The various engine variables like spark advance, engine speed, MAP and fuel consumption, monitored by the data acquisition system are discussed. Moreover, temperature-time profiles of the various engine measuring points are compared. Finally, the effects on engine warm up time and the variation in temperature of the coolant, with reducing coolant flow rate has been discussed in this chapter. To have an initial understanding about the coolant flow and the geometry of the water passages, the reader is advised to refer to the following figures, i.e., figure 6.1 and 6.2.
Figure 6.1: Engine block schematic diagram with coolant transfer holes

Figure 6.2: Schematic of coolant flow in the engine
6.2 General Discussion on Coolant Flow in the Engine

When an engine is started, the water pump, which is located on the exhaust side, sucks coolant from the radiator and heater return line and feeds the coolant to the engine block. However, there is no flow through the radiator at this time, as the thermostat is closed. There are coolant transfer holes on the block deck and the water jacket extends deep inside the block across the cylinder bores. The water jacket is a cavity around the cylinders, which normally finishes 6-10 mm before the cylinder head surface of the block. This is a closed deck design, in contrast to an open deck design, when the water jacket extends up to the head face.

The coolant flow starts from cylinder 1 and gets distributed evenly on both the inlet and exhaust sides of the engine block. It flows around the cylinders and proceeds towards cylinder 6 in the block. Some of the coolant feeds up into the head through the transfer holes provided around each cylinder. However, the quantity of coolant going up from the block to the head is small, because the area of these holes is small. When the coolant reaches cylinder 6 and, since the area of water holes is larger around cylinder 6, the major quantity of coolant rises up in the head here and in the process, there is more circulation of coolant around cylinder 6 in the block.

When the coolant enters the head at cylinder 6, it starts to extract heat from the head and gets warmer as it moves from cylinder 6 towards cylinder 1. The coolant gets heated all the way to cylinder 1 in the head. Although, the area of the water jacket around cylinder 1 is similar to the area of the water jacket around cylinder 6, there is more convection of coolant around cylinder 6 in the block, hence, the temperatures around cylinder 6 are found to be lower.

The thermostat remains closed until the engine reaches its operating temperature, with the flow, which reaches cylinder 1 in the head, diverted through the heater circuit. This heated coolant from the engine, extracted from the thermostat body, heats up the cabin of the vehicle, if the air path in the heater is open, and flows back to the water pump and is fed again into the block.
Once, the engine reaches the operating temperature, the thermostat opens and there is sudden rush of cold coolant into the block which gives a thermal shock to the engine components. Moreover, there is a drop in the flow rate of the coolant flowing through the heater circuit after the thermostat opens and most of it goes into the radiator. With the sudden circulation of cold coolant into the engine, the temperature drops considerably, consequently, bringing the engine temperature down. As the temperature of the engine reduces, the thermostat closes. Although, this cycling continues with the thermostat opening and closing, the radiator temperature still increases gradually due to the mixing of hot and cold coolant, if there is no natural air flow. When the preset radiator temperature is reached, the electric fan on the radiator turns on and it brings the temperature down.

6.3 General Discussion on Lowering the Coolant Flow Rate

When the flow rate of the coolant is lowered in the engine, the lower velocities reduce the Reynold’s number. This lowers the Nusselt number and consequently, the heat transfer coefficient. Hence, the heat transfer is lower and higher component temperatures result to maintain the heat flux. When the engine is cold started, heat is transferred from the components to the coolant flowing around the cylinders, which is further dissipated in the heat capacity of the block and the head externals and with secondary losses to the surrounding air. Since the speed of the mechanical water pump is a function of the engine speed, the flow of coolant increases with engine speed. It follows that the greater is the coolant flow rate, the higher is the heat extracted, which is undesirable at this stage when the engine is running cold.

A technique to test the effects of varying pump speed was to employ an electrical water pump to vary the flow rate of the coolant in the engine. The engine was run at NEDC test points of 1161 rev/min and 48 Nm load. The average coolant flow rate measured at this speed with the mechanical water pump was found to be 17.17 L/min. The testing was mainly focused
on the reduction in warm-up time possible, when the flow rate of water was reduced from 17.17 L/min to a minimum of 3.17 L/min, in several steps and also a no flow condition.

It was expected that as the flow rate of coolant is reduced, there would be decreased heat transfer, consequently, leading to faster warm up of the engine internal component. It was also expected in the results that when the flow rate of the coolant is lowered, and the components heated faster, they would have an impact on increasing the oil temperature. Further, it was expected that with the reduced flow rate, the cylinder bore, liner and the piston temperatures would be higher and thus would lead to heating up the oil faster and subsequently help in reducing the overall engine friction. All these benefits should lead to reduced fuel consumption and engine warm-up time.

6.4 Experimental Results

6.4.1 Results included in the Analysis

The testing has been done for various flow rates for the EWP. These comprise of idle conditions and simulated NEDC condition for which, the flow has been reduced in steps of 3L/min from the normal of 17.17 L/min of the MWP with the EWP to 0 L/min.

For the split cooling exercise, the analysis is presented for 3.17 L/min, flow in head and 3.17 L/min, flow in block. The analysis is presented for all the conditions as an average taken over three to four tests, performed under each condition. The results are presented as the averaged temperatures of the cylinder head, engine block, oil, cylinder head coolant and the fuel consumption. The simulated urban drive cycle fuel used on the test bed at the fixed load and speed point agreed closely with the actual drive cycle data provided for a Falcon car on the Ford ADR 79/01 test facility for 740 seconds of the urban cycle. The fuel consumption agreed within 0.5% and the thermostat open time was only 10 seconds in error, (personal
communication) R. Dingli [97]. This encouraged confidence that the steady state and load test was adequate to replicate the in-car performance over the relatively low speed (18 km/h – average speed) part of the NEDC.

6.5 Discussion of Oil Temperature
Although, there are improvements in fuel consumption, head and block temperatures and coolant temperatures, there was no significant change in the oil temperature when the flow rate of coolant was reduced. Nonetheless, there are some benefits from the reduced friction from the faster heating of the components, even though, the bulk mass of the oil is not significantly affected. It is noted that the oil in the camshafts returns to the sump through an oil way and return gallery was on the inlet side of the engine. As the flow rate of coolant is reduced, there is increase in temperature of the coolant and also the temperature of the components on the exhaust side. The advantage of heating the exhaust side components, faster could not be exploited completely, because the oil return gallery is located on the inlet side of the engine. Furthermore, when the engine is started cold, most of the oil pumped is re-circulated through the pressure relief valve and therefore, bypassing the hotter parts of the engine.

6.6 Discussion of Coolant Temperature in different Cylinders
Coolant temperature data has been recorded for thermocouples in cylinders 1 to 6 on the exhaust side, in the cylinder head, at the thermostat and for the inlet of the water pump.
As previously mentioned, the water pump resides on the front of the engine in the block, the coolant is fed into the engine through cylinder 1 and from there it travels all the way to cylinder 6 in the block. Some coolant passes into the head, in the vertical direction, through the coolant holes provided in the block through the head gasket. However, the quantity of coolant transferred to the head from the block through these holes, in the vertical direction is very small and most of the coolant flows in the horizontal
direction in the block and reaches cylinder 6. The major quantity of coolant is transferred to the head from the block through the vertical passage of cylinder 6. The area of the water jacket around cylinder 6 is also larger as compared to other middle cylinders. This causes a lot of coolant to circulate around cylinder 6 in the block.

Although, the area of the water jacket around cylinder 1 is equal to the area of the water jacket around cylinder 6, there is some difference in the coolant temperature of cylinder 1 and cylinder 6. This happens because the coolant passes through the other middle cylinders before it reaches cylinder 1 in the head and this raises the temperature of the coolant at cylinder 1 in the head.

It has been observed that except for cylinder 6, the coolant temperature in the head for all the cylinders are in close proximity to each other. This trend is observed for the flow rate of 17.17 L/min (maximum), for both MWP and EWP. However, as the flow rate was further lowered to 3.17 L/min (minimum), it was observed that the temperature profile for all the cylinders remained similar, but with an increasing difference cylinder to cylinder as the coolant flow rate decreased.

6.7 Comparison for different Running Conditions

The tables 6.1 to 6.4, which follow, show the temperature variation, with values averaged for each of the cylinder head and engine block, as well as engine oil and cylinder head coolant outlet temperatures from cold start to 300 seconds. These results are discussed in conjunction with the graphical results presented in the next section and the recorded data for the other engine variables such as MAP and Spark advance in Appendix H.
Table 6.1: Average temperature of Cylinder Head

<table>
<thead>
<tr>
<th>Cylinder Head</th>
<th>Idle (9.17 L/min)</th>
<th>MWP (17.17 L/min)</th>
<th>EWP (17.17 L/min)</th>
<th>EWP (3.17 L/min)</th>
<th>EWP (0 L/min)</th>
<th>Split (3.17 L/min, flow in head)</th>
<th>Split (3.17 L/min, flow in block)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time (s)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>50</td>
<td>44.3</td>
<td>45.1</td>
<td>46.7</td>
<td>48.8</td>
<td>51.3</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>100</td>
<td>53.6</td>
<td>56.3</td>
<td>57.5</td>
<td>60.7</td>
<td>69.3</td>
<td>71</td>
<td>77.4</td>
</tr>
<tr>
<td>150</td>
<td>59.7</td>
<td>62.2</td>
<td>63.8</td>
<td>66.2</td>
<td>80.5</td>
<td>77.6</td>
<td>89.6</td>
</tr>
<tr>
<td>200</td>
<td>63.8</td>
<td>67.5</td>
<td>69.1</td>
<td>71.3</td>
<td>89.3</td>
<td>83.6</td>
<td>99.3</td>
</tr>
<tr>
<td>300</td>
<td>70.8</td>
<td>79.7</td>
<td>80.9</td>
<td>81.6</td>
<td>N/A</td>
<td>94.3</td>
<td>111.2</td>
</tr>
</tbody>
</table>

Table 6.2: Average temperature of Engine Block

<table>
<thead>
<tr>
<th>Engine Block</th>
<th>Idle (9.17 L/min)</th>
<th>MWP (17.17 L/min)</th>
<th>EWP (17.17 L/min)</th>
<th>EWP (3.17 L/min)</th>
<th>EWP (0 L/min)</th>
<th>Split (3.17 L/min, flow in head)</th>
<th>Split (3.17 L/min, flow in block)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time (s)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>50</td>
<td>42.6</td>
<td>44.3</td>
<td>45.8</td>
<td>51</td>
<td>52.3</td>
<td>51.3</td>
<td>54.9</td>
</tr>
<tr>
<td>100</td>
<td>51.3</td>
<td>55.2</td>
<td>56.5</td>
<td>63.7</td>
<td>72.4</td>
<td>63.3</td>
<td>72.1</td>
</tr>
<tr>
<td>150</td>
<td>58.6</td>
<td>61</td>
<td>62.3</td>
<td>68.9</td>
<td>84.1</td>
<td>70.5</td>
<td>83.2</td>
</tr>
<tr>
<td>200</td>
<td>62.4</td>
<td>67.1</td>
<td>67.9</td>
<td>74.5</td>
<td>94</td>
<td>76.8</td>
<td>92.5</td>
</tr>
<tr>
<td>300</td>
<td>70.5</td>
<td>79.5</td>
<td>79.1</td>
<td>86</td>
<td>N/A</td>
<td>88</td>
<td>105.5</td>
</tr>
</tbody>
</table>
### Table 6.3: Average temperature of Engine Oil

<table>
<thead>
<tr>
<th>Engine Oil</th>
<th>Idle (9.17 L/min)</th>
<th>MWP (17.17 L/min)</th>
<th>EWP (17.17 L/min)</th>
<th>EWP (3.17 L/min)</th>
<th>EWP (0 L/min)</th>
<th>Split (3.17 L/min, flow in head)</th>
<th>Split (3.17 L/min, flow in block)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time (s)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>50</td>
<td>20.8</td>
<td>21.5</td>
<td>20.8</td>
<td>21.1</td>
<td>20.6</td>
<td>21</td>
<td>20.9</td>
</tr>
<tr>
<td>100</td>
<td>25.2</td>
<td>26.3</td>
<td>24.5</td>
<td>24.1</td>
<td>23.7</td>
<td>24.5</td>
<td>24</td>
</tr>
<tr>
<td>150</td>
<td>28.5</td>
<td>30.2</td>
<td>29.6</td>
<td>28.3</td>
<td>28</td>
<td>29</td>
<td>27.8</td>
</tr>
<tr>
<td>200</td>
<td>32.9</td>
<td>36.5</td>
<td>34.5</td>
<td>33.4</td>
<td>32.6</td>
<td>33.7</td>
<td>31.8</td>
</tr>
<tr>
<td>300</td>
<td>40.1</td>
<td>44.1</td>
<td>42.9</td>
<td>42</td>
<td>N/A</td>
<td>42.5</td>
<td>39.8</td>
</tr>
</tbody>
</table>

### Table 6.4: Average temperature of Cylinder Head Coolant

<table>
<thead>
<tr>
<th>Cylinder Head Coolant</th>
<th>Idle (9.17 L/min)</th>
<th>MWP (17.17 L/min)</th>
<th>EWP (17.17 L/min)</th>
<th>EWP (3.17 L/min)</th>
<th>EWP (0 L/min)</th>
<th>Split (3.17 L/min, flow in head)</th>
<th>Split (3.17 L/min, flow in block)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time (s)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>50</td>
<td>29.7</td>
<td>32.3</td>
<td>30.6</td>
<td>34.6</td>
<td>38.6</td>
<td>32.2</td>
<td>43.3</td>
</tr>
<tr>
<td>100</td>
<td>41</td>
<td>43.8</td>
<td>41.4</td>
<td>46.7</td>
<td>64.3</td>
<td>42.7</td>
<td>67.5</td>
</tr>
<tr>
<td>150</td>
<td>47.8</td>
<td>50.7</td>
<td>48.8</td>
<td>53.8</td>
<td>82.1</td>
<td>51.7</td>
<td>82.4</td>
</tr>
<tr>
<td>200</td>
<td>53.4</td>
<td>57.5</td>
<td>55.3</td>
<td>61</td>
<td>94.8</td>
<td>59.3</td>
<td>93.1</td>
</tr>
<tr>
<td>300</td>
<td>61.5</td>
<td>70</td>
<td>67.4</td>
<td>74</td>
<td>N/A</td>
<td>72.4</td>
<td>97.1</td>
</tr>
</tbody>
</table>
The following paragraphs include a discussion on the temperature distribution of the various engine components, for various coolant flow conditions and which has also been explained in the above tables.

6.7.1 Preliminary Idle Vs MWP (17.17 L/min)

The reader is asked to refer to figure 6.3. The fuel consumption in the idle test is lower because the engine runs on low speed (idle). The coolant flow rate is 9.17 L/min approximately, as measured by the flow meter. The heat produced in the engine is less than when running the engine at higher speed and load conditions and consequently, the fuel consumption and component temperatures are lower.

![Average Data for Idle (9.17 L/min)](image)

Figure 6.3: Average fuel Consumption, Cylinder Head, Engine Block and Oil temperatures for Idle (9.17 L/min)
6.7.2 Main Test - MWP and EWP (17.17 L/min)

For this comparison, the reader is asked to refer to figure 6.4 (a) and 6.4 (b). For more details, the reader is referred to Appendix I (I.1 to I.4). It can be noticed in the results that the fuel consumption is higher in EWP in the initial 100 seconds. There are 3 main reasons proposed to explain the increase in fuel consumption. They are:

1. Probable biasing of the flow with EWP.
2. The capacitance is higher.
3. The flow is higher in the first 20 seconds.

In case of engine running with a MWP, the exit of the pump is designed to distribute the coolant equally to both inlet and exhaust sides of the engine block aided by the swirling motion to the coolant entering the block. This does not happen in case of the EWP because there is no impellor to provide the flow bias. The distribution of coolant is biased to the exhaust side of the block as the data show that the exhaust side of the block is cooler for cylinder 1, with the EWP. Moreover, the installation of EWP involved extending the rubber hose coming from the bottom of the radiator, requiring the heater circuit return hose also to be extended, thus increasing the coolant circuit capacity by about 1.5 liters.
Figure 6.4 (a): Average fuel Consumption, Cylinder Head, Engine Block and Oil temperatures for 17.17 L/min MWP
There is a further difference in the circulation rate between the MWP and EWP at the start. The flow in the engine running with a MWP is 9.17 L/min approximately at idle, but the flow increases to 17.17 L/min approximately, in the loaded condition. However, in case of an EWP, the pump is run by an external power supply and is started before the engine is started, at the flow rate of 17.17 L/min. Therefore, there is additional flow in the first 20 seconds of idle running which hinders the warm-up of the engine and consequently, more fuel is consumed. As a consequence, there is a lag of 2°C approximately in the temperature rise of the EWP compared to the MWP.

Figure 6.4 (b): Average fuel Consumption, Cylinder Head, Engine Block and Oil temperatures for 17.17 L/min EWP
6.7.3 EWP (3.17 L/min)

As the flow rate of the coolant is reduced, using the EWP, from 17.17 L/min to 3.17 L/min, the components heat up faster, as seen in figure 6.5.

Although, the increase in the temperature of the bulk mass of oil is not significant, the block starts to run hotter than the head, as can be noticed in the first 50 seconds, in figure 6.5, that the difference in the temperature of the block is more than 5°C, which indicates that the block is warming up faster. Faster warm up of the block through reducing piston friction, leads to a lower fuel consumption penalty.
Once, the speed and load is applied on the engine, the fuel consumption rises very steeply and it takes around 25 seconds approximately for the fuel consumption to come down to normal. The fuel consumption increase may be related to the fast catalyst light off strategy. Once, the fuel consumption stabilizes, there is a reduction in fuel consumption from 50 -100 seconds, as the block gets heated up. The difference in temperature of the engine block by the end of 300 seconds is quite significant.

6.7.4 EWP (0 L/min)

The engine is run in the 0 L/min test, as shown in figure 6.6, and the coolant capacitance is considered as a trapped mass of coolant both in the cylinder head and in the engine block. The mass of coolant around the combustion chamber is subjected to the largest heat flux and has the highest temperatures, but the pockets of coolant away from the combustion chamber, have lower temperatures. This has been observed by the thermocouples placed in the cylinder head to measure the metal and coolant temperature around each cylinder. These results can be found in Appendix I (I.17 to I.19). The thermocouples used to measure the coolant temperature before thermostat and at the water pump inlet in the block recorded lower temperatures. The reason for this variation in temperatures is that the coolant is not flowing. Although, the engine reached its operating temperature, (when the thermostat opens) the temperature did not come down, because there was no flow in the engine or flow from the radiator.

Local Boiling or Nucleate Boiling: In the 0 L/min run, it can be noticed that the average temperature of coolant in the cylinder head becomes equal to the block temperature after 170 seconds approximately until the thermostat opens, at 220 seconds, but because there is no flow of coolant, it starts to rise again. Moreover, it reaches a stage when the temperature of the coolant becomes higher than the measured metal temperature. This is
the stage when nucleate boiling starts to occur in the head after 120 seconds approximately. Evidence of this is found in Appendix I (figure I.18). Due to no flow of coolant, there is likely some distortion of the metal surrounding the cylinders, leading to higher fuel consumption after 120 seconds, as seen in figure 6.6 and this will be presented in Section 6.10 in more detail. Because of local boiling or nucleate boiling occurring after 120 seconds approximately, there is in places, very high heat transfer from the metal to the coolant (in places where there was no temperature measurement) and as a result, the temperature of the coolant appears to become higher than the metal, after 140 seconds approximately. The coolant temperature surpasses the head temperature after 140 seconds approximately and exceeds the block temperature after 170 seconds approximately. It has been quoted in the literature that when nucleate boiling occurs, the metal temperature becomes independent of the liquid temperature flowing past it.

\footnote{Although the coolant temperature appears to be greater than the metal temperature at places of temperature measurement, the metal will be hotter than coolant.}
6.7.5 Split (3.17 L/min, flow in head)

The reader is advised to refer to figure to 6.7. In the first 100 seconds, there is significant reduction in fuel consumption, in the split (3.17 L/min, flow in head) as compared to 3.17 L/min EWP set-up, shown earlier in figure 6.5. The fuel consumption is also lower after 100 seconds, but it is not as significant as it is in the first 100 seconds.
6.7.6 Split (3.17 L/min, flow in block)

The reader is advised to refer to figure 6.8. There is a significant decrease in fuel consumption for the split (3.17 L/min, flow in block) as compared to the conventional EWP (3.17 L/min), shown in figure 6.5, up to the first 120 seconds and after this time the fuel consumption rate becomes approximately equal for both the running conditions.
6.8 Fuel Consumption

6.8.1 General Understanding of the Fuel Consumption

Table 6.5 below summarizes the fuel consumption values for different coolant flow conditions. The instantaneous values were shown in figures 6.3 to 6.8.
### Table 6.5: Average Fuel Consumption Rate

<table>
<thead>
<tr>
<th>Fuel Cons</th>
<th>Idle (9.17L/min)</th>
<th>MWP (17.17 L/min)</th>
<th>EWP (17.17 L/min)</th>
<th>EWP (3.17 L/min)</th>
<th>EWP (0 L/min)</th>
<th>Split (3.17 L/min, flow in head)</th>
<th>Split (3.17 L/min, flow in block)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time (s)</td>
<td>g/100 ms</td>
<td>g/100 ms</td>
<td>g/100 ms</td>
<td>g/100 ms</td>
<td>g/100 ms</td>
<td>g/100 ms</td>
<td>g/100 ms</td>
</tr>
<tr>
<td>50</td>
<td>0.048</td>
<td>0.075</td>
<td>0.079</td>
<td>0.078</td>
<td>0.072</td>
<td>0.071</td>
<td>0.070</td>
</tr>
<tr>
<td>100</td>
<td>0.041</td>
<td>0.062</td>
<td>0.053</td>
<td>0.062</td>
<td>0.061</td>
<td>0.062</td>
<td>0.060</td>
</tr>
<tr>
<td>150</td>
<td>0.032</td>
<td>0.061</td>
<td>0.057</td>
<td>0.057</td>
<td>0.059</td>
<td>0.056</td>
<td>0.058</td>
</tr>
<tr>
<td>200</td>
<td>0.026</td>
<td>0.064</td>
<td>0.054</td>
<td>0.054</td>
<td>N/A</td>
<td>0.055</td>
<td>0.056</td>
</tr>
<tr>
<td>300</td>
<td>0.026</td>
<td>0.057</td>
<td>0.054</td>
<td>0.054</td>
<td>N/A</td>
<td>0.054</td>
<td>0.056</td>
</tr>
</tbody>
</table>

**First 20 Seconds:** Just after cranking, the PCM gets disabled in every test, and therefore, in the first 5-8 seconds, there is no data recorded. The first 20 seconds is the idling period. When the engine is started, the speed is 850 rev/min approximately. Slowly, the engine gets stabilized and the speed tends to reduce, which also brings the fuel consumption down. Engine speed data are found in Appendix H.

**20-50 Seconds:** After 20 seconds, when the engine is subjected to increase in speed and load, the fuel consumption tends to rise, which is condition of ‘acceleration enrichment’. When sudden acceleration is applied to the vehicle, the fuel consumption rises and once the acceleration becomes steady, it tends to come back to stoichiometric (when the engine is running at steady state). One of the most important factors that can lead to this steep rise in fuel consumption is the inertia of the components connected to the
engine. In the actual vehicle, there is a gearbox, transmission shaft and the torque converter connected to the engine’s flywheel, which gives resistance to the turning of the flywheel and the drive shaft. On the contrary, in the experimental set-up, there is a dynamometer connected to the engine’s drive shaft, instead of the torque converter, gearbox and the transmission shaft. If the inertia of the dynamometer is considered approximately equal to the inertia of the transmission, transmission shaft and the torque converter, then the steep rise in the fuel consumption in the experiment should be approximately equal to the fuel consumption in actual driving conditions, as the engine is mapped for transient running. Nonetheless, the fuel consumption reduces in the next 25 seconds, after the speed and load is applied to the engine. It has also been stated earlier in section 6.7.3, this is related to engine’s catalyst light off strategy, running an initial rich mixture, once the engine is loaded.

- **50-100 Seconds:** There is a decrease in fuel consumption from 50-100 seconds. This is the phase when the engine block warms up and the piston friction reduces, which brings the fuel consumption down.

### 6.8.2 Fuel Consumption for different Conditions

- The results for the individual fuel consumption tests can be referred to figures 6.9 to 6.14.

- **Preliminary Idle:** When the engine is running at idle, it starts at 850 rev/min approximately and then it stabilizes and reduces to 700 rev/min, as it warms up. Hence, the fuel consumption is a function of spark advance in the idle run because there is not much variation in the engine speed.
It can be noticed in the results that the fuel consumption rises immediately after the first cranking occurs, due to initial overfuelling during cold start and then it starts to fall. The fuel consumption reduces in the first 20 seconds approximately and then it starts to rise again. This happens because spark begins to retard after 20 seconds of engine start. The spark retards for 20 seconds, i.e. till the end of 40 seconds approximately, after engine start and the fuel consumption rises with it. After 40 seconds, it can be noticed that the spark begins to advance and the fuel consumption starts coming down with the spark. The spark advances slowly up to 80 seconds and the rate of reduction in fuel consumption is also low up to this time. It is only at between 170 to 200 seconds approximately that the spark advances to its maximum value of 20° BTDC approximately and the fuel consumption reaches its lowest value.

- **Main Tests**: It can be noticed in the results for all conditions of coolant flow, in the loaded phase that the fuel consumption rate rises in the beginning, immediately, after the engine starts. It starts to come down in the first 20 seconds, similarly to the idle case. In the idle case, the fuel consumption starts to rise after 20 seconds, due to spark retard, whereas in the loaded run, the engine speed is ramped up with load after 20 seconds of engine run and the spark does not get retarded. The control system changes the cold start strategy and it forces the system out of the CSSRE (cold start spark retard for reduced emissions). A very steep fuel spike can be noticed here after 20 seconds, which only stabilizes in the next 15 seconds approximately. Nonetheless, this steep rise in fuel consumption is a control strategy during cold start and this is how the engine has been mapped for transient running conditions.
Figure 6.9: Comparison of total fuel consumption for repetitive tests for 17.17 L/min MWP

Figure 6.10: Comparison of total fuel consumption for repetitive tests for 17.17 L/min EWP
Figure 6.11: Comparison of total fuel consumption for repetitive tests for 3.17 L/min EWP

Figure 6.12: Comparison of total fuel consumption for repetitive tests for 0 L/min EWP
Figure 6.13: Comparison of total fuel consumption for repetitive tests for split (3.17 L/min, flow in head)

Figure 6.14: Comparison of total fuel consumption for repetitive tests for split (3.17 L/min, flow in block)
6.8.3 Fuel Summation Results

Fuel summation results plotted for half minute intervals in figure 6.15 show significant reduction in fuel consumption up to the first 120 seconds, as the coolant flow rate is reduced utilizing the electric water pump. The reasons for an increase in fuel consumption after the installation of electric water pump have been discussed earlier and it is not repeated again. However, the 17.17 L/min EWP set-up has been taken as a reference for comparing the fuel consumption, with other coolant flow rate conditions because the coolant capacitance is the same for all the tests performed with the electric water pump.

Nonetheless, it can be noticed that in the first 30-60 seconds, the fuel summation results show little improvement, when 17.17 L/min EWP and 3.17 L/min EWP are compared. However, the improvement in fuel consumption increases for the 0 L/min set-up and split cooling set-ups for both the cases of flow in the head and in the block compared with the 17.17 L/min EWP set-up in the first 30-60 seconds.

The fuel consumption benefits are still evident in the 60-90 second and 90-120 second periods, when 17.17 L/min EWP is compared to lower flow rates. However, the 0 L/min records higher fuel consumption between 90-120 seconds. The best fuel consumption rate has been observed for the split (flow in block 3.17 L/min) up to 120 seconds.
6.8.4 Reduction in Fuel Consumption

The reduction in fuel consumption has been calculated as the difference accumulated over successive half minute intervals. Figure 6.16 shows the percentage reduction in fuel consumption when 3.17 L/min EWP and 0 L/min EWP are compared to the base of 17.17 L/min. In the first two minutes of operation, a maximum of 6% approximately can be saved as compared to the normal flow conditions by having zero flow in the heater circuit. Figure 6.17 shows the % reduction in fuel consumption for the split flow in (block, 3.17 L/min), as compared to the non-split condition for three different coolant flow rates. The split (flow in block, 3.17 L/min) is selected for comparison because it records the least fuel consumption up to 120 seconds of the engine operation. Figure 6.17 presents the estimated reduction in fuel consumption for the conditions of 17.17 L/min, 3.17 L/min and 0 L/min, when they are compared to the condition with lowest fuel consumption, i.e split, flow in block, 3.17 L/min. In the first two minutes of
operation, the best fuel consumption reduction is 10% approximately with 3.17 L/min flow in the block compared to the normal 17.17 L/min flow in the engine as a whole.

Figure 6.16: Comparison of 3.17 L/min and 17.17 L/min (EWP) to 0 L/min EWP for % reduction in fuel consumption
6.9 Cylinder Head Temperatures for all Locations

6.9.1 Correction to Common Start Point

The temperature data of the thermocouples has been corrected to a common start time and temperature. The correction for temperature has been done by adding or subtracting a constant offset to ensure that all the thermocouples start at 20°C. The corrections are small in the range of ±2°C. Similarly, a correction has been done for time. Since, the data recorder was normally started, at slightly varying time, before the engine was started, the time series was corrected to zero at start.
6.9.2 Location of Thermocouples

Thermocouples are located at each cylinder for measuring the coolant temperature in the head. In addition to those around the gasket face for each of the cylinders, temperature is also measured before the thermostat housing, at the Inlet of the water pump and in the exhaust valve bridges. The following tables from 6.6 to 6.13 show the temperature rise with respect to time for different thermocouple locations.

Table 6.6: Temperature profile for Cylinder 1

<table>
<thead>
<tr>
<th>Cylinder 1</th>
<th>9.17 L/min Idle</th>
<th>17.17 L/min MWP</th>
<th>17.17 L/min EWP</th>
<th>3.17 L/min EWP</th>
<th>0 L/min EWP</th>
<th>Split (3.17 L/min, Flow in Head)</th>
<th>Split (3.17 L/min, Flow in Block)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time (s)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>100</td>
<td>35.6</td>
<td>45.2</td>
<td>36.9</td>
<td>51.7</td>
<td>71.7</td>
<td>46.5</td>
<td>74.5</td>
</tr>
<tr>
<td>200</td>
<td>49.5</td>
<td>60.6</td>
<td>52.0</td>
<td>64.3</td>
<td>103.1</td>
<td>62.8</td>
<td>102.8</td>
</tr>
<tr>
<td>300</td>
<td>57.9</td>
<td>72.9</td>
<td>64.1</td>
<td>76.9</td>
<td>N/A</td>
<td>76.2</td>
<td>N/A</td>
</tr>
<tr>
<td>400</td>
<td>64.2</td>
<td>82.6</td>
<td>76.0</td>
<td>87.2</td>
<td>N/A</td>
<td>88.1</td>
<td>N/A</td>
</tr>
<tr>
<td>500</td>
<td>71.0</td>
<td>92.7</td>
<td>85.0</td>
<td>97.2</td>
<td>N/A</td>
<td>99.7</td>
<td>N/A</td>
</tr>
<tr>
<td>600</td>
<td>76.8</td>
<td>97.7</td>
<td>92.3</td>
<td>99.7</td>
<td>N/A</td>
<td>99.3</td>
<td>N/A</td>
</tr>
</tbody>
</table>
### Table 6.7: Temperature profile for Cylinder 2

<table>
<thead>
<tr>
<th>Time (s)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>35.0</td>
<td>45.1</td>
<td>36.5</td>
<td>49.0</td>
<td>68.5</td>
<td>40.8</td>
<td>77.3</td>
</tr>
<tr>
<td>200</td>
<td>49.0</td>
<td>60.3</td>
<td>52.2</td>
<td>62.9</td>
<td>97.5</td>
<td>60.5</td>
<td>106.8</td>
</tr>
<tr>
<td>300</td>
<td>57.3</td>
<td>72.8</td>
<td>64.1</td>
<td>75.0</td>
<td>N/A</td>
<td>73.6</td>
<td>N/A</td>
</tr>
<tr>
<td>400</td>
<td>64.0</td>
<td>82.0</td>
<td>76.0</td>
<td>85.8</td>
<td>N/A</td>
<td>85.1</td>
<td>N/A</td>
</tr>
<tr>
<td>500</td>
<td>70.8</td>
<td>92.5</td>
<td>85.0</td>
<td>95.8</td>
<td>N/A</td>
<td>96.8</td>
<td>N/A</td>
</tr>
<tr>
<td>600</td>
<td>76.6</td>
<td>97.7</td>
<td>92.3</td>
<td>99.5</td>
<td>N/A</td>
<td>95.2</td>
<td>N/A</td>
</tr>
</tbody>
</table>

### Table 6.8: Temperature profile for Cylinder 3

<table>
<thead>
<tr>
<th>Time (s)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
<th>Temp (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>35.0</td>
<td>45.0</td>
<td>36.4</td>
<td>47.0</td>
<td>71.7</td>
<td>44.3</td>
<td>76.9</td>
</tr>
<tr>
<td>200</td>
<td>49.0</td>
<td>60.0</td>
<td>52.1</td>
<td>60.5</td>
<td>103.1</td>
<td>56.4</td>
<td>106.8</td>
</tr>
<tr>
<td>300</td>
<td>57.4</td>
<td>72.7</td>
<td>64.1</td>
<td>73.6</td>
<td>N/A</td>
<td>70.6</td>
<td>N/A</td>
</tr>
<tr>
<td>400</td>
<td>64.0</td>
<td>81.8</td>
<td>75.9</td>
<td>84.5</td>
<td>N/A</td>
<td>82.6</td>
<td>N/A</td>
</tr>
<tr>
<td>500</td>
<td>70.8</td>
<td>92.4</td>
<td>84.6</td>
<td>94.2</td>
<td>N/A</td>
<td>93.7</td>
<td>N/A</td>
</tr>
<tr>
<td>600</td>
<td>76.4</td>
<td>97.4</td>
<td>92.0</td>
<td>98.1</td>
<td>N/A</td>
<td>90.0</td>
<td>N/A</td>
</tr>
</tbody>
</table>
Table 6.9: Temperature profile for Cylinder 4

<table>
<thead>
<tr>
<th>Cylinder 4</th>
<th>9.17 L/min Idle</th>
<th>17.17 L/min MWP</th>
<th>17.17 L/min EWP</th>
<th>3.17 L/min EWP</th>
<th>0 L/min EWP</th>
<th>Split (3.17 L/min, Flow in Head)</th>
<th>Split (3.17 L/min, Flow in Block)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time (s)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>100</td>
<td>35.0</td>
<td>45.0</td>
<td>35.9</td>
<td>46.6</td>
<td>71.7</td>
<td>35.8</td>
<td>77.5</td>
</tr>
<tr>
<td>200</td>
<td>49.0</td>
<td>60.0</td>
<td>51.9</td>
<td>59.8</td>
<td>103.1</td>
<td>52.4</td>
<td>106.8</td>
</tr>
<tr>
<td>300</td>
<td>57.3</td>
<td>72.5</td>
<td>64.1</td>
<td>73.2</td>
<td>N/A</td>
<td>67.0</td>
<td>N/A</td>
</tr>
<tr>
<td>400</td>
<td>64.0</td>
<td>81.9</td>
<td>75.9</td>
<td>84.4</td>
<td>N/A</td>
<td>78.5</td>
<td>N/A</td>
</tr>
<tr>
<td>500</td>
<td>70.6</td>
<td>92.5</td>
<td>84.4</td>
<td>93.9</td>
<td>N/A</td>
<td>89.3</td>
<td>N/A</td>
</tr>
<tr>
<td>600</td>
<td>76.4</td>
<td>97.3</td>
<td>92.0</td>
<td>97.6</td>
<td>N/A</td>
<td>90.0</td>
<td>N/A</td>
</tr>
</tbody>
</table>

Table 6.10: Temperature profile for Cylinder 5

<table>
<thead>
<tr>
<th>Cylinder 5</th>
<th>9.17 L/min Idle</th>
<th>17.17 L/min MWP</th>
<th>17.17 L/min EWP</th>
<th>3.17 L/min EWP</th>
<th>0 L/min EWP</th>
<th>Split (3.17 L/min, Flow in Head)</th>
<th>Split (3.17 L/min, Flow in Block)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time (s)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>100</td>
<td>38.1</td>
<td>45.3</td>
<td>38.9</td>
<td>51.9</td>
<td>71.7</td>
<td>39.3</td>
<td>76.9</td>
</tr>
<tr>
<td>200</td>
<td>51.5</td>
<td>60.7</td>
<td>54.0</td>
<td>62.9</td>
<td>103.1</td>
<td>56.4</td>
<td>102.7</td>
</tr>
<tr>
<td>300</td>
<td>58.6</td>
<td>73.1</td>
<td>65.7</td>
<td>75.0</td>
<td>N/A</td>
<td>70.6</td>
<td>N/A</td>
</tr>
<tr>
<td>400</td>
<td>64.3</td>
<td>83.4</td>
<td>76.4</td>
<td>85.8</td>
<td>N/A</td>
<td>82.6</td>
<td>N/A</td>
</tr>
<tr>
<td>500</td>
<td>71.2</td>
<td>93.0</td>
<td>85.5</td>
<td>95.8</td>
<td>N/A</td>
<td>93.3</td>
<td>N/A</td>
</tr>
<tr>
<td>600</td>
<td>77.3</td>
<td>97.8</td>
<td>93.6</td>
<td>99.7</td>
<td>N/A</td>
<td>90.0</td>
<td>N/A</td>
</tr>
</tbody>
</table>
### Table 6.11: Temperature profile for Cylinder 6

<table>
<thead>
<tr>
<th>Cylinder 6</th>
<th>9.17 L/min</th>
<th>17.17 L/min</th>
<th>17.17 L/min</th>
<th>3.17 L/min</th>
<th>0 L/min</th>
<th>Split (3.17 L/min, Flow in Head)</th>
<th>Split (3.17 L/min, Flow in Block)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Idle</td>
<td>MWP</td>
<td>EWP</td>
<td>EWP</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Time (s)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>100</td>
<td>30.9</td>
<td>39.9</td>
<td>32.8</td>
<td>33.8</td>
<td>68.5</td>
<td>28.6</td>
<td>71.1</td>
</tr>
<tr>
<td>200</td>
<td>44.5</td>
<td>55.9</td>
<td>49.0</td>
<td>50.2</td>
<td>97.5</td>
<td>46.6</td>
<td>95.5</td>
</tr>
<tr>
<td>300</td>
<td>53.3</td>
<td>68.4</td>
<td>60.7</td>
<td>62.6</td>
<td>N/A</td>
<td>60.4</td>
<td>N/A</td>
</tr>
<tr>
<td>400</td>
<td>60.4</td>
<td>78.7</td>
<td>72.0</td>
<td>72.8</td>
<td>N/A</td>
<td>72.4</td>
<td>N/A</td>
</tr>
<tr>
<td>500</td>
<td>67.0</td>
<td>88.5</td>
<td>80.6</td>
<td>82.3</td>
<td>N/A</td>
<td>82.2</td>
<td>N/A</td>
</tr>
<tr>
<td>600</td>
<td>72.7</td>
<td>92.2</td>
<td>88.9</td>
<td>85.7</td>
<td>N/A</td>
<td>77.2</td>
<td>N/A</td>
</tr>
</tbody>
</table>

### Table 6.12: Temperature profile for Water Pump Inlet

<table>
<thead>
<tr>
<th>Water Pump Inlet</th>
<th>9.17 L/min</th>
<th>17.17 L/min</th>
<th>17.17 L/min</th>
<th>3.17 L/min</th>
<th>0 L/min</th>
<th>Split (3.17 L/min, Flow in Head)</th>
<th>Split (3.17 L/min, Flow in Block)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Idle</td>
<td>MWP</td>
<td>EWP</td>
<td>EWP</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Time (s)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>100</td>
<td>26.4</td>
<td>33.9</td>
<td>25.2</td>
<td>24.0</td>
<td>21.8</td>
<td>21.0</td>
<td>24.6</td>
</tr>
<tr>
<td>200</td>
<td>41.4</td>
<td>50.0</td>
<td>40.8</td>
<td>40.1</td>
<td>27.6</td>
<td>26.5</td>
<td>32.5</td>
</tr>
<tr>
<td>300</td>
<td>50.0</td>
<td>61.4</td>
<td>52.2</td>
<td>51.8</td>
<td>N/A</td>
<td>36.7</td>
<td>N/A</td>
</tr>
<tr>
<td>400</td>
<td>57.0</td>
<td>71.1</td>
<td>62.1</td>
<td>61.4</td>
<td>N/A</td>
<td>46.6</td>
<td>N/A</td>
</tr>
<tr>
<td>500</td>
<td>62.6</td>
<td>80.4</td>
<td>71.0</td>
<td>70.0</td>
<td>N/A</td>
<td>57.2</td>
<td>N/A</td>
</tr>
<tr>
<td>600</td>
<td>67.9</td>
<td>85.8</td>
<td>78.7</td>
<td>72.7</td>
<td>N/A</td>
<td>65.3</td>
<td>N/A</td>
</tr>
</tbody>
</table>
Table 6.13: Temperature profile for Thermostat

<table>
<thead>
<tr>
<th>Thermostat</th>
<th>9.17 L/min Idle</th>
<th>17.17 L/min MWP</th>
<th>17.17 L/min EWP</th>
<th>3.17 L/min EWP</th>
<th>0 L/min EWP</th>
<th>Split (3.17 L/min, Flow in Head)</th>
<th>Split (3.17 L/min, Flow in Block)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time (s)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
<td>Temp (°C)</td>
</tr>
<tr>
<td>100</td>
<td>35 N/A</td>
<td>36.9</td>
<td>47.0</td>
<td>31.7</td>
<td>44.8</td>
<td>29.6</td>
<td></td>
</tr>
<tr>
<td>200</td>
<td>48.6 N/A</td>
<td>52.2</td>
<td>60.5</td>
<td>60.7</td>
<td>60.9</td>
<td>50.2</td>
<td></td>
</tr>
<tr>
<td>300</td>
<td>57.0 N/A</td>
<td>64.1</td>
<td>73.6</td>
<td>N/A</td>
<td>73.6</td>
<td>N/A</td>
<td></td>
</tr>
<tr>
<td>400</td>
<td>63.4 N/A</td>
<td>76.2</td>
<td>84.5</td>
<td>N/A</td>
<td>85.1</td>
<td>N/A</td>
<td></td>
</tr>
<tr>
<td>500</td>
<td>70.4 N/A</td>
<td>85.3</td>
<td>93.9</td>
<td>N/A</td>
<td>95.1</td>
<td>N/A</td>
<td></td>
</tr>
<tr>
<td>600</td>
<td>76.1 N/A</td>
<td>92.9</td>
<td>97.6</td>
<td>N/A</td>
<td>99.3</td>
<td>N/A</td>
<td></td>
</tr>
</tbody>
</table>

6.9.3 Preliminary Test

The results are graphically presented in the figure 6.18. Numerical comparisons are found in the second column of the preceding tables 6.6 to 6.13. The temperatures of all thermocouples are lying close to each other, except for ‘cylinder 6’ and ‘Water Pump Inlet’. The coolant reaches the operating temperature of 95°C in approximately 1020 seconds.
6.9.4 Main Test - MWP (17.17 L/min)

The results are given in figure 6.19. The engine reaches the operating temperature in approximately 560 seconds.

Figure 6.18: Comparison of coolant thermocouples at Idle (9.17 L/min)
6.9.5 EWP (17.17 L/min)

The results are given in figure 6.20. The engine reaches the operating temperature of $95^\circ C$ in 600 seconds approximately. Although, the coolant flow rate is same as for the MWP, the warm up time and fuel consumption have slightly increased in this case, while using the EWP. This is due to two reasons:

- The capacitance is higher.
- The flow is higher in the first 20 seconds.

When the speed of the engine increased from idle (700 – 800 rev/min) to 1161 rev/min, the coolant flow rate increases from 9.17 L/min to 17.17 L/min approximately. Therefore, in the case of engine running with a MWP, a lower flow rate was observed for the first 20 seconds and then it increased after 20 seconds. However, in case of an EWP, as the pump ran
with an external power supply and it was started even before the engine was started, at the flow rate of 17.17 L/min. Consequently, there was faster flow in the first 20 seconds of idle running, which hinders the warm-up of the engine and led to more fuel consumption.

It has to be noted that the fall in temperature of the water pump inlet thermocouple is because of the increased length of hose from the bottom of the radiator to the EWP and from there to the MWP, causing addition of an extra hose, consequently, led to an increase in the heat capacitance of the coolant.

![Figure 6.20: Comparison of coolant thermocouples for 17.17 L/min EWP](image)

### 6.9.6 EWP (3.17 L/min)

Figure 6.21 shows the results for this running condition. The engine warm up time is seen to be decreased by 80 seconds approximately, by lowering
the coolant flow rate to 3.17 L/min. The engine reaches its operating temperature in 520 seconds approximately.

It has to be noted that the temperature of the water pump inlet further dropped because of the greater temperature reduction of the externally circulating coolant at the reduced flow in the engine. The temperature profiles of the coolant thermocouples in the head are more scattered and the difference between temperatures of the coolant in different cylinders has increased, because of the reduced flow, as compared to the previous cases with higher flows.

![Graph showing temperature profiles for different coolant thermocouples](image)

Figure 6.21: Comparison of coolant thermocouples for 3.17 L/min EWP

### 6.9.7 EWP (0 L/min)

The results are presented in figure 6.22. This is the case when the engine ran without any coolant flow. The trapped mass of coolant in the engine is
heated when the engine is running. The figure shows that the thermostat opens at 95°C approximately, but the coolant continues to absorb heat from the engine, because there is no flow of coolant. Taking into consideration the rising temperatures in the engine and to prevent the engine from any damage, the 0 L/min tests were only run for 270 seconds. It has to be noted that the engine reached the operating temperature in only 160 seconds approximately.

Since the engine is running with out any coolant circulation, the mass of coolant surrounding the cylinders is heated to a larger extent, while the mass of coolant away from the cylinders is cooler.

![Figure 6.22: Comparison of coolant thermocouples for 0 L/min EWP](image)

6.9.8 Split Cooling (Flow in Head 3.17 L/min)

The discussion that follows is based on figure 6.23. In the split (3.17 L/min, flow in head), one of the findings from the results of this set-up is that
cylinder 5 is not running as the hottest cylinder. The reason to this fall in temperature of cylinder 5 is that the coolant is fed to the engine through cylinder 6 and not through cylinder 1, as it was in the previous set-up. Moreover, the temperature profiles of all the cylinders are more scattered than in the conventional set-up with higher flow rates. The reduced flow has increased the difference between the coolant temperatures of different cylinders. The engine reached the operating temperature of $95^\circ C$ in 450 seconds approximately, which is a gain in engine warm up time, as compared to the EWP 3.17 L/min set up, where the warm-up time was 520 seconds approximately. One of the disadvantages associated with running the flow in head and the block with no flow, is that some boiling was observed in the engine block at higher temperatures.

6.9.9 Split Cooling (Flow in Block 3.17 L/min)

The results are presented in figure 6.24. The temperature profile of this set-up is similar to the 0 L/min flow EWP, except for the fact that the temperature of the coolant in the head becomes constant after reaching $110^\circ C$. The engine reached the operating temperature in 160 seconds approximately, which is the same as 0 L/min EWP set-up, but the temperature starts to become constant even though there was no flow in the head. The reason for the temperature becoming constant is the occurrence of boiling, which starts after the coolant temperature reaches 110 degrees. The thermostat temperature did not rise much as it was located remote from the cylinders, which is similar to the 0 L/min condition. One of the major advantages of running the flow in block was that no boiling is observed either in the head or block till the operating temperature of the engine was reached and therefore, it is safer to run a split system instead of 0 L/min EWP, because the warm-up time in these set-ups is approximately the same. In contrast, in the 0 L/min EWP, set up, local boiling starts to occur from 120 seconds, when the operating temperature of the engine is $80^\circ C$ approximately.
Figure 6.23: Comparison of coolant thermocouples for split (3.17 L/min flow in head)

Figure 6.24: Comparison of coolant thermocouples for split (3.17 L/min flow in block)
6.10 Biasing of Flow with Electric Water Pump

Non uniformity in the coolant flow was identified from the results. The reader is advised to refer to appendix I for further details on biasing of the flow and its effects on engine heat transfer.

6.11 Summary

In this chapter, the effects of reducing the coolant flow rate on the engine warm-up time and fuel consumption have been reported and discussed. The temperature difference of the components due to the variation in coolant flow rate has also been included.

The outcome of these analyses shows that there is a lot of potential in reducing the coolant flow rate of the engine. The flow rate in the experiments has been reduced from a maximum of 17.17 L/min to zero. However, these results show that heating up the coolant faster does not affect the bulk temperature of the oil. Nonetheless, there are still some benefits in reducing friction, due to faster heating of the block, which reduces piston frictional losses. Best fuel consumption reduction was 10% in the first two minutes of operation using the split flow system in the block.

Some unexpected effects like the biasing of the flow when using the electric water pump and its effects on the components are further discussed in Appendix I. In addition, discussion on the behaviour of the Ford controller (PCM) for spark advance, engine speed and MAP has been done in Appendix H.
Chapter 7 – Comparison of Theory and Measurement

7.1 Introduction

This chapter draws together the modeling and experimental results in the previous chapters, with a comparison of the modeling and experimental results. It was discussed earlier that the model has been calibrated for running conditions of 1161 rev/min and 48 Nm of load and validated for 700 rev/min and 0 Nm of load. Comparison has been made of the temperatures of the cylinder head, water pump inlet to the engine block, cylinder head coolant and engine oil. It should to be noted that since the block is not modeled as receiving heat directly from the combustion chamber, a more appropriate comparison is between the temperatures of the water pump inlet in the experiment, where the heat transfer is from the coolant to the block metal and the block temperature in the model.

7.2 Comparison of Results from Modeling and Experiment

7.2.1 Cylinder Head Temperature

The comparison of cylinder head temperature in figure 7.1 (a) and 7.1 (b) shows little deviation at the beginning in the first 50 seconds of engine
running in both conditions of speed and load. This is due to three main reasons:

(1) As per the drive cycle requirement, the experiment is run at idle for the first 20 seconds and then a speed of 1161 rev/min and a load of 48 Nm was applied to the engine, whereas, the model is being simulated for the condition of 1161 rev/min and 48 Nm of load from the start.

(2) In the experiments, some thermocouple locations were either far away from the combustion chamber or on the inlet side of the head. These locations are comparatively colder to the ones, which are nearer to the combustion chamber and it takes some time for the heat to be transferred from the hotter side to the colder side through conduction. Since an average temperature of the cylinder head is considered, a small downside kink is observed in the experimental result. However, in the present simulation of the model, the cylinder head is treated as a uniform average temperature, with no variations for different parts of the head.
Comparison of Experimental and Modeling results for Engine Speed 1161 rpm

Comparison of modeling and experimental results for idle speed

Fig 7.1 (a) Comparison of predicted and experimental cylinder head temperature for 1161 rev/min and 48 Nm Load

Fig 7.1 (b) Comparison of predicted and experimental cylinder head temperature for 700 rev/min and 0 Nm Load
7.2.2 Engine Block Temperature

The comparison is done in figure 7.2 (a) and 7.2 (b). In the present model under study, heat is transferred from the combustion chamber to the walls, walls to the coolant and from the coolant to the engine block, which is theoretically based on the open deck design. However, the test engine is a closed deck design and the block thermocouples’ locations in the experiments are at the engine block top deck surface, which directly receives heat from the combustion chamber and the cylinder head. Thus, the block surface temperature can not be compared to the predicted temperature of the engine block in the model.

Therefore, the thermocouple for measuring the water temperature at the inlet of the water pump has been selected for comparison. At this location, heat is transferred from the coolant to the engine block and the engine block experiences the same temperature as the coolant, only with a certain time delay. It can be noticed that the predicted results of the model are in agreement with the experimental results for the engine block temperature.
7.2.3 Engine Oil Temperature

The comparison of engine oil in figure 7.3 (a) and 7.3 (b) shows fair agreement between the predicted temperature of the model and the experimental data. However, there is a little deviation in the middle of the run, i.e., after approximately 100 seconds, as Figure 7.2 (b) shows that the temperature of the model does not rise as significantly as the experimental results, which is due to the heat from bearing friction not being included in the model used in this study.
Comparison of Experimental and Modeling results for Engine Speed 1161 rpm

Comparison of modeling and experimental results for idle speed

Fig 7.3 (a) Comparison of predicted and experimental engine oil temperature for 1161 rev/min and 48 Nm Load

Fig 7.3 (b) Comparison of predicted and experimental engine oil temperature for 700 rev/min and 0 Nm Load
7.2.4 Cylinder Head Coolant Temperature

There is considerable deviation in the predicted temperature of the cylinder head coolant, comparing the experimental data with the modeling results, in figure 7.4 (a) and 7.4 (b). This is due to the temperature in the experiments, being measured only on the exhaust side of the cylinder head, while in the present model under study, it is the overall average of the coolant temperature in the cylinder head.

![Comparison of Experimental and Modeling results for Engine Speed 1161 rpm](image)

Fig 7.4 (a) Comparison of predicted and experimental cylinder head coolant temperature for 1161 rev/min and 48 Nm Load
7.3 Summary

The modeling results have been compared with the experimental results in this chapter. The comparison of results for cylinder head temperature, engine block temperature, and engine oil temperature have shown fair agreement with each other, whereas, the cylinder head coolant temperature shows some variation, which is due to the temperature in the experiments being measured on the exhaust side of the head, which is higher than the overall average temperature of the coolant in the engine. The flow of coolant in the cylinder head particularly is extremely complex with high velocity small passages, where temperatures may become critical and lower flows elsewhere. The assumption of a hydraulic mean flow area is a limitation of the current model. However, the model is probably adequate to guide EWP speed for a closed loop control based on engine air flow (as a

Fig 7.4 (b) Comparison of predicted and experimental cylinder head coolant temperature for 700 rev/min and 0 Nm Load
surrogate for torque) and engine speed, which are readily available data in the engine management system.
Chapter 8 - Conclusions

8.1 Introduction

The major conclusions that are drawn from the research undertaken are included in this final chapter. The knowledge and direction gained from this research is then used to formulate recommendations including future research.

8.2 Conclusions

To date, the heat transfer and fuel consumption in a 6-cylinder engine is a very complex problem. Inspite of the best efforts done to optimize the performance of the engine, there is always a trade off between the performance of the engine and the cost of achieving that performance. This research gives an insight and knowledge of how the engine’s performance is affected when the coolant flow rate is reduced. Several conclusions can be drawn from the research undertaken. They are described below under different subheadings.

8.2.1 Coolant Flow Rate

The research undertaken on the current Ford I-6 engine shows that under the representative city cycle driving conditions, the coolant flow rate in the
engine, through the heater circuit, running cold with the thermostat closed was found to be 17.17 L/min. However, it has been verified experimentally that it can be decreased to a zero flow, to achieve a faster warm-up of the engine, thereby, bringing the fuel consumption down. This was achieved using an EWP, which allows variable speed and flow control of the coolant. However, the utilization of electric water pump is not the best solution, because the after market pump does not have vehicle’s lifetime proven durability. On the other hand, the mechanical water pump is more durable and a flow limiting valve could be an alternative for the electric water pump to control the flow rate.

8.2.2 Fuel Consumption
At the outset, it was not expected that the engine management system would produce a spike in fuel consumption on loading the engine to NEDC set point. This is likely a part of the catalyst light off strategy. Thus the study of fuel consumption changes was limited to after 30 seconds of operation. Considerable reduction in fuel consumption was shown with decreased coolant flow rate during the first two minutes of operation. Best reduction in fuel consumption was 6%, achieved with zero flow and 10%, achieved with a small flow in the block, in the split flow system, during the first two minutes of operation. However, after this time, the reductions are not very significant. It is during the initial cold start operation that the effects of cold oil and friction are most prominent on the engine’s performance and there is a benefit from the component’s quicker warm-up.

8.2.3 Split Cooling Supply
It has been experimentally verified that running the head and block at different temperatures help to reduce the engine warm-up time and fuel consumption. However, it is not easy to redesign the engine to run on split cooling supply for practical use on the road. Therefore, it follows that the
best practical alternative is to use a mechanical water pump, with a flow diverter valve for reducing the coolant flow rate.

8.2.4 Engine Oil Warm-up

Although, there has been significant reduction in fuel consumption and engine warm-up time by reducing the flow rate of the coolant in the engine, but there are no significant benefits achieved in raising the temperature of the bulk mass of oil in the oil sump. This is due to several reasons, but largely because when the oil is cold, it largely recirculates through the pump pressure relief valve, with only a small flow to the warming up regions of the engine. However, the faster heating of the engine block, as shown in the experiments, reduces the friction in the boundary lubricant between the piston and cylinder, thereby, reducing the fuel consumption.

8.2.5 Improvement in Warm-up Time

The two strategies for engine warm-up that give best reduced fuel consumption i.e., reduced flow and split cooling, were enabled to achieve faster warm-up of the engine. For the two best strategies, the time to thermostat open was reduced from 600 seconds (17.17 L/min flow with EWP) to 170 seconds (zero flow and small flow in block, in split flow). There are well known secondary benefits of faster warm-up such as reduced hydrocarbon emissions from the combustion chamber and faster catalyst light off. Such an investigation was beyond the scope of this study.

8.2.6 Engine Heat Transfer Modeling

Modeling engine heat transfer has been found to be a very useful tool in the prediction of engine heat transfer in transient conditions. The model used in this research gives a good understanding of the engine heat transfer for the engine, assuming that it behaves as six identical cylinders, under transient
cold start running conditions. However, further knowledge of the flow distribution around each cylinder and combustion chamber is needed to predict the non identical behaviour of the individual cylinders and cylinder head conditions of the I-6 engine. The modeling can also be used to gauge and optimize the performance of the engine, mainly for the coolant flow rate under different light load – speed conditions.

8.3 Recommendation for Future Work
There has been a lot done to improve the performance and fuel consumption of 6-cylinder engines. However, the present research shows that there are still several areas which can be investigated, in order to achieve a faster warm-up of the engine and to achieve reduction in fuel consumption.

8.3.1 Further Work in Modeling
The model discussed earlier is developed for heat transfer in an average cylinder of the Ford Falcon engine. Further work in modeling can be undertaken to make this model work for all the six cylinders in sequence, when modeling of the flow of water from one cylinder to another has been estimated. This was beyond the scope of this exploratory project.

8.3.2 Trapping Hot Coolant on Exhaust side of the Head
The machining of bosses done to install the thermocouples in the cylinder head can be further adapted to draw out coolant from the exhaust side of the head. This coolant can be fed into an additional rail or pipe fitted on the exhaust side of the engine and supplying it to an oil and water heat exchanger. This process can be used to heat up the oil because heating up the oil with the hottest coolant available from around the exhaust port bridge region would help reducing the warm-up time of the engine. Full potential of the warm-up strategy can not be exploited completely unless an
early oil warm-up is achieved. To attain the benefits of reducing friction, the need to heat up the oil faster is one of the next steps in this direction.

8.3.3 Investigate Thermal Shock

Due to the sudden mixing of hot and cold coolant in the water pump, when the thermostat opens, in close to a step response, there is a sudden thermal shock experienced by the hot engine components. Coolant at a temperature around 20°C or 30°C (when the radiator is warmed up by the heat generated from the engine) from the bottom of the radiator mixes with the heater return coolant and enters the engine block. The thermal shock not only potentially increases wear and tear of the hot moving components of the engine but may also generate unnecessary noise as clearances increase. An interesting opportunity in this direction is to minimize the thermal shock associated with the opening of the thermostat through a flow control valve replacing the mechanical thermostat.

8.3.4 Measuring In-cylinder Pressure

Machining was done to install the pressure transducers in cylinders 1 and 6, which can be used in future experiments to measure the in-cylinder pressure to precisely evaluate the change in mechanical efficiency during the cold start and warm-up.

8.3.5 Flow Diverter Valve

It has been concluded in this research that there is a lot of potential in reducing the warm-up time of the engine, with reducing the coolant flow rate. Although, the engine’s mechanical water pump delivers the least flow rate during the low engine speeds at start-up, the results show that the coolant flow rate can be further reduced. The normal flow measured in the engine is 17.17 L/min through the heater circuit, when the thermostat is
closed, which could be reduced to close to zero until the engine is warmed up.

A small electronically controlled flow diverter valve is a suitable option for controlling the coolant during cold starting. The valve can limit the coolant flow through the heater circuit. A suitable location for the installation of the valve can be the junction, at the meeting point of the thermostat and the heater circuit.

8.3.6 Modification

It has been concluded in the results that there is not much improvement in the oil temperature when the coolant flow rate is reduced. A much suspected reason is that there is lot of by-passing of the oil-ways in the block and head, when the oil is cold. The pressure relief valve in the oil pump diverts the oil from the oil galleries, since the pressure of the oil is already high because of high viscosity. Hence, a lot of energy is wasted by the oil pump to pump oil and almost 60-70% of the oil by-passes the galleries, when it is cold. This creates a lag in the oil temperature rise.

A suitable option is to modify the engine to make it a dry sump system and connect an external oil tank to the engine. The external tank would have a heater in it to heat up the oil. An external pump can be utilized to feed the oil directly into the crank case. Therefore, the engine can be fed with hot oil rather than cold oil at start. This investigation can be helpful to find out the effects of hot oil on the engine warm-up time and fuel consumption. If the effects are significant, then there can be some recommendations for design changes. The recommended changes would accommodate exhaust side high temperature coolant energy to heat up the oil faster. Figure 8.1 shows a schematic diagram to attach a heater equipped external oil tank.
8.3.7 Design Improvement

As shown in the experiments, the metal temperatures in the engine block at cylinder 6 are lower to other cylinders. This is due to more flow of coolant around cylinder 6. Therefore, there is still some room for improvement in the design of the water jacket, to reduce the size of the water holes in the block, which can be a step further in achieving a faster warm-up of the engine.

Figure 8.1: Schematic Diagram for External Oil Sump
Reference List


92. M. Tiller, Physical System Modeling in Modelica,
96. F. Keynejad, PhD Thesis under Completion, Department of Mechanical and Manufacturing Engineering, University of Melbourne, Melbourne, 2005.


102. P.K.Diwakar, Recent Advances in IC Engines-GDI Engines and Advanced Optical Diagnostic Techniques. ME690.


110. Variable Area Flowmeters (Data sheet – Series 10A4500)

   http://www.abb.us/cawp/usabb046/8b9ad09ae0ec7acbc1256f9d006b771b.aspx


Appendix A – Mixture Preparation

A.1 Mechanism of Fuel Transportation

Fuel enters the combustion chamber in the form of vapor, fuel droplets or a fuel film, because the injected fuel droplets either get converted into a fuel film or the fuel droplets evaporate or pass into the engine.

When the inlet valve opens, sometimes the velocity of air is not enough to break up the fuel film in the inlet port and it is carried into the combustion chamber. However, this fuel film may be further broken down into fuel droplets inside the combustion chamber, and some evaporates in the compression process. Furthermore, when the inlet valve closes, some part of the fuel film, present on the valve head gets squeezed. This fuel film either enters the combustion chamber in the form of fine droplets, where it gets evaporated during the compression stroke or remains in the inlet port.

Moreover, there is also some evaporation of fuel in the inlet port itself not only due to the higher temperature of the inlet port but also due to the back flow of the residual gases, from the combustion chamber\(^6\). Mixture preparation and its ill effects on cold start performance is analyzed and discussed in [23, 27]. Other factors contributing to excess fuel consumption are discussed by Sorrel and Stone [23] including the frictional losses associated with cold starting and also the seepage of liquid fuel into the oil sump, especially, when the engine is cold.

\(^6\) It should be noted that under part load conditions, at low MAP, there is initially flow of residual gas from the cylinder into the inlet port due to its lower pressure.
There have also been suggestions made by Sorrel and Stone [23] that the most promising means for improvement in the engine warm-up is to improve the mixture preparation and to reduce the thermal capacity of the engine and coolant system, so that during the warm-up phase the engine is capable of utilizing the heat attained from combustion for the purpose of quickly warming up the engine. Moreover, it has also been suggested to make use of the exhaust gas energy to decrease the fuel consumption, as the experimental results show that the exhaust gas temperature rise and approaches its steady state very quickly, compared to the coolant temperature, which might also be used to overcome a slight delay before the coolant temperature starts to rise.

The effects of mixture preparation have been captured by a high-speed camera, in a different study done by Shin et al [24]. The fuel burnt in the combustion chamber is not only the fuel injected in the current cycle but also the fuel left from previous cycle. The in-cylinder entry of the fuel contains both the liquid and vapor phase and the fuel enters in the form of droplets. However, all the fuel injected does not form the mixture for combustion and some part of it goes into small crevice regions where it is trapped until towards the end of power stroke and then flushed out with the burnt gas in the exhaust stroke. This unburnt fuel is the main reason for unburnt hydrocarbon emissions (UHC). This insufficient fuel vaporization is due to the cold combustion chamber and poor mixture preparation.

Sampson and Heywood [7] investigated the behavior of fuel in the first few cycles after the engine start-up. The authors chose two engines for comparison and it has been observed that the MAP, engine speed and the combustion quality begins to stabilize approximately by the seventh cycle after engine start-up. The authors found that when the engine is started, there is an initial over fueling (approximately up to 5 times the stoichiometric mixture, for the first two cycles) to facilitate the process of combustion, because at this time the flow of air is less and the evaporation of fuel is poor. Some of the extra fuel injected in the first few cycles, which is still lying either on the back of the valve or at the walls, is used later on.
and some of it leads to the formation of unburnt hydrocarbons. Figure A.1 explains the initial over-fueling in the first few cycles.

![Figure A.1: Mass of Fuel Injected and Fuel Required for Stoichiometric Mixture for Cylinder 1 of 4.6 L V8 showing initial over-fueling and then under-fueling [7]](image)

Shin et al [24] used a high-speed camera to view the process of injection in the port in a firing 4-valve SI engine under part load. The experimentations were carried out under warm-up conditions when the engine metal temperature is still low. The study describes the events from fuel injection to fuel actually been burnt and the importance of mixture preparation is realized. When the fuel enters the inlet port, there is a backflow from the cylinder to the inlet port, as soon as the inlet valve opens, because the pressure of the inlet manifold is lower than the cylinder. This flow is further enhanced, when there is substantial valve overlap and the exhaust valve is still open, thus, allowing the hot burnt gases to flow from the exhaust port, through the cylinder, into the inlet port. This process not only increases the tendency for the fuel to evaporate better but it also helps in a better mixture preparation. The back flow of gases occurring from the cylinder to the inlet port is important to be studied because it changes the thermal environment inside the inlet port. The back flow gases are at a higher temperature than
the fresh intake charge and it subsequently enhances the mixture preparation and leads to better evaporation of fuel. However, there are still some large fuel droplets, which do not enter the cylinder and remain in the inlet port either sticking to the valve stem or the back of the valve. In this study, the authors also approximately calculated the backflow volume using an engine simulation program and the distance of the fuel blown back depends on engine load. It is low at medium and high loads but substantial at low loads. Schurov and Collings [8] not only discussed the flow of air fuel mixture into the combustion chamber but also the stages when the back flow occurs. This study gives a very good understanding of the fuel behavior during cold start because not all the fuel injected into the port goes into the combustion chamber but a substantial part resides on the walls of the inlet port and goes into the combustion chamber only in subsequent cycles. As shown in the study, the back flow occurs in two stages, first when the inlet valve closes in the early compression stroke and secondly when the exhaust valve opens in the final stages of the exhaust stroke. The back flow of gases occurs due to the pressure difference between the combustion chamber and the inlet port. During the early stages of the compression stroke, the inlet valve closes after BDC to maximize the volumetric efficiency. But at low engine speeds, generally when the engine is just started, it decreases the charging characteristics, because the inertia of the intake charge is relatively low. The pressure in the inlet manifold and inlet port drops in comparison to the pressure in the combustion chamber, which is high due to the start of compression. The back flow reduces the volumetric efficiency to a small extent but the bigger advantage of the back flow is that it changes the thermal environment of the inlet port and helps fuel evaporation. The second type of back flow occurs when the there is a valve overlap period; i.e., when the exhaust valve closes late and before it is closed, the inlet valve opens. This creates a pressure difference between the inlet valve and the combustion chamber and due to this pressure difference, the back flow occurs until the exhaust valve is closed. This is
indeed helpful in a cold start operating condition. Figure A.2 shows possible intervals of mixture back flow.

![Figure A.2: Possible Intervals of Mixture Back Flow in the Inlet Port [8]](image)

It has been explained by Schurov and Collings [8] that there is significant temperature rise in the inlet port for a limited time, when the second back flow occurs; i.e., when the exhaust valve opens and the inlet valve is still not closed. The first back flow has more extended time frame, i.e., when the inlet valve opens. Although the time frame for this back flow is longer but the temperature change is not very high, so the consequences are not so pronounced. One of the other important results discussed in this study is the fall of temperature around the area in the inlet port where there are fuel droplets because when these fuel droplets vaporize, they cool the air and the temperature falls. Figure A.3 presents the temperature profile of the inlet port.
It was also reported by Schurov and Collings [8] that the injected fuel in general will reach the cylinder 1.2 seconds after engine cranking has been started, which means that there is no substantial amount of fuel mass in the cylinder, for a few cycles, prior to the completion of engine cranking, when firing occurs. This is because the injected fuel forms a film, whose mass depends on the temperature of the walls. The higher the temperature of the walls, the smaller is the film and the better is the fuel evaporation. This process of fuel film mass evaporation continues and tends to go towards equilibrium but this equilibrium is not permanent and the wall fuel film mass begins to decrease as the engine warms up through the boiling of the film mass and eventually drying up of the parts of the inlet port wall. During the injection of the fuel in the port, most of the fuel builds around a particular point in the port. This pool of fuel is moved into the combustion
chamber with the inflow of gas and this inflow occurs within 20 to 30 degrees of crank angle rotation, when there is maximum air flow. Schurov and Collings [8] discussed two sources of fuel evaporation in the inlet port. One is from the evaporation of the fuel film and other source is from the evaporation of the airborne fuel droplets. However, when the engine is running cold, there is no real fuel film evaporation and so the only source of evaporation is from the airborne fuel droplets. It is seen from the results that the biggest growth of fuel vapor concentration takes place in the region of the droplet flight during the closed inlet valve period and this concentration reaches its peak when valve overlap takes place and there is back flow of gas from the cylinder to the inlet port. By the time of $85^\circ$ to $90^\circ$ of the crank angle rotation, the bulk of vaporized fuel leaves the inlet port and enters the cylinder.

It has been found by Shayler et al [14] that at 2-3 seconds after cold starting, the discrepancy between the air/fuel ratio between inlet and exhaust is more than 3 air/fuel ratio units. This value drops to half an air/fuel ratio unit after 50 seconds of engine running. Due to the extra fuel used at the engine start, the flow rate of exhaust hydrocarbons is higher at the start. It reaches a peak within a few seconds and then experiences a decay, when the engine warms up. The reduction is most significant during the time 25 to 100 seconds. Heat transfer in the exhaust system is further discussed by Fu et al [98], where the authors discussed the effects of inlet condition, pipe geometry, external heat transfer and wall thickness on outlet temperature. Figure A.4 shows the results explained by Shayler et al [14]. The described extra fuel needed during cold start and warm up not only contributes to increased hydrocarbon emissions but also to extra fuel consumption.
A.2 Factors affecting Mixture Preparation

A.2.1 Cyclic Variability and Spark Retard

Cycle to cycle variations have effects on mixture preparation and consequently on fuel consumption and emissions during cold starting. In a study done on the effects of cycle to cycle variations on the engine’s performance by Russ et al [12], the authors focused their work in understanding the causes of cycle-to-cycle variations with retarded ignition. It has been investigated in this study that the major advantage of spark retard is that it increases the burned gas temperature on the exhaust side because the burned gas is not ideally expanded and does not perform as much work on the piston as with MBT timing. The disadvantage of spark retard discussed by the author is a very small fuel economy penalty and
increased cycle-to cycle variations. Cyclic variation can be improved by improving in-cylinder motion by introducing swirl and tumble. However, in terms of fuel economy and HC emissions, a limited spark retard can be helpful, because it increases the exhaust side burned gas temperature, which in turn will elevate the coolant temperature on the exhaust side during a transient cold start operation. Due to the spark retard, the combustion may be slower later in the expansion stroke and even incomplete before exhaust valve opening and the combustion appears to continue in the exhaust stroke and in the exhaust port leading to the high temperatures. The experimental results discussed by Russ et al [12] demonstrated that the exhaust temperature increases by $80 \degree C$ approximately, as the spark is retarded from $15 \degree$ BTDC to TDC at an air fuel ratio of 14.6:1. Figure A.5 shows the results for spark retard at various A/F ratios.

![Figure A.5: Cold Fluid Spark Retard Results at various A/F](image)

Engine, 1200 rev/min, 1.0 Bar BMEP, 20 $\degree C$ Fluid Temperature [12]
This study clearly indicates that the main cause of cyclic variations in a cold retarded operational engine is the variation in combustion phasing. The expansion ratio decreases rapidly and hence the combustion phasing of an individual cycle determines the cycle thermal efficiency and IMEP. As the combustion occurs close to EVO, very little expansion work is done and thus late burn has little impact on IMEP. The study of the flame regime also suggests that the flame moves away from the quenching limit and misfire in a retarded spark operation. It signifies that retarding spark to a small extent can be helpful in raising exhaust side gas temperature without causing problems of severe cyclic variations, thus raising the coolant temperature, which can be beneficial if a heat exchanger is used to heat up the oil from the coolant and vice versa.

The benefit of a limited spark retard is not only limited to elevating the exhaust side gas temperature, but it also has positive effects on emissions. The investigation reported by Choi et al [99] states that the mass of THC (total hydrocarbon) is reduced by about 40% with spark timing 7.8° ATDC CA during 15 seconds from the start of the engine. One reason for the reduction in THC concentration before CCC (close coupled catalyst) and the other reason is the reduction in the light off temperature of the catalyst. Looking at the data of THC concentration in the exhaust port [99], spark retard is inversely proportional to the flame speed and hence the duration of the main flame is longer. This gives sufficient time for the fuel wetted on the wall of the cylinder liner and piston crevice to evaporate and get burned, resulting in lesser THC emissions.

**A.2.2 Effects of Swirl and Tumble**

The effect of swirl and tumble flows generated in the cylinder during induction in a cold start engine may be slightly relevant to improving the fuel consumption, but it can’t be ignored as it helps in getting a good mixture preparation. As reported in the introduction, poor mixture preparation is one of the primary causes of fuel economy penalty and more emissions in a
cold start engine, the results in this study show that both swirling and tumbling motion enhance and improve fuel vaporization and fuel air mixing, resulting in a better performance both in terms of fuel economy and emissions. Another advantage of making use of swirling and tumbling motion is that the ignition timing can be delayed, due to faster burning, which allows more time for fuel vaporization and mixing [100].
Appendix B
Faster Warm-up Strategies

B.1 Development and Advancement of Engines

As the environmental standards get tougher year by year, the need to bring down the rate of other harmful pollutant gases like $NO_x$ and HC is increasing. The development and production of GDI engines has been a major breakthrough in this area of research. The development and recent research have not only brought the emissions under certain limits but they have also given a big boost to fuel economy, as improved and advanced control systems replace conventional mechanical systems.

A basic GDI (Gasoline Direct Injection) engine is an engine that operates in early injection, homogeneous, and stoichiometric state. However, a much more complex GDI engine can be one, which operates unthrottled, lean and stratified when running at low loads and stoichiometric and homogeneous at high loads. Unthrottled operation is important because the reduction in pumping losses increase the fuel economy. The complexity of this engine type, particularly in transition from one phase to another from full load or part load requires very advanced sensors and sophisticated control system. Advantages of recent GDI engines have been discussed in [101-103] and have been stated as one of the most promising solutions for cold start problems. Direct injection into the cylinder not only enables proper mixing
and control of the fuel air mixture but also it eliminates the fuelling delay. DIHC\textsuperscript{7} and DISC\textsuperscript{8} have been discussed and DIHC has been considered to be more advantageous over the latter in cold start operation, because it produces higher exhaust gas temperature. Tumble flow has been found to be more useful in the DIHC engines whereas swirl flow has more utility in the DISC engines and conventional SI engines.

**B.2 Downsizing the Engine for better Fuel Economy**

Watson et al [10] discussed the importance of downsizing the engine for better fuel efficiency and predicted that downsizing and technology could bring down the total fuel consumption, 10 to 15.3\% in small cars and 19 to 30\% in large cars. A significant contribution from downsizing at the same power output is the reduced friction and faster warm-up of the smaller engine matching the power of its larger counterpart.

**B.3 Varying Engine Parameters**

Trapy and Damiral [5] examined lubricant heat transfer as the most likely candidate to improve the engine warm-up process. The aspects that have been very keenly examined to improve engine warm-up are:

- Increasing the engine speed,
- Decreasing the oil quantity in the oil sump and
- To fit a heat exchanger between oil and coolant flow network.

Although the positive effects of decreasing the oil quantity are very small, engine speed has a great influence on the warm-up duration because the oil receives a larger amount of heat from friction in the journal bearings and the cylinder piston boundary lubrication at higher speeds. The third aspect of installing a heat exchanger was not fully explained in the study, but it can be the most important tool to expedite the engine warm-up process. By installing a heat exchanger between the oil and coolant flow network on the exhaust side of the engine, exhaust energy can be used to heat up the

\textsuperscript{7} Direct injection homogeneous charge
\textsuperscript{8} Direct injection stratified charge
coolant faster, which goes into the heat exchanger to heat up the oil and the oil further goes into the oil pump and then into the engine. The reason to fit the heat exchanger on the exhaust side is because the temperatures are highest in this region, resulting in higher coolant temperatures in this region. Not only can the heat exchanger be helpful in getting the engine warmed up faster but it can also be an additional sink for the heat in the oil, when the engine is overheated or running in extremely hot conditions.

Trapy and Damiral [5] also discussed in their study the engine thermal energy balance. Their experiments reveal that under cold start light load conditions, about 53\% of the total energy is transferred to the cylinder head and port surfaces and the remaining 47\% is used in overcoming frictional effects, doing effective work and in the exhaust (9\% effective work, 25\% exhaust and 13\% frictional losses). Out of the 53\% transferred to the combustion chamber walls, more than half of it goes in to the ambient as a waste, mostly in exhaust gas enthalpy and the rest of it is used to heat up the oil, engine metal parts and coolant. Figure B.1 explains an engine thermal balance.

![Figure B.1: Engine Thermal Balance and Heat Used during Warm-up (Engine Speed = 2000 rev/min and Torque = 10 Nm [5].](image-url)
B.4 Effects on UHC Emissions with Varying Coolant Temperature

Besides having effects on fuel consumption, the coolant temperature also affects UHC to a great extent, mainly during warm-up period as about 80% of the unburned hydrocarbons are released during this period considering the typical city driving conditions, which are relatively short trips. The main sources of these unburned hydrocarbons come from crevices, oil film, and deposits in the quench region etc., as discussed by Guillemot et al [61]. The unburned hydrocarbons from these sources are partially oxidized when they are mixed with high temperature gases during the expansion and exhaust strokes, which are directly affected by the coolant temperature. The engine temperature also plays an important role, because the changing piston and cylinder wall temperatures change the gas density and also the crevice volume.

To study the influence of coolant temperature on different engine mechanisms, Guillemot et al [61] developed a dual cooling system. A specially designed gasket was used to separate the engine block from the cylinder head. A mechanically driven water pump was used for the block and an electric water pump was used for supplying water in the head. Nevertheless, the author does not discuss the cost of the entire set up, if the benefit of reduction in fuel consumption was achieved at higher cost than the previous conventional thermostat. Thus, it is important to do more research and cut the cost.
Appendix C – Engine Modifications and Thermocouple Locations

C.1 Modifications in the Engine

The test engine was modified to install K-type thermocouples for the temperature measurement of different components of the test engine. The thermocouples used were the K-type miniature thermocouples. K-type is a base metal thermocouple system using Nickel alloy. These thermocouples are able to withstand a temperature of $1100^\circ C$ and the plastic end or plug can withstand a maximum temperature of $220^\circ C$. Moreover, these thermocouples can be bent easily so it is easier to install them at positions where it is difficult to find access. All these properties of the K-type thermocouples make them suitable for all sorts of industrial applications involving high pressure, high temperature, high vacuum and high vibrations. The thermocouple junction is located at the tip of the thermocouple and it is insulated from the sheath [104]. The thermocouples miniature plugs are connected to the data logging system by extension cables using a set of extension plugs and socket of the same type. The thermocouples used were 100 mm and 150 mm long depending upon their location on the engine.

On the test engine, there were 56 thermocouples installed for temperature measurement. The test engine was stripped down, with every component of
the engine taken apart. The cylinder head and the engine block were pulled apart and grooved in a milling machine to the author’s drawings, to fit the thermocouples in their right positions. The grooves were 4mm deep and 4mm wide to fit the thermocouples, which were 3mm in diameter. The grooves were filled with DEVCON⁹ and finished to smoothness. Assembly and disassembly of the engine followed the BA engine service manual provided by Ford Motor Co., Australia [105].

C.2 Thermocouple Location in Engine Block

C.2.1 Block Inlet and Exhaust Thermocouples

All the thermocouples were installed in locations recommended by Ford Motor Co., Australia for correlation with their previous experience. In the engine block, for measuring the metal temperature, there were 27 thermocouples used. Out of these 27 thermocouples, 4 were installed to measure the metal side temperature on the inlet side of the engine on cylinder number 1, 4, 5 and 6. The tip of the thermocouple was 60 mm away from the center of the cylinder and the thermocouples were 100 mm in length. Similarly, on the exhaust side, there were another 4 thermocouples for measuring the metal temperature on cylinder number 1, 4, 5 and 6. Once again, the tip of the thermocouple was 60 mm away from the center of the cylinder. However, these thermocouples were greater in length i.e. 150 mm, to keep them away from the exhaust manifold because there is higher heat transfer on the exhaust side and the other end of the thermocouples, which was made of plastic (withstanding a maximum temperature of 220°C) might have melted at high temperatures.

⁹ Devcon is a high technology titanium-reinforced epoxy, suitable for repair of machinery and equipments. It has a temperature resistance of 180°C and a compressive resistance of 1300 bar. It is resistant to all chemicals, acids, bases, solvents and alkalis.
C.2.2 Block Bore Thermocouples

There were 7 thermocouples installed in the metal region between each cylinder, in the middle of the cylinder. These thermocouples were also 150 mm in length because they were protruding out on the exhaust side of the block.

C.2.3 Block Circumference Thermocouples

There were 4 thermocouples each on the circumference of cylinder 4, 5 and 6, which made 12 thermocouples, altogether, installed to measure the metal temperature of the bore circumference. These thermocouples were placed at a position such that the tip of these thermocouples was 3 mm away from the edge of the bore. Therefore, the groove was made up to a point 3mm away from the bore. This is the location of the pressure seal on the head gasket. Due to this pressure seal, the head gasket seals to the surface, otherwise it would blow out under high temperature and pressure created inside the combustion chamber.

Since, the testing was only supposed to be done for cold start idle and low speed and load conditions, it was decided to go ahead with the testing because the gasket would not blow up under lighter running conditions.

Figure C.1 shows the thermocouples locations on engine block.
C.3 Thermocouple Location in the Cylinder Head

C.3.1 Head Inlet and Exhaust Thermocouples

There were 21 thermocouples in the cylinder head. For the thermocouple positions on the inlet and exhaust side, all the positions were duplicated as in the block and there were 4 thermocouples on the inlet side and 4 thermocouples on the exhaust side of the cylinder head.
C.3.2 Head Bore Thermocouples
There were 7 thermocouples installed in the metal region between each cylinder, similar to the bore thermocouples in the engine block.

C.3.3 Head Exhaust Valve Bridge Thermocouples
In addition to the above thermocouple positions, there were 4 additional thermocouple positions in the exhaust valve bridge, which were 150 mm in length. The thermocouples positioned in the exhaust valve bridge were at cylinders 1, 4, 5 and 6. A hole was drilled down on the exhaust side of the cylinder head up to the centre of the exhaust port. The drilling of the hole was started 3 mm below the metal surface, and special care was taken, so that the drill does not pierce the water jacket. Moreover, the hole was drilled at an angle upwards, and the hole was intended to end just below the centre of the exhaust valve bridge so that the temperature measured was the temperature of the metal region just below the exhaust valve bridge. Figure C.2 shows the thermocouples locations on cylinder head.
C.4 Thermocouple Locations for Cylinder Head Coolant

Apart from the arrangement mentioned earlier, there were thermocouples installed on the exhaust side of the cylinder head to measure the coolant temperature in the head. Just near the exhaust side opening, where the exhaust manifold starts, holes were drilled in the water jacket of each cylinder. Thermocouples were installed on each cylinder from number 1 to 6. The thermocouples were installed with Swagelok fittings not only to prevent any leakage of the coolant, when the coolant is flowing but also to have an option of closing the holes whenever required. The thermocouple holes drilled here would serve a dual purpose. Firstly, thermocouples can
be installed to measure temperature of the coolant and secondly, they would form an outlet, to collect the coolant at its highest temperature. The purpose of collecting the hot coolant is to have an understanding of the temperature of the coolant in this region and explore the possibility of installing an oil and water heat exchanger on the exhaust side to make best use of the heat from this coolant, which is a future work suggested in this study in chapter 9. Figure C.3 represents the thermocouples locations for measuring the coolant temperature in the cylinder head.

![Figure C.3: Thermocouple location for exhaust side coolant temperature](image)

**C.5 Thermocouple Location on Coolant Inlet and Outlet**
A thermocouple was installed to measure the coolant temperature at the water pump where the coolant enters the block. A hole was drilled into the block with Swagelok fitting to hold the thermocouple without any leakage of coolant.
Another thermocouple location was where the coolant leaves the engine in the head, i.e., just before the thermostat. This thermocouple location was made by drilling a hole in the heater pipe, because the coolant flow is diverted into the heater pipe, when the thermostat is closed. Figure C.4 and figure C.5 give an understanding about the installation of thermocouple before thermostat and before water pump inlet respectively.

Figure C.4: Thermocouple location before thermostat (coolant temperature at outlet from the engine)
C.6 Thermocouple Extension Cable
As the thermocouples were not long enough to reach the data acquisition junction box (thermocouple module), they were connected to the junction box via extension cables. The extension cables were made by selecting the correct length of cable and connecting the thermocouple plug and socket at two ends of it. One end of this extension cable, from the socket side was connected to the thermocouple and the other end is connected to the data acquisition junction box.

C.7 Different Group of Thermocouples
Since the number of ATI channels was limited, the total number of 56 thermocouples were divided into 4 groups, with 4 thermocouples repeated in every test to check the repeatability of the results. However, two more
thermocouple modules were arranged, in the later stage of the experiment, before the installation of the electric water pump from Ford Motor Co., Australia. Hence, it became possible to log all the thermocouples data at the same time.

C.8 Installation of Air Bleed

To bleed the air pockets formed in the engine, an air bleed was installed in the heater circuit, which ended at the header tank. The idea behind installing the air bleed at this location is because this is the highest point on the engine to where an air pocket could rise. At the start of the engine, the thermostat is closed and the flow is diverted through the heater circuit. Therefore, any air pocket or bubble inside the coolant stream would easily bleed out through the header tank. Figure C.6 shows the installation of the air bleed on the thermostat housing, going up to the header tank of the engine.

Figure C.6: Installation of air bleed
C.9 Installation of Electric Water Pump

Once, the flow rate of the coolant was measured, the experimentation moved further in the next phase, which involved the installation of an electric water pump. The electric water pump was installed in series with the mechanical water pump. The mechanical water pump was made redundant by taking the impellor off from it. It was not possible to take the mechanical pump out, even though it was redundant because the serpentine belt that runs on it. So, the mechanical pump was still in the circuit but it did no work.

Moreover, there were also some more changes in the heater circuit with the installation of the electric water pump. As the engine runs with the thermostat closed, the only flow in the engine is through the heater circuit. Therefore, the heater circuit is connected just before the pump in the mechanical water pump assembly. To make the set-up work in the same way, the heater circuit was extended to the electric water pump and in the process, the capacitance of the coolant was increased because of the added hoses to the heater circuit. Figure C.7 shows the installation of the electric water pump and the additional hoses required to do it.
C.10 Experimental Set-up for Split Cooling Supply

Once, the baseline testing for engine warm-up was completed, the engine was again taken apart to make it workable for split cooling exercise. The idea behind running the engine with two cooling supplies is to monitor the change or improvement in warm-up time and fuel consumption as compared to the previous results.

C.10.1 Modifications for First Part of Split cooling

The engine’s head was taken apart and the water holes in the head were blocked with DEVCON. One of the Welsh Plugs on the transmission side of the engine’s cylinder head was taken out and made into an inlet for coolant in the cylinder head. Furthermore, this inlet was connected to the electric water pump, with a long hose running from the electric water pump on the other side of the engine. The electric water pump fed coolant to the cylinder
head at one end and the coolant outlet was on the other end, i.e., where the thermostat resided. The coolant would start to flow in the radiator once the thermostat is opened, otherwise it would be diverted through the heater circuit.

The electric water pump was connected to the cylinder head, therefore, first part of the strategy was to run the engine block unpressurized, without any flow of coolant, open to the atmosphere. The idea was to feed the coolant in the block through an open source, without connecting it to any source of pumping to run the block un-pressurized. To bleed the air out of the block, a tiny hole was drilled on one of the top corners of the block, right into the water jacket. A 3/4 mm thick plastic tube was fitted into this hole with a Swagelok fitting. Any coolant that was fed from one side would fill up the block and all the air would rise up and come out through this small tube. On the other hand, the cylinder head was connected to the electric water pump and it was fed with the least flow of coolant, i.e., 3.17 L/min, measured in the earlier part of experimentation. Figure C.8 shows the modified engine for split cooling (3.17 L/min of coolant flow in the head and no flow in the block).
C.10.2 Modifications for Second Part of Split Cooling

The second part of the split cooling strategy involved running the cylinder head un-pressurized, without any flow, open to atmosphere. The Welsh plug which was used to feed coolant in the head and was connected to the electric water pump was opened to atmosphere and coolant was fed into the cylinder head till the cylinder head was completely filled. The air bleed system installed just before the thermostat bled all the air out through the header tank. The hose coming from the thermostat to the radiator was blocked by putting a solid piece of steel of the same diameter and then putting a hose clamp on it. The bottom end of the radiator was connected to the water pump, which fed the minimum amount of coolant, i.e., 3.17 L/min into the block. One of the Welsh plugs was also taken out from the block and it formed an outlet for the coolant. This outlet of the coolant was connected to the heater hose, going back to the water pump. Therefore, the only flow in the block is through the heater circuit as the upper end of the

Figure C.8: Split cooling supply with 3.17 L/min flow in head and 0 L/min flow in block
radiator was blocked. Figure C.9 shows the modified engine for split cooling (3.17 L/min of coolant flow in the block and no flow in the head).

Figure C.9: Split cooling supply with 3.17 L/min flow in block and 0 L/min flow in head

The table C.1 shown below explains the split cooling strategy:

Table C.1: Split cooling strategy

<table>
<thead>
<tr>
<th></th>
<th>Split Cooling strategy (part 1)</th>
<th>Split Cooling strategy (part 2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylinder head</td>
<td>3.17 L/min of flow</td>
<td>No flow</td>
</tr>
<tr>
<td>Engine block</td>
<td>No flow</td>
<td>3.17 L/min of flow</td>
</tr>
</tbody>
</table>
Appendix D – Engine Wiring and Data Acquisition System

D.1 Engine Wiring

The author was responsible for the engine wiring, which was done in correspondence to the Ford Falcon wiring. Components not required in the experimental set-up were ignored and not connected, otherwise the complete car wiring diagram was followed to make the engine wiring set-up. Complete details about engine wiring were referred to the Ford Wiring Manual [106].

D.1.1 Engine PCM Wiring

The engine control unit (ECU) used for the engine is the Ford ECU normally called engine power-train control module (PCM) because it is used to control the whole gear-box and not just the engine. The engine PCM contained three pockets, with wires running through these pockets, namely Pocket ‘A’, Pocket ‘B’ and Pocket ‘C’. Pocket ‘A’ and Pocket ‘C’ were connected to the engine harness through in-built sockets in the harness. Through the Pocket ‘C’ connector, there were connections to the ATI via two terminals. Through the Pocket ‘B’ connector, the following connections were made:

- Throttle pedal, which had five circuits.
• Alternator, which had two circuits.
• Broadband manifold solenoid, through two circuits.
• Fuel tank through a relay.
• Fan 1 and 2 through two different relays.

D.1.2 Mains through a 12 Volts Battery

The main supply was provided by a 12 volts battery, from where a main power cable was connected to the fuse box. This power cable supplied current to all the wires, which were running through the fuse box via a current loop. The wires which were powered through this power cable were:

• Ignition Switch
• Starter Motor
• Fuel Pump
• Fan 1 and 2 of the Radiator
• HEGO sensor

These wires were going to separate relays provided beside the engine PCM. From the fuse box, there was a battery power wire feed to the engine PCM, and also to the main ignition switch at the engine control side. From the engine control side, the ignition returned wire returns power to the ignition relay. Therefore, on the ignition switch, the battery power wire was on the feed side, while on the switched side, it was returning power to the ignition relay. The switched side was also connected to another momentary (cranking) switch, which provided power to this switch and the switched side of this momentary switch returned power to the crank relay.

D.1.3 Ignition Relay

The positive side of the relay was powered from the ignition switch and after the circuit becoming a closed circuit, i.e., when the relay was triggered, it used to operate through the battery.
D.1.4 Radiator and Fan Assembly

To create the most representative dynamic conditions, a car radiator, pressurized header tank and a thermatic fan were used. There were two circuits that ran from the two fans to separate relays. The positive side of the fan relays drew power from the ignition relays and when the circuit was open, i.e., when the relay was triggered, the battery used to operate the fans. Figure D.1 shows the radiator with fan assembly and fuel tank.

![Figure D.1: Radiator and Fan Assembly and Fuel Tank](image)

D.1.5 Fuel Tank

The engine’s fuel lines coming from the fuel rail were disconnected and a separate fuel tank was used to supply fuel to the engine.

D.2 Dynamometer

The dynamometer used in this study was an eddy current, Heenan and Froude Dynamic Dynamometer. The dynamometer was used in the first quadrant speed control mode, enabling fixed speed testing. Instructions for using the dynamometer were referred to its instruction manual [107].
D.3 Data Acquisition and Instrumentation

D.3.1 Data Acquisition System

The data acquisition system used in the experimentation was the ATI systems (Accurate Technologies Inc.). The complete Data Acquisition kit consisted of the following modules:

- 3 Thermocouple modules.
- 1 Analog output module.
- 2 Can-Bus.
- ATI Hub.

The Hub was the main Data Acquisition device, which was connected to the computer through the USB port of the computer. The signal from the thermocouple was amplified by the thermocouple module, this amplified signal was further converted from a voltage to a digital output by the analog output module and the signal traveled through the Can-Bus. The digital form of the signal was read by the computer through the Main Hub. Figure D.2 shows the ATI Hub and thermocouple modules.

Figure D.2: ATI Vision Hub and Thermocouples Modules
D.3.2 Instrumentation

A Pentium PC was used to store and display data. A number of variables were measured by the Ford PCM and the signals transmitted to the computer by the DAQ system. The variables measured were:

- Temperature of 48 thermocouples measured by 3 thermocouple modules (16 thermocouples channels on each module).
- Engine coolant temperature measured from the engine sensor.
- Engine oil temperature measured from the engine sensor.
- Air charge temperature.
- Spark advance.
- Manifold absolute pressure (MAP).
- Engine speed.
- Fuel consumption.
- A/F ratio measured by the white band sensor or lambda sensor.
- Torque measured by the dynamometer load cell.
Appendix E – Pressure Transducer and Flow Meter

E.1 Pressure Transducer

The cylinder was adapted so that two pressure transducers could be installed in the engine, to measure the cylinder pressure, one in cylinder 1 and the other one in cylinder 6. The purpose of the pressure sensor was to calculate the friction. Generating an IMEP trace for each cycle, it is possible to calculate the FMEP by subtracting BMEP from IMEP. To install the pressure transducer, a hole was drilled on the sides of cylinder 1 and 6. The hole opened inside the combustion chamber in a region just below the centre of the inlet and exhaust valves. There were threads to mount the housing for the pressure transducer and the pressure transducer was screwed inside this housing. The pressure transducer location that was selected is called flush mounting. The pressure transducers used are manufactured by Kistler Co. Pty. Ltd and the model is called 601B and 603B, as discussed in the Kistler operating instruction manual [108]. However, both the pressure transducer cavities were plugged later with a blanking plug, as the installation was only done for future work and not for the current experimentation. Figure E.1 shows the hole drilled for the installation of the pressure transducer.
The pressure transducer generally installed in the combustion chamber gives inaccurate data due to thermal shocks from the passage of the flame across the active face of the transducer. The temperature shock results in contraction and expansion of the diaphragm and consequently changes the force acting on the quartz in the pressure transducer. This has been analyzed and explained in a study done by Lee et al [109].

**E.2 Flow Meter**

The flow meter used to measure the flow rate of the coolant was installed in the heater circuit. It was a simple rotameter, which measured the flow velocity in the system and provided a reading on the scale of the device. The value on the scale was in percentage of the total flow rate, which the device could measure. With the manufacturer’s specification of maximum
flow rate, for which the device was manufactured, the reading was then converted into an actual value in L/min.

The flow meter was installed in the heater circuit because when the engine is initially started, the thermostat is closed and the only flow in the engine is through the heater circuit. The idea was to measure the maximum coolant flow rate at the engine operating parameters of 1161 rev/min and 48 Nm of load, which has been found out to be 17.17L/min. This flow rate was treated as the baseline flow rate, which was the maximum flow rate in the engine at the engine operating parameters of 1161 rev/min and 48 Nm of load. However, the installation of the flow meter caused a small increase in the capacitance of coolant in the circuit, which consequently increased the warm-up time to a small extent.

To find the complete explanation to the manufacturer’s specifications of the flow meter, and to have an understanding about the Australian standards, the reader is advised to refer to [110] and [111] respectively. The following figure E.2 shows the variable area flow meter, 10A4500.
Coolant flow rate data was recorded for various speed and load conditions, which has been included in the following section.

**E.2.1 Flow Rate at different Speeds**

The flow rate data was recorded at various speed and load conditions. Table E.1 shows the flow rate data for thermostat closed.
Table E.1: Flow meter reading on different speed and load

<table>
<thead>
<tr>
<th>Engine speed (rev/min)</th>
<th>Load (Nm)</th>
<th>Flow meter Reading (%)</th>
<th>Flow meter Reading (L/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Idle (700)</td>
<td>0</td>
<td>59</td>
<td>9.33</td>
</tr>
<tr>
<td>900</td>
<td>15</td>
<td>73</td>
<td>12.16</td>
</tr>
<tr>
<td>1000</td>
<td>20</td>
<td>84</td>
<td>14</td>
</tr>
<tr>
<td>1100</td>
<td>20</td>
<td>89</td>
<td>15.83</td>
</tr>
<tr>
<td>1150</td>
<td>48</td>
<td>103</td>
<td>17.17</td>
</tr>
</tbody>
</table>

Figure E.3 shows the coolant flow rate with respect to engine speed, with the thermostat closed.

Figure E.3: Coolant Flow rate for different speed and load (thermostat closed)
E.2.2 Flow Rate Data supplied by the Manufacturer

However, Ford Motor Co., Australia supplied information about the coolant flow rate under steady state conditions with thermostat open [3]. Figure E.4 presents the details.

![Coolant Flow Rate Data](image)

Figure E.4: Coolant Flow rate data supplied by Ford Motor Co. (Thermostat open) [3]

E.3 Experimental Error Estimates

The following table E.2 shows a list of probable errors based on the instrumentation accuracy and observed repeatability. The errors have been estimated by observation in the repeatability of the tests under same running conditions.
<table>
<thead>
<tr>
<th>Variable</th>
<th>Unit</th>
<th>Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed</td>
<td>rev/min</td>
<td>± 2</td>
</tr>
<tr>
<td>Torque</td>
<td>Nm</td>
<td>± 3</td>
</tr>
<tr>
<td>Fuel flow</td>
<td>g/100msec</td>
<td>± 0.5%</td>
</tr>
<tr>
<td>Coolant</td>
<td>L/min</td>
<td>± 0.05</td>
</tr>
<tr>
<td>Temperature</td>
<td>°C</td>
<td>± 0.5</td>
</tr>
<tr>
<td>Time</td>
<td>s</td>
<td>± 0.05</td>
</tr>
<tr>
<td>MAP</td>
<td>Pa</td>
<td>± 0.4</td>
</tr>
<tr>
<td>Spark Advance</td>
<td>deg</td>
<td>± 0.3</td>
</tr>
</tbody>
</table>
Appendix F – Experimental Strategy and Procedure

F.1 Initial Plan for Experimentation

The initial plan for experimentation was to run the engine on a simulated NEDC (New European Drive cycle) for variable load and speed, but this facility was not available with the current dynamometer (Henan and Froude dynamic dynamometer), which is available in the research laboratory of The University of Melbourne. Furthermore, it was not easily possible to make this dynamometer adaptive of the changes without delaying the start of experimentation. Moreover, previous experience in the laboratory indicated that dynamic torque and speed control tended to add extra variability to engine test results compared with steady state conditions.

Therefore, the experimentation strategy was changed to perform the experiments at fixed load and speed, which the current dynamometer allowed. The idea was to run the engine at fixed load and speed, representing the calculated average of the speed and load of the engine running over the urban part of the simulated NEDC for 180 seconds. It was decided in the current strategy to run the engine at idle for the first 20 seconds and then increase the load and speed up to the set point of 1161 rev/min and 48 Nm of load, which was the average calculated speed and load.

The experimental procedure was divided into 4 stages, they were:
• Preliminary tests: To understand the default calibration of the Ford PCM and also to check the working conditions of all the thermocouples.
• Main tests: Main experiments were performed at the average speed of 1161 rev/min and 48 Nm load, data was recorded and the experiments were repeated 4 times to check the repeatability of the results.
• Variable flow rate testing with EWP: The engine was run at variable flow rate of coolant with the utilization of an electric water pump. The idea was to monitor the effects on engine heat transfer and warm-up time, with varying coolant flow rate, from maximum to zero.
• Split cooling system: To make a split cooling supply for engine block and cylinder head and investigate the effects on engine heat transfer and warm-up time.

F.2 New European Drive Cycle (NEDC) [1]
NEDC is the New European Drive Cycle. This is a combined ECE and EUDC test cycle, which is performed on the chasis dynamometer. This cycle is also known as MVEG-A cycle and is used for emission certification of light duty vehicles in Europe. The first segment of the cycle is the ECE segment, which is repeated without interruption. Before the test, the vehicle is allowed to soak for 4 to 6 hours at a temperature of 20°C to 30°C and then allowed to run at idle for 40 seconds after starting. The ECE is also known as UDC (urban drive cycle) and it represents city driving conditions. This cycle is characterized by low emissions temperature, low vehicle speed and low engine load.
After the year 2000, the idling period was eliminated and the emissions sampling begin as the engine is started. This modified cold start emission cycle is called the NEDC. The EUDC segment has been added as a fourth segment to the ECE, which accounts for heavy driving conditions. The maximum speed limit for the EUDC is 120 Km/h for the vehicles generating higher power but for low power vehicles, the maximum speed limit, which has been set, is 90Km/h.
F.3 Preliminary Testing

In this phase of testing, the engine was run at idle (no load) and the thermocouples were divided into 4 groups, with 4 thermocouples repeated in every group. The radiator fan (low speed fan) was always turned on because the engine was running in the guard mode and the PCM turned the fan on. The reason to the engine running in the guard mode is the absence of AC. The PCM reads the AC pressure lower than the appropriate value, as related to the engine speed and it goes in the guard mode and turns the low speed fan on. The troubleshooting of the engine faults was referred to OBD –II Diagnostic Trouble Code Definitions [112].

Although, the data was affected, as the engine was running colder than normal because of the radiator fan being turned on all the time during the experiment, but, as the idle phase of testing was only to understand the default calibration and the working condition of the thermocouples, this factor was overlooked.
F.3.1 Thermocouples in the First Group

Table F.1: First Group Thermocouples in Preliminary Tests

<table>
<thead>
<tr>
<th>Channel</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Water Pump Inlet</td>
</tr>
<tr>
<td>2</td>
<td>BB4</td>
</tr>
<tr>
<td>3</td>
<td>HV4</td>
</tr>
<tr>
<td>4</td>
<td>HE4</td>
</tr>
<tr>
<td>5</td>
<td>BB5</td>
</tr>
<tr>
<td>6</td>
<td>BC43</td>
</tr>
<tr>
<td>7</td>
<td>Water Cylinder 6</td>
</tr>
<tr>
<td>8</td>
<td>HI4</td>
</tr>
<tr>
<td>9</td>
<td>HV1</td>
</tr>
<tr>
<td>10</td>
<td>BE1 (Repeat)</td>
</tr>
<tr>
<td>11</td>
<td>BB3</td>
</tr>
<tr>
<td>12</td>
<td>BC62 (Repeat)</td>
</tr>
<tr>
<td>13</td>
<td>HB1 (Repeat)</td>
</tr>
<tr>
<td>14</td>
<td>BC41</td>
</tr>
<tr>
<td>15</td>
<td>HB7 (Repeat)</td>
</tr>
<tr>
<td>16</td>
<td>HB4</td>
</tr>
</tbody>
</table>
F.3.2 Thermocouples in the Second Group

Table F.2: Second Group Thermocouples in Preliminary Tests

<table>
<thead>
<tr>
<th>Channel 1</th>
<th>BC42</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel 2</td>
<td>HI6</td>
</tr>
<tr>
<td>Channel 3</td>
<td>HV5</td>
</tr>
<tr>
<td>Channel 4</td>
<td>BE6</td>
</tr>
<tr>
<td>Channel 5</td>
<td>BC61</td>
</tr>
<tr>
<td>Channel 6</td>
<td>HI1</td>
</tr>
<tr>
<td>Channel 7</td>
<td>BI6</td>
</tr>
<tr>
<td>Channel 8</td>
<td>BC62 (Repeat)</td>
</tr>
<tr>
<td>Channel 9</td>
<td>HE1</td>
</tr>
<tr>
<td>Channel 10</td>
<td>BE1 (Repeat)</td>
</tr>
<tr>
<td>Channel 11</td>
<td>HB6</td>
</tr>
<tr>
<td>Channel 12</td>
<td>BC44</td>
</tr>
<tr>
<td>Channel 13</td>
<td>HB1 (Repeat)</td>
</tr>
<tr>
<td>Channel 14</td>
<td>BI1</td>
</tr>
<tr>
<td>Channel 15</td>
<td>HB7 (Repeat)</td>
</tr>
<tr>
<td>Channel 16</td>
<td>HB2</td>
</tr>
</tbody>
</table>
### F.3.3 Thermocouples in the Third Group

Table F.3: Third Group Thermocouples in Preliminary Tests

<table>
<thead>
<tr>
<th>Channel</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel 1</td>
<td>BE4</td>
</tr>
<tr>
<td>Channel 2</td>
<td>BI5</td>
</tr>
<tr>
<td>Channel 3</td>
<td>Water Cylinder 5</td>
</tr>
<tr>
<td>Channel 4</td>
<td>BB5</td>
</tr>
<tr>
<td>Channel 5</td>
<td>HE6</td>
</tr>
<tr>
<td>Channel 6</td>
<td>BI4</td>
</tr>
<tr>
<td>Channel 7</td>
<td>HI5</td>
</tr>
<tr>
<td>Channel 8</td>
<td>BC62 (Repeat)</td>
</tr>
<tr>
<td>Channel 9</td>
<td>BC52</td>
</tr>
<tr>
<td>Channel 10</td>
<td>BE1 (Repeat)</td>
</tr>
<tr>
<td>Channel 11</td>
<td>BC51</td>
</tr>
<tr>
<td>Channel 12</td>
<td>HB5</td>
</tr>
<tr>
<td>Channel 13</td>
<td>HB1 (Repeat)</td>
</tr>
<tr>
<td>Channel 14</td>
<td>HE5</td>
</tr>
<tr>
<td>Channel 15</td>
<td>HB7 (Repeat)</td>
</tr>
<tr>
<td>Channel 16</td>
<td>HB3</td>
</tr>
</tbody>
</table>
### F.3.4 Thermocouples in the Fourth Group

Table F.4: Fourth Group Thermocouples in Preliminary Tests

<table>
<thead>
<tr>
<th>Channel 1</th>
<th>Water Cylinder 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel 2</td>
<td>Water Cylinder 2</td>
</tr>
<tr>
<td>Channel 3</td>
<td>Water Pump Inlet (Repeat)</td>
</tr>
<tr>
<td>Channel 4</td>
<td>Water Cylinder 5</td>
</tr>
<tr>
<td>Channel 5</td>
<td>Water before Thermostat</td>
</tr>
<tr>
<td>Channel 6</td>
<td>Water Cylinder 4</td>
</tr>
<tr>
<td>Channel 7</td>
<td>Water Cylinder 6</td>
</tr>
<tr>
<td>Channel 8</td>
<td>BC62 (Repeat)</td>
</tr>
<tr>
<td>Channel 9</td>
<td>Water Cylinder 3</td>
</tr>
<tr>
<td>Channel 10</td>
<td>BE1 (Repeat)</td>
</tr>
<tr>
<td>Channel 11</td>
<td>BC53</td>
</tr>
<tr>
<td>Channel 12</td>
<td>BC63</td>
</tr>
<tr>
<td>Channel 13</td>
<td>HB1 (Repeat)</td>
</tr>
<tr>
<td>Channel 14</td>
<td>BC64</td>
</tr>
<tr>
<td>Channel 15</td>
<td>HB7 (Repeat)</td>
</tr>
<tr>
<td>Channel 16</td>
<td>BC54</td>
</tr>
</tbody>
</table>

### F.4 Main Testing

In this phase of testing the engine was run at idle for the first 20 seconds and then at a constant speed of 1161 rev/min (approximately) and a constant load of 48 Nm (approximately) was applied to the engine.
## F.4.1 Thermocouples in the First Group

Table F.5: First Group Thermocouples in Main Tests

<table>
<thead>
<tr>
<th>Channel</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Water Pump Inlet</td>
</tr>
<tr>
<td>2</td>
<td>BB4</td>
</tr>
<tr>
<td>3</td>
<td>HV4</td>
</tr>
<tr>
<td>4</td>
<td>HE4</td>
</tr>
<tr>
<td>5</td>
<td>BB5</td>
</tr>
<tr>
<td>6</td>
<td>Water before Thermostat</td>
</tr>
<tr>
<td>7</td>
<td>Water Cylinder 6</td>
</tr>
<tr>
<td>8</td>
<td>HI4</td>
</tr>
<tr>
<td>9</td>
<td>HV1</td>
</tr>
<tr>
<td>10</td>
<td>BE1 (Repeat)</td>
</tr>
<tr>
<td>11</td>
<td>BB3</td>
</tr>
<tr>
<td>12</td>
<td>BC62 (Repeat)</td>
</tr>
<tr>
<td>13</td>
<td>HB1 (Repeat)</td>
</tr>
<tr>
<td>14</td>
<td>BC41</td>
</tr>
<tr>
<td>15</td>
<td>HB7 (Repeat)</td>
</tr>
<tr>
<td>16</td>
<td>HB4</td>
</tr>
</tbody>
</table>
F.4.2 Thermocouples in the Second Group

Table F.6: Second Group Thermocouples in Main Tests

<table>
<thead>
<tr>
<th>Channel 1</th>
<th>BC42</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel 2</td>
<td>HI6</td>
</tr>
<tr>
<td>Channel 3</td>
<td>HV5</td>
</tr>
<tr>
<td>Channel 4</td>
<td>BE6</td>
</tr>
<tr>
<td>Channel 5</td>
<td>BC61</td>
</tr>
<tr>
<td>Channel 6</td>
<td>HI1</td>
</tr>
<tr>
<td>Channel 7</td>
<td>BI6</td>
</tr>
<tr>
<td>Channel 8</td>
<td>BC62 (Repeat)</td>
</tr>
<tr>
<td>Channel 9</td>
<td>HE1</td>
</tr>
<tr>
<td>Channel 10</td>
<td>BE1 (Repeat)</td>
</tr>
<tr>
<td>Channel 11</td>
<td>HB6</td>
</tr>
<tr>
<td>Channel 12</td>
<td>BC44</td>
</tr>
<tr>
<td>Channel 13</td>
<td>HB1 (Repeat)</td>
</tr>
<tr>
<td>Channel 14</td>
<td>BI1</td>
</tr>
<tr>
<td>Channel 15</td>
<td>HB7 (Repeat)</td>
</tr>
<tr>
<td>Channel 16</td>
<td>HB2</td>
</tr>
</tbody>
</table>
### F.4.3 Thermocouples in the Third Group

Table F.7: Third Group Thermocouples in Main Tests

<table>
<thead>
<tr>
<th>Channel 1</th>
<th>BE4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel 2</td>
<td>BI5</td>
</tr>
<tr>
<td>Channel 3</td>
<td>Water Cylinder 5</td>
</tr>
<tr>
<td>Channel 4</td>
<td>BB5</td>
</tr>
<tr>
<td>Channel 5</td>
<td>HE6</td>
</tr>
<tr>
<td>Channel 6</td>
<td>BI4</td>
</tr>
<tr>
<td>Channel 7</td>
<td>HI5</td>
</tr>
<tr>
<td>Channel 8</td>
<td>BC62 (Repeat)</td>
</tr>
<tr>
<td>Channel 9</td>
<td>BC52</td>
</tr>
<tr>
<td>Channel 10</td>
<td>BE1 (Repeat)</td>
</tr>
<tr>
<td>Channel 11</td>
<td>BC51</td>
</tr>
<tr>
<td>Channel 12</td>
<td>HB5</td>
</tr>
<tr>
<td>Channel 13</td>
<td>HB1 (Repeat)</td>
</tr>
<tr>
<td>Channel 14</td>
<td>HE5</td>
</tr>
<tr>
<td>Channel 15</td>
<td>HB7 (Repeat)</td>
</tr>
<tr>
<td>Channel 16</td>
<td>HB3</td>
</tr>
</tbody>
</table>
### F.4.4 Thermocouples in the Fourth Group

Table F.8: Fourth Group Thermocouples in Main Tests

<table>
<thead>
<tr>
<th>Channel</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel 1</td>
<td>Water Cylinder 1</td>
</tr>
<tr>
<td>Channel 2</td>
<td>Water Cylinder 2</td>
</tr>
<tr>
<td>Channel 3</td>
<td>Water Pump Inlet (Repeat)</td>
</tr>
<tr>
<td>Channel 4</td>
<td>Water Cylinder 5</td>
</tr>
<tr>
<td>Channel 5</td>
<td>Water before Thermostat</td>
</tr>
<tr>
<td>Channel 6</td>
<td>Water Cylinder 4</td>
</tr>
<tr>
<td>Channel 7</td>
<td>Water Cylinder 6</td>
</tr>
<tr>
<td>Channel 8</td>
<td>BC62 (Repeat)</td>
</tr>
<tr>
<td>Channel 9</td>
<td>Water Cylinder 3</td>
</tr>
<tr>
<td>Channel 10</td>
<td>BE1 (Repeat)</td>
</tr>
<tr>
<td>Channel 11</td>
<td>BC53</td>
</tr>
<tr>
<td>Channel 12</td>
<td>BC63</td>
</tr>
<tr>
<td>Channel 13</td>
<td>HB1 (Repeat)</td>
</tr>
<tr>
<td>Channel 14</td>
<td>BC64</td>
</tr>
<tr>
<td>Channel 15</td>
<td>HB7 (Repeat)</td>
</tr>
<tr>
<td>Channel 16</td>
<td>BC54</td>
</tr>
</tbody>
</table>
F.5 Installation of Flow meter and Thermocouple Modules

Before the third phase of experimentation, i.e., installation of an electric water pump, a flow meter was installed on the engine to measure the flow rate of the current mechanical water pump. The idea of adding the flow meter to the heater circuit was to measure the flow rate of coolant at different speeds for thermostat closed. Since, the thermostat is closed at the start of the engine, there is least flow in the engine, which is going through the heater circuit. Hence, this flow rate is the maximum flow rate, on which the engine has been designed by the manufacturer and this was made the reference point for this research. An electric water pump was used at this stage to decrease the flow rate of the coolant from this reference point and to investigate its effects on engine warm-up time. However, the installation of the flow meter has increased the capacitance of the coolant in the engine circuit, which increased the engine warm-up time. Although, the additional hoses used to install the flow meter were well insulated but the engine warm-up time was increased by 40 seconds approximately.

Moreover, at this stage, two more thermocouple modules were added to the system to monitor all the thermocouples data at the same time. The sequence of the thermocouple modules is given as follows:
## F.5.1 Thermocouple Module 1

Table F.9: Thermocouple Module 1 in Main Tests with Electric Water Pump

<table>
<thead>
<tr>
<th>Channel</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel 1</td>
<td>Water Cylinder 1</td>
</tr>
<tr>
<td>Channel 2</td>
<td>Water Cylinder 2</td>
</tr>
<tr>
<td>Channel 3</td>
<td>Water pump inlet</td>
</tr>
<tr>
<td>Channel 4</td>
<td>Water Cylinder 5</td>
</tr>
<tr>
<td>Channel 5</td>
<td>Water before Thermostat</td>
</tr>
<tr>
<td>Channel 6</td>
<td>Water Cylinder 4</td>
</tr>
<tr>
<td>Channel 7</td>
<td>Water Cylinder 6</td>
</tr>
<tr>
<td>Channel 8</td>
<td>BC62</td>
</tr>
<tr>
<td>Channel 9</td>
<td>Water Cylinder 3</td>
</tr>
<tr>
<td>Channel 10</td>
<td>BE1</td>
</tr>
<tr>
<td>Channel 11</td>
<td>HI3</td>
</tr>
<tr>
<td>Channel 12</td>
<td>BC63</td>
</tr>
<tr>
<td>Channel 13</td>
<td>HB1</td>
</tr>
<tr>
<td>Channel 14</td>
<td>BC64</td>
</tr>
<tr>
<td>Channel 15</td>
<td>HB7</td>
</tr>
<tr>
<td>Channel 16</td>
<td>BC54</td>
</tr>
</tbody>
</table>
### F.5.2 Thermocouple Module 2

Table F.10: Thermocouple Module 2 in Main Tests with Electric Water Pump

<table>
<thead>
<tr>
<th>Channel 1</th>
<th>BE4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel 2</td>
<td>BB3</td>
</tr>
<tr>
<td>Channel 3</td>
<td>BC61</td>
</tr>
<tr>
<td>Channel 4</td>
<td>BB5</td>
</tr>
<tr>
<td>Channel 5</td>
<td>HE6</td>
</tr>
<tr>
<td>Channel 6</td>
<td>HE1</td>
</tr>
<tr>
<td>Channel 7</td>
<td>BC41</td>
</tr>
<tr>
<td>Channel 8</td>
<td>BC52</td>
</tr>
<tr>
<td>Channel 9</td>
<td>BC51</td>
</tr>
<tr>
<td>Channel 10</td>
<td>BB4</td>
</tr>
<tr>
<td>Channel 11</td>
<td>HE5</td>
</tr>
<tr>
<td>Channel 12</td>
<td>HV4</td>
</tr>
<tr>
<td>Channel 13</td>
<td>BC42</td>
</tr>
<tr>
<td>Channel 14</td>
<td>HE4</td>
</tr>
<tr>
<td>Channel 15</td>
<td>HV5</td>
</tr>
<tr>
<td>Channel 16</td>
<td>BE6</td>
</tr>
</tbody>
</table>
F.5.3 Thermocouple Module 3

Table F.11: Thermocouple Module 3 in Main Tests with Electric Water Pump

<table>
<thead>
<tr>
<th>Channel</th>
<th>Code</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel 1</td>
<td>Bi1</td>
</tr>
<tr>
<td>Channel 2</td>
<td>HI1</td>
</tr>
<tr>
<td>Channel 3</td>
<td>BI6</td>
</tr>
<tr>
<td>Channel 4</td>
<td>BI5</td>
</tr>
<tr>
<td>Channel 5</td>
<td>HB6</td>
</tr>
<tr>
<td>Channel 6</td>
<td>BC44</td>
</tr>
<tr>
<td>Channel 7</td>
<td>BI4</td>
</tr>
<tr>
<td>Channel 8</td>
<td>HB2</td>
</tr>
<tr>
<td>Channel 9</td>
<td>HI5</td>
</tr>
<tr>
<td>Channel 10</td>
<td>HB5</td>
</tr>
<tr>
<td>Channel 11</td>
<td>HB3</td>
</tr>
<tr>
<td>Channel 12</td>
<td>HI6</td>
</tr>
<tr>
<td>Channel 13</td>
<td>BC53</td>
</tr>
<tr>
<td>Channel 14</td>
<td>BC43</td>
</tr>
<tr>
<td>Channel 15</td>
<td>HB4</td>
</tr>
<tr>
<td>Channel 16</td>
<td>HV1</td>
</tr>
</tbody>
</table>

After the installation of the flow meter, 2 tests for idle and 3 tests for loaded phase were done. The reason for performing these tests was not only to measure the flow rate of coolant in the heater circuit, but also to look at all the thermocouples at the same time and to check the repeatability of the thermocouple data.

F.6 Electric Water Pump Testing

In this phase of experimentation, the electric water pump was installed in series with the mechanical water pump. The mechanical water pump was
not doing any work because the impellor was removed and its presence was only to make the serpentine belt moving. The coolant flow rate was varied by using the electric water pump, which was running independently of the engine speed.

The heater circuit was once again elongated, to have the same direction of flow and to connect it before the pump. This increased the capacitance of the coolant in the engine and consequently, increased the engine warm-up time. Viewing the results, it was noticed that the engine warm-up time was increased by 40 seconds approximately, which was a total increase of 80 seconds approximately in the engine warm-up time, with the installation of the electric water pump.

A series of 9 tests were done with the EWP and the details are as follows:

Table F.12: Series of Tests undertaken with EWP

<table>
<thead>
<tr>
<th>S.No.</th>
<th>Engine Speed (rev/min)</th>
<th>Load (Nm)</th>
<th>Coolant Flow Rate (L/min)</th>
<th>% Flow Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test 1</td>
<td>1161</td>
<td>48</td>
<td>17.17</td>
<td>103</td>
</tr>
<tr>
<td>Test 2</td>
<td>1161</td>
<td>48</td>
<td>15.17</td>
<td>91</td>
</tr>
<tr>
<td>Test 3</td>
<td>1161</td>
<td>48</td>
<td>13.17</td>
<td>79</td>
</tr>
<tr>
<td>Test 4</td>
<td>1161</td>
<td>48</td>
<td>11.17</td>
<td>67</td>
</tr>
<tr>
<td>Test 5</td>
<td>1161</td>
<td>48</td>
<td>9.17</td>
<td>55</td>
</tr>
<tr>
<td>Test 6</td>
<td>1161</td>
<td>48</td>
<td>7.17</td>
<td>43</td>
</tr>
<tr>
<td>Test 7</td>
<td>1161</td>
<td>48</td>
<td>5.17</td>
<td>31</td>
</tr>
<tr>
<td>Test 8</td>
<td>1161</td>
<td>48</td>
<td>3.17</td>
<td>19</td>
</tr>
<tr>
<td>Test 9</td>
<td>1161</td>
<td>48</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

**F.7 Split Cooling System Testing**

Table F.13 shows the different running conditions for split cooling tests.
The engine was run at the same flow rate of 17.17 L/min to generate a reference point of testing to compare the results with the old tests, both with electric water pump and mechanical water pump. However, it was later on decided to analyze the results for the best case scenario, which was the 3.17 L/min case.

It was noticed that the warm-up time was greater in case of split cooling system, comparing the flow rate because of the following two reasons:

- Increased capacitance with the addition of more than a meter long hose of 3.8mm (internal diameter).
- The flow is only through the cylinder head and not through the engine block.
Appendix G – Heat Transfer Model

G.1 Model for Representative Cylinder

The model for an average cylinder representing as all the six cylinders was built in Dymola, which has been broken down into several sub-models. These sub-models are explained as following:

G.1.1 Combustion Chamber

The combustion chamber is the backbone of the heat transfer model built in this study. In the combustion chamber model, there are two submodels for combustion pressure and temperature. The combustion temperature and pressure data with respect to time and crank angle has been used from Keynejad [96]. Both of these submodels take theta, the crank angle as real input and give a real output as combustion pressure and temperature respectively.

Another submodel is for the gas heat transfer coefficient, which take combustion pressure, combustion temperature, crank angle, bore and stroke as real inputs and give a real output as ‘gas coefficient’. Furthermore, heat is convected through convection resistors to the head plate, piston and the upper and middle cylinder walls, which are exposed to the gas.
The fundamentals equations used to calculate the gas coefficient are based on Annand’s work [75], which is stated as

\[ Nu = a Re^b \] (G.1)

where the value of \( a = 0.49 \) and the value of \( b = 0.7 \), \( Re = \rho DU_m / \mu \), \( U_m = N * s / 30 \) is the mean piston speed in radians, \( D \) is the bore, \( \rho = P / 287 * T \) is the density of gas and \( T \) is the combustion temperature, \( \mu = 3.3^{-7} T^{-0.7} \) is the dynamic viscosity, which is referred to Heywood [34].

The gas coefficient

\[ h = kNu / D \] (G.2)

has been referred to Holman [113], whereas

\[ k = 1.52^{-4} + 4.42^{-5}T + 8^{-9}T^2 \] (G.3)

is as stated in Ferguson [114]. Heat transfer fundamentals are also referred to Incropera and Dewitt [115].

The convection resistors are calculated as the product of area of a specific component and the gas coefficient. There is a submodel for the area of the piston, which takes cylinder bore as a real input for calculating the area of the piston.

For calculating the area of the middle wall, there is another submodel, defined as the ‘wall area’, because heat transfer takes place from the combustion gases to the middle cylinder wall during the time, when the middle wall is exposed to the hot gases. When the piston is at the TDC, it is only the upper layer of the wall, which is exposed to the gas. However, when the piston reduces in the expansion stroke and exposes the middle wall to the hot gases, heat is transferred to the middle wall. Therefore, heat transfer to the middle layer of the cylinder wall is a function of piston movement, i.e. crank angle.

G.1.2 Crankshaft

The model crankshaft takes a real input as engine speed \( N \) and gives theta, the crank angle as a real output.
The fundamental equation used to calculate the modulus of the engine speed, $N$ is

$$\text{mod} = (N \ast t \ast \pi / 30,4\pi) \quad (G.4)$$

Where $t$ is the time for complete engine cycle for 720° of crank angle rotation, which is converted into radians by dividing by a factor of 30. The value of crank angle theta is used in the combustion model as a real input.

**G.1.3 Coolant Model**

There is a submodel named as ‘coolant’, which defines the flow of coolant. The coolant model takes $mC_p$ and $T_i$ as real input data and through two heat ports, defined as inlet and outlet, gives the actual quantity of water entering and the same quantity leaving the coolant model and flowing into another component. The quantity or mass of water flowing into and flowing out of the model is based on the law of conservation of energy. The fundamental equation used is as follows:

$$\dot{Q} = mC_p(T - T_i) \text{ and } \dot{Q}_{\text{inlet}} = \dot{Q}_{\text{outlet}} \quad (G.5)$$

**G.1.4 Direction of Coolant Flow in the Model**

The coolant starts flowing from the engine block where it flows around the cylinder liner and draws heat out of the cylinder walls. The coolant then reaches the cylinder head, flows around the head and finally goes into the head heat exchanger. The coolant cycle completes here and it is again fed into the engine block to begin the next cycle.

**G.1.5 Upper Cylinder Wall Model**

The upper cylinder wall model contains two capacitances for the coolant in the upper cylinder wall and the upper cylinder wall metal. The two
capacitances are connected to each other via a convection resistor, which defines the convection heat transfer between the coolant and the cylinder wall. The flow of heat is through the heat ports connected to the each capacitance. Heat from combustion is received by the wall metal and flows to the coolant. There is also a flow of heat from the upper to the middle wall through conduction between the two walls, which is defined in the submodel of ‘wall conductance’. This submodel takes $B$ as the bore, $t$ as the wall thickness, $k$ as thermal conductivity of the metal, which is Aluminium, $h_z$ as the length of the wall and a constant $C$, which is a correction factor applied and the value of constant $C$ is 0.7. The heat flow is defined as the amount of heat entering through a heat port as inlet from the upper wall and transferred into the middle wall through the heat port as outlet following the fundamental equation of conservation of energy. The fundamental equation used is the heat conduction equation between two metals, which is stated as follows:

$$\dot{Q} = kA(T_{inlet} - T_{outlet})$$

where $A = \pi Bt$ is the area of the upper cylinder wall, $B$ is the cylinder bore, $t$ is the wall thickness, $h_z$ is the length of each cylinder wall.

### G.1.6 Middle Cylinder Wall Model

The coolant submodel is duplicated in all the models. The submodel of the middle cylinder wall is also duplicated as the upper cylinder wall, except for an additional submodel, connected by a heat port to the middle wall submodel, which considers the conductance from the piston rings to the middle cylinder wall via the oil film. The thickness of the oil film is taken as $6\mu m = 6 \times 10^{-6} m$ and the thickness of the bottommost ring, which conducts heat to the oil, is taken as 2mm, as referred to Bohac et al [69]. The heat through thermal conduction is transferred from the piston crown to the
piston ring through the oil film in the bottommost ring, which is further transferred to the middle cylinder wall. This conduction from the ring to the middle wall is shown in the submodel, ‘G_cylindrical’. This submodel is modeled on the principle of energy conservation, the amount of energy entering as inlet is equal to the amount leaving as outlet,

\[ Q_{inlet} = Q_{outlet} \quad (G.7) \]

and the thermal conductance from the ring to the middle wall is calculated by the expression

\[ Q = \frac{\Delta T}{R} \quad (G.8) \]

where

\[ \Delta T = (T_{inlet} - T_{outlet}) \quad (G.9) \]

\[ R = \frac{\ln(r_2/r_1)}{2\pi hk} \quad (G.10) \]

where \( r_2 = B/2 \), \( B \) is the cylinder bore, \( r_1 = (r_2 - t) \), \( t = 6\mu m = 6 \times 10^{-6} m \) is the oil film thickness and \( h = 2\,mm \) is the ring thickness and \( k = 0.145 \) is the thermal conductivity of oil. The data used in the present model under study is referred to the model used by Bohac et al [69].

Another submodel is the wall conductance, which is exactly the same as the one in upper cylinder wall and defines the heat conducted to the lower cylinder wall, with the only difference that the value of the constant which was 0.7 in the upper wall is 3 in the middle wall.

Another submodel ‘skirt middlewall’ defines the heat conduction through the oil film, which has got a different thickness than the one used for rings. This oil film thickness is taken as \( 25\mu m = 25 \times 10^{-6} m \), as referred to Bohac et al [69]. This submodel is the same as the model ‘G_cylindrical’ mentioned above, which calculates the conductance from the ring to the middle cylinder wall, with the only difference that the oil film thickness is different in this case. Moreover, in the previous submodel, heat is coming from the crown and going to the middle cylinder wall but in this submodel, the heat is coming from the skirt and going to the middle cylinder wall.
G.1.7 Lower Cylinder Wall Model

Besides having the same coolant submodel, there is also another submodel called ‘lower cylinder wall’. This submodel defines two capacitances, one for the coolant flowing through the wall another for the metal capacitance of the lower cylinder wall. There is convection heat transfer from the metal part of the lower wall to the coolant.

Another submodel, ‘skirt lower wall’ defines the heat conduction from the oil film to the lower cylinder wall, which is the same as the one which is used in the middle cylinder wall and the oil film thickness is also the same i.e., \( 25 \mu m = 25 \times 10^{-6} \text{ m} \).

G.1.8 Oil Sump Model

The oil sump model has a capacitance for the quantity of oil used in the oil sump. Furthermore, there are four convection resistors in this model, which define convection heat transfer from the oil in the sump to the piston crown, piston skirt, the lower cylinder wall and the ambient. For the convection to the piston skirt and crown, we use the splash oil coefficient values of 1000 \( W/m^2K \) for crown underside and 240 \( W/m^2K \) for skirt underside, as referred to Bohac et al [69]. Other values of coefficient to the lower wall and to the ambient, some values are based on assumptions, which are later calibrated against experimental results in chapter 5.

G.1.9 Cylinder Volume Model

The cylinder volume is a function of piston movement and in turn a function of the crank angle theta. The volume of the cylinder is \( V_o \), when the piston is at the TDC. It reduces during the downward stroke and the volume of the cylinder starts to increase and it is maximum, when the piston is at the BDC. The governing equation used is as follows:
\[ Vol = V_o + s(\pi D^2/4)(1 - \cos\theta)/2, \quad [34] \tag{G.11} \]

### G.1.10 Piston and Piston Crown Model

The piston model consists of two capacitances, one for the piston skirt and another for the crown. The two capacitances are connected to each other by a thermal conduction between the piston skirt and piston crown. The submodel ‘G_cylindrical’ as discussed above takes into account this conduction heat transfer. It only differs in the real inputs. The real inputs used in the middle wall and lower wall ‘G_cylindrical’ submodel are for oil because there is an oil film, which carries heat from the piston crown and piston skirt to the cylinder wall. While, in this submodel, the real inputs used are for Aluminium because the heat transfer is between the piston skirt and piston crown.

First real input is the bore diameter, while the second real input is calculated from the product of the bore and 0.5 to make it half (as in the relation \( r_1/2, r_2/2 \) is used). Furthermore, an addition function is used and 0.001 is added to it, so that the value never becomes zero.

The piston model has two real input, one is the volume of the cylinder and other is the cylinder diameter bore. Thereafter, it calculates a real output \( h \), as height of the piston by the equation given below.

\[ h = \frac{Vol}{Area} = \frac{V}{\pi D^2 / 4} \tag{G.12} \]

### G.1.11 Inlet and Exhaust Port Model

The mass flow rate of the inlet and the exhaust port is fed into the inlet and exhaust port models as a real input. There is a temperature sensor connected to the capacitance of the gas inside the port, which gauges the temperature of the gas in the port. Once, the temperature of the gas is obtained, it is fed as a real input into a submodel called ‘gasproperties’. Another real input fed into the above submodel is pressure. For inlet port,
the pressure was assumed to be 50 kPa and to get the exhaust pressure, the real input was connected to the combustion chamber pressure submodel. The real outputs that are obtained out of this submodel are dynamic viscosity $\mu$, density $\rho$, thermal conductivity $k$, and specific heat of the gas $C_p$.

where,

$$\rho = \frac{P}{287 * T}$$ is the density of gas, $T$ is the gas temperature,

$$\mu = 3.3^{-7}T^{0.7}$$ is the dynamic viscosity and

$$k = 1.52^{-4} + 4.42^{-5}T + 8.9T^2$$ as stated in Ferguson [114]. The values of $C_p$ in a temperature range of $200^\circ C$ to $1000^\circ C$ have been referred have been referred to the NASA website [116].

Another submodel in the port model is the ‘flowheat’ submodel, which has four real inputs, namely, $T_1$, which is the initial temperature of the gas entering the port, $T_2$ is the temperature of the gas inside the port, $C_p$ is the specific heat and $m \_ dot$ is the mass flow rate for the concerned port (inlet or exhaust). The fundamental equation used is as follows:

$$Q = mC_p(T_1 - T_2) \tag{G.13}$$

which calculates the heat flow as a real output and this is provided as an input into the gas capacitance through a prescribed heat flow.

Another submodel which calculates the convection in the port is the ‘convector’ submodel. It defines 6 real inputs, as the port diameter $D$, mass flow rate $m \_ dot$, dynamic viscosity $\mu$, density $\rho$, thermal conductivity $k$, and specific heat of the gas $C_p$.

The fundamental equation used is the empirical relation for pipe and tube flow are given below.

$$Nu_d = 0.023 Re_0^{0.8} Pr^{0.4} \tag{G.14}$$

$$Nu = hD/k \tag{G.15}$$

$$Re = \rho DU_m/\mu \tag{G.16}$$

$$Pr = C_p \mu / k \tag{G.17}$$
This further implies that

\[ h = \frac{0.023}{D^{0.2}} (\rho V)^{0.8} k^{0.6} (C_p / \mu)^{0.4} \quad (G.18) \]

### G.1.12 Head Model

The head model contains a coolant submodel, which is the same as the one used in all the cylinder walls because the coolant flows from the cylinder walls to the head plate before going into the heat exchanger. Inside the head model, there is a submodel, which has two capacitances, one for the metal portion of the head plate, which is in direct contact with the hot combustion gases, and the other one for the coolant flowing through it. Both the capacitances are connected to each other by a convection resistor, which defines the convection of heat from the head metal plate to the coolant in the head. The flow of heat is defined by various heat flow ports, which state the direction of heat flow either from the combustion chamber to the head plate or from the head plate to the coolant, block, and to the ambient.

### G.1.13 Combined Ports and Head Model

The head upper plate and the two ports; exhaust and inlet are combined together so that the coolant capacitance can be treated as one capacitance. Both the port models are connected to the coolant capacitance via convective resistors. The real inputs in the port models are extended to communicate with the outside main model.

### G.1.14 Head Heat Exchanger Model

The head heat exchanger model contains two convection resistors, one defining the heat generated due to free convection and the other one convecting heat to the ambient. The resistor, which convects heat to the ambient takes the value of the temperature from another submodel ‘Thermostat’. The ‘Thermostat’ submodel defines the condition that if the temperature \( T \) of the heat exchanger is more than \( 65^\circ C \), (the base
temperature, which is already defined), the output in the 'Thermostat' submodel takes this value, otherwise its value is zero. The second convection resistor, i.e. convection to the ambient only works if the output in the thermostat takes a value and it is not zero. The value of the temperature output i.e. $T$ is fed into all the coolant circuits as a real input. If this value is less than $65^\circ C$, the circuit of the thermostat is not completed, otherwise it is completed and convection to the ambient starts.

G.1.15 Oil Heat Exchanger Model

The oil heat exchanger model is similar to the heat exchanger model with one end connected to the coolant and the other end connected to the ambient via heat ports. The circuit is similar to head heat exchanger, having two convection resistors, one for the internal heating of the fluid in the capacitance due to free convection and the other one for the convection to the ambient. The two real input used in the oil heat exchanger are $mC_p$ (Oil) and $T_i$ (oil), which have different values than the coolant flow rate and coolant temperature. Another difference in the circuit is the value of the 'base temperature' when the Oil thermostat opens, which is $77^\circ C$ in this case, whereas in the coolant heat exchanger, it is $65^\circ C$.

G.1.16 Engine Block Model

The engine block has been divided into two parts; namely upper block and lower block. Two capacitances have been defined, one for the upper metal part of the engine block and the other one for the lower metal part of the engine block. Heat is conducted between these two capacitances via conduction and it is defined by a conduction resistor. It is assumed that the coolant is not flowing in the bottommost part of the cylinder and hence, there is heat conduction through the lower cylinder wall metal to the lower mass of the engine block whereas, heat is transferred via convection from the coolant to the upper mass of the engine block. The lower mass of the
engine block also receives heat through convection from the oil sump. Furthermore, both the upper and lower block capacitances convects heat to the ambient.
Appendix H – Results for MAP, Spark Advance and Engine Speed

H.1 MAP Results

H.1.1 MAP in Preliminary Test
The manifold absolute pressure reduces from 101 kPa to 38 kPa approximately, immediately after the start of the engine. By the end of 150 seconds, it slowly drops down to 28 kPa approximately and stays there till the end of the run, having little variations with engine speed and load.

H.1.2 MAP in Main Test
The manifold absolute pressure reduces from 101 kPa to 38 kPa immediately after the start of the engine and then keeps on varying from 40 kPa to 36 kPa up till the end of 50 seconds. It reduces from 40 kPa to 36 kPa by the end of 100 seconds and stays at 36 kPa up to the end.
The reader is advised to refer to figures H1 to H7 for the MAP results.
Figure H.1: Comparison of MAP for Idle

Figure H.2: Comparison of MAP for 17.17 L/min MWP

Figure H.3: Comparison of MAP for 17.17 L/min EWP

Figure H.4: Comparison of MAP for 3.17 L/min EWP
Comparison of 1st 2nd 3rd and 4th runs for MAP for EWP for 0 Lt/min

Time (Sec)  
MAP (kpa)

<table>
<thead>
<tr>
<th></th>
<th>1st run</th>
<th>2nd run</th>
<th>3rd run</th>
<th>4th run</th>
</tr>
</thead>
<tbody>
<tr>
<td>MAP</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Comparison of 1st 2nd 3rd and 4th runs of MAP for Split, Flow in Head, 3.17 Lt/min

Time (Sec)  
MAP (kpa)

<table>
<thead>
<tr>
<th></th>
<th>1st run</th>
<th>2nd run</th>
<th>3rd run</th>
<th>4th run</th>
</tr>
</thead>
<tbody>
<tr>
<td>MAP</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Comparison of 1st 2nd 3rd and 4th runs for Split, Flow in Block, 3.17 Lt/min

Time (Sec)  
MAP (kpa)

<table>
<thead>
<tr>
<th></th>
<th>1st run</th>
<th>2nd run</th>
<th>3rd run</th>
<th>4th run</th>
</tr>
</thead>
<tbody>
<tr>
<td>MAP</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Figure H.5: Comparison of MAP for 0 L/min EWP

Figure H.6: Comparison of MAP for split (3.17 L/min, flow in head)

Figure H.7: Comparison of MAP for split (3.17 L/min, flow in block)
H.2 Spark Advance

H.2.1 General Understanding of the Controller Varying the Spark

The spark advance is set to 10° BTDC, when the engine starts. It starts with little variation at that position for up to 20 seconds approximately before it goes into a retarded operation after 20 seconds at idle to heat up the catalyst faster. However, if the engine is ramped with speed and load, the control system forces the system out of the retarded operation. The variation in spark advance is function of many factors like engine speed, MAP, engine starting temperature and air charge temperature. However, it can be noticed that although, there are no variations in engine speed, but there are still variations in the spark advance curves, which is mainly due to the engine starting at different temperatures.

H.2.2 Comparison for Spark Advance

H.2.3 Spark Advance in Preliminary Idle Test

When the engine starts, the spark tends to retard as a control strategy to warm-up the catalyst faster. When an engine is run with retarded spark, the combustion starts late and continues even when the exhaust valve opens. As a result, less work is done on the piston in the expansion stroke. Consequently, the end gas temperature is higher and more heat is produced in the exhaust port. This is usually done to heat up the catalyst faster in a cold start engine operation to make the catalyst reach the light-off temperature as quickly as possible.

In the default Ford calibration, the value of spark advance is 10° BTDC at MBT and it tends to retards after the engine starts, for the next 35 seconds approximately. When the engine starts, there are variations in spark advance and it does not retard immediately after start. The spark retards
only after 20 seconds after engine start. It retards up to value of 7.5° BTDC approximately, in the next 20 seconds and then it starts to advance. It reaches to a value of 12° BTDC approximately in the next 30 seconds. The rate of advancement in the spark increases after 70 seconds and it advances at a faster rate than before. It then reaches to a value of 20° approximately in the next 40 seconds and stays there till the engine warms up completely in 530 seconds approximately. Once the engine warms-up, the spark varies as a function of temperature and the variation increases. Now, the spark advance varies between 24° to 17° BTDC.

**H.2.4 Spark Advance in Main Test**

When the engine is started, the spark does not retard immediately and it varies for the first 20 seconds. As mentioned before, in the idle case, it begins to retard only after 20 seconds, but in this run, after 20 seconds, the engine is subjected to speed and load conditions. It has to be noted that the spark does not retard after speed and load are applied to the engine. A reasonable explanation to the behavior of spark advance can be due to the application of speed and load. When the engine is operating cold at idle and no load, the control system takes the system into CSSRE (Cold Start Spark Retard for Reduced Emissions) and this is happening in the idle case, but in the loaded case. Hence, the application of speed and load forces the system out of CSSRE. However, there are still other factors, which affect the spark advance, like variation in throttle, temperature, MAP and air charge temperature.

After 20 seconds of engine start, the spark advances and reaches to a value of 17° in 80 seconds after start. This advancement from 10° to 17° is at a slower rate and it happens in 60 seconds. After 80 seconds, it shoots up to value of 38° approximately in just 50 seconds, i.e. after completion of 130 seconds after start.

The spark then advances slowly in the next 60 seconds, i.e. after the completion of 180 seconds, the value of spark advance is 40° BTDC.
approximately. This value of spark advance stays as it is up to the end of the run.

There are some discrepancies in the spark advance for different coolant flow rate conditions, even though the speed and load conditions remain unchanged. The main reason for these discrepancies is the inconsistencies in throttle and load control and the temperature of the ambient at engine start. These factors force the control system to behave differently in the first 70 seconds of engine start, especially in the time from 20-70 seconds. The following figures from H8 to H14 give an understanding of the spark advance for various engine running conditions.

Figure H.8: Comparison of Total Spark Advance for Idle tests
Figure H.9: Comparison of Total Spark Advance for 17.17 L/min MWP
Comparison of 1st 2nd 3rd and 4th runs for EWP, 17.17 Lt/min

Comparison of 1st 2nd 3rd and 4th runs for EWP, 3.17 Lt/min

Comparison of 1st 2nd 3rd and 4th runs for Spark Advance for EWP for 0 Lt/min

Comparison of 1st 2nd 3rd and 4th runs for Split, Flow in Head, 3.17 Lt/min

Figure H.10: Comparison of Total Spark Advance for 17.17 L/min EWP

Figure H.11: Comparison of Total Spark Advance for 3.17 L/min EWP

Figure H.12: Comparison of Total Spark Advance for 0 L/min EWP

Figure H.13: Comparison of Total Spark Advance for split (3.17 L/min, flow in head)
H.3 Engine Speed

H.3.1 Preliminary Idle Tests

The engine starts at 850 rev/min approximately. The speed reduces slowly stabilizing the engine, which brings the fuel consumption down. In the idle runs, it is noted that it takes 175 seconds approximately for the speed to come down to 700 rev/min from the starting speed of 850 rev/min and after that the engine remains stabilized for the rest of the test.

H.3.2 Main Tests

In the loaded tests, the engine is only run at idle for 20 seconds and then it is subjected to a speed of 1161 rev/min and 48 Nm of load approximately. By the time the speed and load is applied on the engine, the speed usually drops from 850 rev/min to 800 rev/min in 20 seconds. The following figures
from H15 to H20 presents the engine speed profiles for different engine conditions.

Figure H.15: Comparison of engine speed for 17.17 L/min MWP

Figure H.16: Comparison of engine speed for 17.17 L/min EWP

Figure H.17: Comparison of engine speed for 3.17 L/min EWP

Figure H.18: Comparison of engine speed for 0 L/min EWP
Comparison of 1st, 2nd, 3rd and 4th runs for Split, Flow in Head, 3.17 L/min

<table>
<thead>
<tr>
<th>Time (Sec)</th>
<th>Engine Speed (rpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td></td>
</tr>
<tr>
<td>50</td>
<td></td>
</tr>
<tr>
<td>100</td>
<td></td>
</tr>
<tr>
<td>150</td>
<td></td>
</tr>
<tr>
<td>200</td>
<td></td>
</tr>
<tr>
<td>250</td>
<td></td>
</tr>
<tr>
<td>300</td>
<td></td>
</tr>
</tbody>
</table>

Comparison of 1st, 2nd, 3rd and 4th runs for Split, Flow in Block, 3.17 L/min

Figure H.19: Comparison of engine speed for split (3.17 L/min, flow in head)

Figure H.20: Comparison of engine speed for split (3.17 L/min, flow in block)
Appendix – I : Biasing of Flow

I.1 Introduction
Chapter 6 covers the results on coolant thermocouples, which includes the temperature distribution results of the coolant inside the engine for different flow rate conditions. Nonetheless, the discussion on the behaviour of thermocouples generates the need to discuss the flow pattern of the coolant in the engine, as it affects the engine cooling and components temperatures. The current appendix attempts to discuss the flow pattern of the coolant with the use of an electric water pump. The main points covered in the current appendix are as follows:

- The biasing of the flow with the use of an electric water pump
- Comparison of flow pattern between 17.17 L/min EWP and 3.17 L/min EWP
- Discussion on metal temperatures

I.2 Biasing of the Flow in 17.17 L/min EWP
Analyzing the block on the basis of inlet and exhaust temperatures, it has been observed that the flow is biased in the case of the EWP. When the engine operates with a MWP, the swirling motion of the impellor makes the coolant to agitate and gives some swirl to the flow of coolant. Therefore, the
coolant enters the engine block in an evenly distributed manner on the exhaust and inlet side. However, in case of the EWP, due to the absence of the impellor, the flow is not agitated and it makes more coolant to flow on the exhaust side of the engine block as compared to the inlet side.

It has to be noted in figure I.1 (b) for block inlet and exhaust thermocouples that BI1 (EWP) and BE1 (EWP) are running close to each other, while there is significant difference between BI1 (MWP) and BE1 (MWP). This indicates that there is some extra cooling provided to the exhaust in case of EWP, which is only possible when there is an extra flow of coolant on the exhaust side.

Moreover, when the coolant moves further from cylinder 1 to the next cylinders, the circular motion of the coolant around the cylinder helps it to regain evenness and makes the coolant even in distribution on both sides of the engine block. As a result, this trend is not repeated in cylinder 4.

This biasing of the flow is also responsible for extra fuel consumed in the first 100 seconds of the EWP.

Detailed discussion on flow biasing can be found in the following sections, which covers the comparison for MWP and EWP for 17.17 L/min of flow rate and comparison of 17.17 L/min and 3.17 L/min of flow rate for EWP.

**I.3 Comparison of MWP and EWP for 17.17 L/min**

**I.3.1 Cylinder 1**

The biasing of the flow has effects on the head temperature. As stated in the discussion for water temperatures in the preceding section, the flow of coolant going in the block, in case of EWP has uneven distribution of coolant around cylinder 1. As the volume of coolant moving on the exhaust side is higher than the inlet side, it tends to draw more heat from the metal and consequently, the exhaust side block surface runs cooler in case of EWP as compared to the MWP. As more heat is drawn by the coolant in the block, its makes the coolant in the block hotter in case of EWP and the
consequences are seen on the exhaust valve bridge temperature for the first 100 seconds. It can be noticed in figure I.1 (a) for cylinder 1 that the temperature of the exhaust valve bridge, for the EWP, runs very close to the temperature of exhaust valve bridge for MWP. The effects of higher capacitance of the coolant in the engine circuit are seen only after 100 seconds approximately, when the temperature of the exhaust valve bridge starts to come down as compared to the MWP.

Similar trends can be noticed, while analyzing the head surface temperature. The temperature of the head surface in case of EWP runs very close to the MWP, for the first 100 seconds. The temperature only starts to come down after 100 seconds approximately, which is mainly the effects of higher coolant capacitance.

The bump in the temperature of the exhaust valve bridge after 100 seconds is because the engine control system produces an advance in spark after 100 seconds approximately. Between 70-120 seconds, the spark advances from $17-18^\circ$ to $36^\circ$ BTDC approximately and this causes a fall in temperature on the exhaust side.

### I.3.2 Cylinder 4

As the flow becomes uniform, when the coolant moves from cylinder 1 towards cylinder 4, no variations are seen in the temperature profiles of the block inlet and block exhaust thermocouples. The temperature difference between the two thermocouples remains steady and does not vary, as MWP and EWP are compared.

While analyzing the block bore thermocouples, it has been noticed that as the flow starts to become uniform, it only affects the far end surface temperature of the engine block. The inner core temperature of the engine block, i.e. the temperature of the block bore, still follows the same trend as cylinder 1. The temperature of the block bore thermocouple for EWP runs close to the MWP, but there is a fall in temperature after 100 seconds.
approximately, which is due to the higher capacitance of the coolant in the cooling circuit. The same temperature trend can be observed in the head bore and exhaust valve bridge thermocouples. It has been noticed that the head bore and head exhaust valve bridge thermocouples for EWP are running close to their counterparts in case of MWP, but after 100 seconds approximately, there is a fall in temperature for both the thermocouples due to the higher capacitance.

These results have been presented in figure I.3 (a) and figure I.3 (b).

I.3.3 Cylinder 5

Similar trends can be observed for cylinder 5, referring to figure I.4, for the temperatures of the block bore, head bore and exhaust valve bridge, as discussed for cylinder 4. The temperature tends to be close for both the cases, but it tends to fall down, in case of EWP after 100 seconds approximately. The block bore temperature for EWP starts to fall after 150 seconds approximately, while the head bore and exhaust valve bridge temperature reduces after 100 seconds approximately.

I.3.4 Cylinder 6

HB6 and HB7 are the thermocouples in the head bore, on either sides of cylinder 6. It can be noticed in the figure I.5 (a) that there is a significant difference in the temperature of one side of the cylinder to the other side. The coolant which is running in the part of the water jacket between cylinder 5 and cylinder 6 experiences a higher temperature and similar trend has been observed for the metal parts.

There is a difference of $10^\circ C$ approximately between HB6 and HB7 thermocouples, which means that there is a difference of $10^\circ C$ approximately in the head bore temperature, on either sides of cylinder 6. The temperature of the HB6 thermocouple for EWP is running quite close to
HB6 of MWP, for the first 100 seconds and then the temperature difference increases.

HB6 is following the trend of previous thermocouples, but in case of HB7, it has been observed that there is no sudden fall in temperature for EWP and the difference between the temperature for EWP and MWP has always been steady throughout the run. The reason is because major portion of the coolant rises and go into the head through cylinder 6, from the far end. Since, the flow pattern is the same here for both the cases, the difference in temperature has always been steady. The steady temperature difference can also be noticed in the coolant thermocouples in cylinder 6 because this thermocouple was installed in the head, on the back of the engine.

In the block exhaust and inlet thermocouples, referred to figure 1.5 (b), it has been observed that the exhaust thermocouple is experiencing the same temperature profile as in previous cylinders, i.e. running close to its counterpart (MWP) for the first 100 seconds and then dropping down, widening the gap between the two after 100 seconds approximately. The inlet thermocouple also shows some increase in temperature after 100 seconds and then the difference becomes steady.
Figure I.1 (a): Comparison for cylinder 1 (MWP and EWP 17.17 L/min)

Figure I.1 (b): Comparison for BE1 and BI1 (MWP and EWP 17.17 L/min)

Figure I.2 (a): Comparison for cylinder 3 (MWP and EWP 17.17 L/min)

Figure I.2 (b): Comparison for HI3 (MWP and EWP 17.17 L/min)
Figure I.3 (a): Comparison for cylinder 4 (MWP and EWP 17.17 L/min)

Figure I.3 (b): Comparison for BE4 and BI4 (MWP and EWP 17.17 L/min)

Figure I.4: Comparison for cylinder 5 (MWP and EWP 17.17 L/min)
I.4 General discussion on Cylinder 1 Vs Cylinder 6 for 17.17 L/min

I.4.1 Comparison between Coolant Temperatures

While comparing cylinder 1 and cylinder 6, it has been noticed that the head coolant temperature of cylinder 1 is always higher than cylinder 6. This is because the coolant extracts heat from the all the cylinders and its temperature rises, while on its way from cylinder 6 to cylinder 1. Moreover, the flow velocities are higher around cylinder 1 as compared to cylinder 6. When the coolant reaches cylinder 1, it rushes towards the thermostat. Even if the thermostat is closed, the coolant finds its path through the smaller diameter heater circuit pipe and since the diameter of the pipe is smaller, the velocity increases and consequently, there is a higher heat transfer.
These results can be analyzed in the light of the heat transfer fundamentals.

\[ \text{Re} = \frac{\rho v \gamma}{\mu}, \quad Nu = \frac{hd}{k} \text{ and } Nu = f(n(Re, Pr)). \]

When the flow velocity is higher, the Reynold’s number is higher and hence the Nusselt number is higher. Consequently, the heat transfer coefficient is higher and there is a higher heat transfer. The higher heat transfer leads to a rise in metal temperatures.

**I.4.2 Comparison between BE1 and BE6, BI1 and BI6**

It can be noticed from figure I.5 (b) that cylinder 6 is running hotter than cylinder 1. BE6 is running 5–7°C approximately more than BE1 at all times.

While comparing the inlet thermocouples, in figure I.6, it can be noted that BI6 is lower than BI1. The difference is 2–3°C approximately in case of the MWP, but in case of the EWP, the difference is 5°C approximately.

**I.4.3 Comparison between HB1, HB6 and HB7**

It can be noticed that the metal temperatures for cylinder 6 are higher than cylinder 1. The thermocouples HB6 and HB1 are running with a difference of 12°C approximately, whereas HB7 is even lower than HB1 and there is a difference of 5°C approximately between HB1 and HB7, which means that there is a difference of 15°C approximately on either sides of the bore for cylinder 6. The reason is that the coolant starts to accumulate around cylinder 6 because of the large area of the water jacket and this makes a lot of coolant to flow around cylinder 6, in the block. Furthermore, the velocity of the coolant drops, while the coolant rises up into the cylinder head, which acts as a hindrance for the coolant to rise up. Consequently, some portion of the coolant returns to the inlet side of the block, which increases the circulation of coolant around cylinder 6 in the engine block. Moreover, the coolant that goes up into the head is cooler due to more circulation around
cylinder 6 in the block. This is the reason the thermocouple in the head at cylinder 6 is experiencing a lower temperature as compared to other cylinders. The results can be found in figure I.8.

I.4.4 Comparison between HE1 and HE6, HI1 and HI6

One of the most important findings is that HI1 is running hotter than HI6. This discrepancy is due to the high velocity of the coolant in the head. When the coolant rises up from the block to go in the head, it tends to flow more on the inlet side. Hence, the velocity of the coolant is higher on the inlet side and consequently more heat is dissipated from the metal in contact.

The difference between the temperature of HI1 and HI6 is \(5^\circ C\) approximately, while the difference between BI1 and BI6 is \(2^\circ C\) approximately. The results can be referred to figure I.7.

Comparing the head exhaust thermocouples, in figure I.9, it can be noticed that the difference between HE6 and HE1 is \(5 - 7^\circ C\) approximately, which is similar to the block exhaust thermocouples.

I.4.5 Higher Capacitance in EWP

There is a slight fall in temperature after 100 seconds approximately in case of EWP and the reason is the higher capacitance of coolant in the engine coolant circuit. As the engine runs, the coolant in the engine gets displaced and it flows from the heater circuit and comes back to the EWP. It is only because of the increased capacitance of the coolant that the engine is still running cold coolant through it and this makes the temperature drop after 100 seconds approximately.
Figure I.6: Comparison for BI1 and BI6 (MWP and EWP 17.17 L/min)

Figure I.7: Comparison for HI1 and HI6 (MWP and EWP 17.17 L/min)

Figure I.8: Comparison for HB1, HB6 and HB7 (MWP and EWP 17.17 L/min)

Figure I.9: Comparison for HE1 and HE6 (MWP and EWP 17.17 L/min)
I.5 Comparison between 17.17 L/min Vs 3.17 L/min EWP

I.5.1 Cylinder 1
While analyzing 17.17 L/min, EWP, it was noticed that there was no difference in the temperatures of the inlet and exhaust thermocouples of the block of cylinder 1, due to the difference in flow pattern. It was concluded in that discussion that because of the variation in the flow pattern, more flow of coolant was diverted to the exhaust side of the engine.

However, it can be noticed in the comparison made for 17.17 L/min and 3.17 L/min for EWP, that there is a difference in the temperature in the exhaust and inlet thermocouples in the block for 3.17 L/min, which is not present in case of the 17.17 L/min. Since the flow rate was lowered from 17.17 L/min to 3.17 L/min, the variations in the flow pattern have also come down or disappeared. Moreover, analyzing the 3.17 L/min, it can be noticed that BE1 is running hotter than the BI1 for only 270 seconds approximately and then the temperature of BE1 drops and they coincide with each other.

The temperature of the coolant at cylinder 1, for 3.17 L/min shows a sharp rise in the beginning for up to 100 seconds approximately, as compared to the coolant temperature at cylinder 1, for 17.17 L/min.

The thermocouples for head bore and exhaust valve bridge do not show any sharp rise in temperature, in the beginning, for both the cases. A constant temperature difference of $10^\circ C$ can be noticed in both the runs.

The results can be referred to figure I.10 (a) and I.10 (b).

I.5.2 Cylinder 2 and Cylinder 3
The trend for coolant temperature is the same for cylinder 2 and cylinder 3. A sharp rise in temperature can be noticed in the beginning till 100 seconds approximately, for 3.17 L/min, EWP, which is not the case while comparing the head bore thermocouples. The results are shown in figures I.11 and I.12.
I.5.3 Cylinder 4

While comparing cylinder 4, it can be noticed that although, the temperature profile is similar to the previous cylinders, there are more fluctuations in the coolant temperature in 3.17 L/min flow rate. The temperature rise is more in the first 100 seconds for 3.17 L/min. It reduces after 100 seconds approximately and after 100 seconds, there is a steady temperature difference between the two temperature profiles. These temperature fluctuations in the coolant thermocouples could be due to noise in the data or due the lowering of the flow rate or both. Both the inlet and exhaust thermocouples for 3.17 L/min flow rate are running $5^\circ C$ approximately hotter than in the 17.17 L/min flow rate run. The results can be found in figure I.13 (a) and I.13 (b).

I.5.4 Cylinder 5

Similar trends for temperature profile and fluctuations can be observed, while comparing coolant temperatures for cylinder 5 for the above mentioned flow rates. It can be noticed that the temperature difference between the inlet thermocouples for cylinder 5 is more in the beginning for the first 100 seconds approximately and then it starts to decrease and both the profiles begin to converge. The results are shown in figure I.14 (a) and figure I.14 (b).

I.5.5 Cylinder 6

While comparing cylinder 6, the results, in figure I.15 (a) and figure I.15 (b) show that the coolant temperature for 3.17 L/min is running lower than 17.17 L/min. The main reason is lowering of coolant flow rate. The flow rate is not too high to force the coolant to go up into the head, which is making the coolant circulate in the block itself. This can be justified by viewing the temperatures of BI5 and BI6.
Results show that the temperature of BI5, for 3.17 L/min, which starts to drop after 100 seconds, intersects BI5, for 17.17 L/min at the end of 550 seconds approximately. Furthermore, the temperature of BI6, for 3.17 L/min, takes only 150 seconds approximately to intersect BI6, for 17.17 L/min and it even runs lower after that time. Due to the large area of the water jacket around cylinder 6 and the lower flow rate of the coolant, a large amount of coolant is accumulating around cylinder 6. Moreover, the velocity of the coolant is not enough to force the coolant go up into the head. This leads to more circulation of the coolant around cylinder 6 in the block and therefore, a further decrease in the coolant temperature at cylinder 6 has been observed in case of 3.17 L/min.

It has been noticed that due to this reason, there is a downfall in the temperature of the head bore thermocouples. The temperature of HB7, for 3.17 L/min, starts to come down after 100 seconds approximately and it intersects HB7, for 17.17 L/min at the end of 550 seconds approximately. No change can be observed in HB6 because it is in between cylinder 5 and 6, whereas, the effects of coolant accumulation are seen in the far corner of cylinder 6.

Figure I.10 (a): Comparison for cylinder 1(17.17 L/min and 3.17 L/min EWP)

Figure I.10 (b): Comparison for BE1 and BI1 (17.17 L/min and 3.17 L/min EWP)
Figure I.11: Comparison for cylinder 2 (17.17 L/min and 3.17 L/min EWP)

Figure I.12: Comparison for cylinder 3 (17.17 L/min and 3.17 L/min EWP)

Figure I.13 (a): Comparison for cylinder 4 (17.17 L/min and 3.17 L/min EWP)

Figure I.13 (b): Comparison for BE4 and BI4 (17.17 L/min and 3.17 L/min EWP)
Figure I.14 (a): Comparison for cylinder 5 (17.17 L/min and 3.17 L/min EWP)

Figure I.14 (b): Comparison for BI5 (17.17 L/min and 3.17 L/min EWP)

Figure I.15 (a): Comparison for cylinder 6 (17.17 L/min and 3.17 L/min EWP)

Figure I.15 (b): Comparison for BE6 and BI6 (17.17 L/min and 3.17 L/min EWP)
I.6 Discussion on Metal Temperatures

- **Comparison of BB Thermocouples for 3.17 L/min EWP and 0 L/min EWP**: The temperature starts to grow up faster in the 0 L/min run after 50 seconds approximately. At the end of 200 seconds, the temperatures of the bore, for 0 L/min, are $110\,^\circ C$ approximately as compared to the 3.17 L/min run, the temperatures of the bore at the end of 200 seconds are $90\,^\circ C$ approximately. The results can be found in figure I.17.

- **Comparison of BE and HV Thermocouples for 3.17 L/min EWP and 0 L/min EWP**: Results in figure I.18, show that the bore temperature at cylinder 6 is more than the bore temperature at cylinder 1, in both of the flow conditions, while comparing the block exhaust thermocouples, for 3.17 L/min and 0 L/min. There is a difference in temperature of $10\,^\circ C$ approximately, while comparing BE1 and BE6 for 3.17 L/min, whereas, for 0 L/min, this difference in temperature is increasing with time. The difference is $10\,^\circ C$ approximately at 100 seconds and it rises to $15\,^\circ C$ approximately at the end of 200 seconds.

It has been noticed that the exhaust valve thermocouples for 3.17 L/min are not running with any significant difference with each other. The approximate difference in temperature is $2\,^\circ C$ and cylinder 5 is running the hottest. In case of 0 L/min, exhaust valve thermocouple at cylinder 1 deviates due to some reason after 50 seconds approximately (a fault developed in the thermocouple), while the thermocouples at the exhaust valve of cylinder 4 and 5 are still running close to each other.

The results show that the block bore thermocouples, for 3.17 L/min, at cylinder 3, 4 and 5 reach $100\,^\circ C$ in 300 seconds approximately,
whereas, in case of 0 L/min, it reaches 100°C in 150 seconds approximately.

- **Comparison of Block Bore temperatures for Split cooling:** There is a higher rise in temperature in the block for split (3.17 L/min, flow in block) as compared to split (3.17 L/min, flow in head). The block bore temperature is 100°C at the end of 130 seconds approximately from start for the split (flow in block), while it reaches 100°C at the end of 230 seconds for split (flow in head). With this difference in temperature, it can be concluded that if the block is heated faster, the piston friction would reduce, leading to fuel consumption benefits. This has also been discussed in the fuel consumption results. The results have been shown in figure I.16.

- **Metal temperatures for 0 L/min:** The exhaust valve bridge temperature reaches 100°C in 50 seconds from the engine start. The other thermocouples which are either on the exhaust side of the head or on the block reach 100°C in 170 seconds approximately. The coolant around the cylinder reaches the engine operating temperature i.e. 95°C in 100 seconds approximately. The results can be referred to figure I.19.
Figure I.16: Comparison for block bore temperatures for split cooling (flow in head and block)

Figure I.17: Comparison for block bore temperatures for 3.17 L/min and 0 L/min EWP

Figure I.18: Comparison for BE and HV thermocouples for 3.17 L/min and 0 L/min EWP

Figure I.19: Metal temperatures for 0 L/min EWP
I.7 Summary

It has been shown in the analysis that the coolant flow pattern changes with the utilization of the electric water pump, due to which there are effects on the temperatures of engine components. Moreover, it also affects the warm-up time of the engine. In this chapter, comparisons have been made between mechanical and electric water pump and various metal thermocouples to realize the effects of flow variations. This is considered to be an important analysis because it provides an understanding about the flow pattern of the coolant in the engine. Moreover, as different flow patterns are encountered in EWP and MWP, this analysis also becomes helpful in understanding the initial flow pattern in the engine with the MWP.