Development and Analysis of a Split-Cycle Engine Fuelled with Methane

Iain Cameron
University of Windsor

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DEVELOPMENT AND ANALYSIS OF A SPLIT-CYCLE ENGINE FUELLED WITH METHANE
by:
Iain A.S. Cameron

A Dissertation
Submitted to the Faculty of Graduate Studies through Mechanical, Automotive, and Materials Engineering in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy at the University of Windsor

Windsor, Ontario, Canada

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by

Iain A.S. Cameron

APPROVED BY:

_________________________________________
Dr. R. Fraser, External Examiner
University of Waterloo

_________________________________________
Dr. D. Green
Department of Mechanical, Automotive, and Materials Engineering

_________________________________________
Dr. G. Rankin
Department of Mechanical, Automotive, and Materials Engineering

_________________________________________
Dr. B. Minaker
Department of Mechanical, Automotive, and Materials Engineering

_________________________________________
Dr. A. Sobiesiak, Advisor
Department of Mechanical, Automotive, and Materials Engineering

January 13, 2016
Author’s Declaration of Originality

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Abstract

Natural gas, which is primarily composed of methane, offers many advantages over other hydrocarbon fuels for use in reciprocating piston engines. Generally, these include: a high octane rating, wide flammability limits, a high gravimetric energy content, reduced CO\textsubscript{2} output, and lower levels of harmful exhaust gas emissions. However, natural gas presents some unique challenges due to its low density and slow laminar burning velocity. The former causes issues with volumetric efficiency and/or charge homogeneity depending on the fuel delivery method. The latter results in prolonged combustion durations that are counter-productive to high fuel conversion efficiencies.

A split-cycle engine divides the conventional four-stroke engine process between two adjoining cylinders: one cylinder for intake and compression, the second cylinder for combustion and exhaust. The passage that connects these two cylinders together provides an alternative location for fuel injection and mixing. Furthermore, the fluid exchange process occurring from this passage to the combustion chamber is a source of turbulence generation desired to enhance the rate of combustion.

In this work a spark ignition split-cycle research engine has been developed and tested for the purpose of evaluating its ability to alleviate the aforementioned problems associated with natural gas (methane) fuelled engines. A novel fuel injector location and timing have been employed and the results show excellent mixture homogeneity was achieved. The fuelling strategy also decoupled the injection event from the engine’s intake air flow rate; however, the volumetric efficiency still remained low, between 71–75\%, due to flow losses in the compression cylinder. Combustion rates were found to be very rapid with both early and main burn duration periods on the order of 10–15°CA (crank angle), despite unfavourable burning conditions (i.e. low cylinder temperature and late combustion phasing). The exhaust gas emission levels of nitrogen oxide, carbon monoxide, and total unburned hydrocarbons were all below average values listed for spark ignition engines.
Dedicated to my parents, Elaine and Stewart.
Acknowledgements

I would like to begin by thanking my supervisor, Dr. Andrzej Sobiesiak, whose guidance and encouragement has brought me to a fulfilling and successful conclusion of my doctoral studies. The liberty that he afforded me in my research allowed me to freely pursue my interests, and while this was not always the easiest route, it has led to a richer and more rewarding experience. I thank him for his patience and unwavering support.

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Iain A.S. Cameron
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Abbreviations

ABDC  After Bottom Dead Centre
AC   Alternating Current
AFR  Air-to-Fuel Ratio
ASTM American Society for Testing and Materials
ATDC After Top Dead Centre
BBDC Before Bottom Dead Centre
BDC  Bottom Dead Centre
BHN  Brinell Hardness Number
BIP  Bowl-In-Piston
BMEP Brake Mean Effective Pressure
BSFC Brake Specific Fuel Consumption
BTDC Before Top Dead Centre
CAD  Computer Aided Drafting
CFD  Computational Fluid Dynamics
CGI  Compacted Graphite Iron
CI   Compression Ignition
CNC  Computer Numerically Controlled
CNG  Compressed Natural Gas
COV  Coefficient of Variation
DAQ  Data Acquisition System
DI   Direct Injection
DLC  Diamond-Like Carbon
DOE  Design of Experiments
EDM  Electrical Discharge Machining
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<th>Description</th>
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<tbody>
<tr>
<td>EGR</td>
<td>Exhaust Gas Recirculation</td>
</tr>
<tr>
<td>EGT</td>
<td>Exhaust Gas Temperature</td>
</tr>
<tr>
<td>EOC</td>
<td>End of Combustion</td>
</tr>
<tr>
<td>EPA</td>
<td>Environmental Protection Agency</td>
</tr>
<tr>
<td>EVC</td>
<td>Exhaust Valve Closing</td>
</tr>
<tr>
<td>EVO</td>
<td>Exhaust Valve Opening</td>
</tr>
<tr>
<td>FEA</td>
<td>Finite Element Analysis</td>
</tr>
<tr>
<td>FMEP</td>
<td>Friction Mean Effective Pressure</td>
</tr>
<tr>
<td>FPGA</td>
<td>Field Programmable Gate Array</td>
</tr>
<tr>
<td>GHG</td>
<td>Greenhouse Gas</td>
</tr>
<tr>
<td>HC</td>
<td>Hydrocarbon</td>
</tr>
<tr>
<td>HEX</td>
<td>Heat Exchanger</td>
</tr>
<tr>
<td>HRC</td>
<td>Hardness Rockwell C-scale</td>
</tr>
<tr>
<td>IDI</td>
<td>Indirect Injection</td>
</tr>
<tr>
<td>IMEP</td>
<td>Indicated Mean Effective Pressure</td>
</tr>
<tr>
<td>ISO</td>
<td>International Organization for Standardization</td>
</tr>
<tr>
<td>IVC</td>
<td>Intake Valve Closure</td>
</tr>
<tr>
<td>IVO</td>
<td>Intake Valve Opening</td>
</tr>
<tr>
<td>LFE</td>
<td>Laminar Flow Element</td>
</tr>
<tr>
<td>LNG</td>
<td>Liquefied Natural Gas</td>
</tr>
<tr>
<td>LPP</td>
<td>Location of Peak Pressure</td>
</tr>
<tr>
<td>LTC</td>
<td>Low Temperature Combustion</td>
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<tr>
<td>MBT</td>
<td>Mean Best Torque</td>
</tr>
<tr>
<td>MEP</td>
<td>Mean Effective Pressure</td>
</tr>
<tr>
<td>MFB</td>
<td>Mass Fraction Burned</td>
</tr>
<tr>
<td>MRW</td>
<td>Modified Rassweiler and Withrow</td>
</tr>
<tr>
<td>NEDC</td>
<td>New European Driving Cycle</td>
</tr>
<tr>
<td>NG</td>
<td>Natural Gas</td>
</tr>
<tr>
<td>NI</td>
<td>National Instruments</td>
</tr>
<tr>
<td>NMHC</td>
<td>Non-Methane Hydrocarbon</td>
</tr>
<tr>
<td>NMOG</td>
<td>Non-Methane Organic Gas</td>
</tr>
<tr>
<td>NTP</td>
<td>Normal Temperature and Pressure</td>
</tr>
<tr>
<td>Abbreviation</td>
<td>Description</td>
</tr>
<tr>
<td>--------------</td>
<td>-------------</td>
</tr>
<tr>
<td>OD</td>
<td>Outside Diameter</td>
</tr>
<tr>
<td>OE</td>
<td>Original Equipment</td>
</tr>
<tr>
<td>OHC</td>
<td>Overhead Camshaft</td>
</tr>
<tr>
<td>PR</td>
<td>Pressure Ratio</td>
</tr>
<tr>
<td>PRN</td>
<td>Normalized Pressure Ratio</td>
</tr>
<tr>
<td>PRR</td>
<td>Pressure Rise Rate</td>
</tr>
<tr>
<td>RH</td>
<td>Relative Humidity</td>
</tr>
<tr>
<td>RON</td>
<td>Research Octane Number</td>
</tr>
<tr>
<td>RPM</td>
<td>Revolutions Per Minute</td>
</tr>
<tr>
<td>RPV</td>
<td>Reverse Poppet Valve</td>
</tr>
<tr>
<td>RSS</td>
<td>Root Sum Squared</td>
</tr>
<tr>
<td>RT</td>
<td>Real Time</td>
</tr>
<tr>
<td>SE</td>
<td>Standard Error (of the mean)</td>
</tr>
<tr>
<td>SI</td>
<td>Spark Ignition</td>
</tr>
<tr>
<td>SOC</td>
<td>Start of Combustion</td>
</tr>
<tr>
<td>SOI</td>
<td>Start of Injection</td>
</tr>
<tr>
<td>TC</td>
<td>Thermocouple</td>
</tr>
<tr>
<td>TDC</td>
<td>Top Dead Centre</td>
</tr>
<tr>
<td>THC</td>
<td>Total Hydrocarbon</td>
</tr>
<tr>
<td>TKE</td>
<td>Turbulent Kinetic Energy</td>
</tr>
<tr>
<td>TTL</td>
<td>Transistor-Transistor Logic</td>
</tr>
<tr>
<td>TWCC</td>
<td>Three-Way Catalytic Converter</td>
</tr>
<tr>
<td>UEGO</td>
<td>Universal Exhaust Gas Oxygen</td>
</tr>
<tr>
<td>UHC</td>
<td>Unburned Hydrocarbon</td>
</tr>
<tr>
<td>VI</td>
<td>Virtual Instrument</td>
</tr>
<tr>
<td>WOT</td>
<td>Wide Open Throttle</td>
</tr>
<tr>
<td>XIVC</td>
<td>Crossover Inlet Valve Closing</td>
</tr>
<tr>
<td>XIVO</td>
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<tr>
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<td>XOVO</td>
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## Units

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<thead>
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<tbody>
<tr>
<td>%</td>
<td>Percent</td>
</tr>
<tr>
<td>ppm</td>
<td>Parts Per Million</td>
</tr>
<tr>
<td>°</td>
<td>Degrees (temperature or angle)</td>
</tr>
<tr>
<td>C</td>
<td>Celsius</td>
</tr>
<tr>
<td>K</td>
<td>Kelvin</td>
</tr>
<tr>
<td>µm</td>
<td>Micrometre</td>
</tr>
<tr>
<td>mm</td>
<td>Millimetre</td>
</tr>
<tr>
<td>cm</td>
<td>Centimetre</td>
</tr>
<tr>
<td>m</td>
<td>Metre</td>
</tr>
<tr>
<td>km</td>
<td>Kilometre</td>
</tr>
<tr>
<td>L</td>
<td>Litre</td>
</tr>
<tr>
<td>SLPM</td>
<td>Standard Litres Per Minute</td>
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<tr>
<td>g</td>
<td>Gram</td>
</tr>
<tr>
<td>kg</td>
<td>Kilogram</td>
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<tr>
<td>N</td>
<td>Newton</td>
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<tr>
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<td>Kilonewton</td>
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<tr>
<td>Pa</td>
<td>Pascal</td>
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<td>Megapascal</td>
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<tr>
<td>GPa</td>
<td>Gigapascal</td>
</tr>
<tr>
<td>W</td>
<td>Watt</td>
</tr>
<tr>
<td>MW</td>
<td>Megawatt</td>
</tr>
<tr>
<td>kWh</td>
<td>Kilowatt-hour</td>
</tr>
<tr>
<td>HP</td>
<td>Horsepower</td>
</tr>
<tr>
<td>s</td>
<td>Second</td>
</tr>
<tr>
<td>Hz</td>
<td>Hertz</td>
</tr>
<tr>
<td>kHz</td>
<td>Kilohertz</td>
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<td>Mega-sample</td>
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<tr>
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<td>Milliamp</td>
</tr>
<tr>
<td>A</td>
<td>Amp</td>
</tr>
<tr>
<td>kV</td>
<td>Kilovolt</td>
</tr>
<tr>
<td>VDC</td>
<td>Volts Direct Current</td>
</tr>
</tbody>
</table>
Notation

\begin{itemize}
  \item \textit{CA} \quad \text{Degree crank angle}
  \item \textit{B} \quad \text{Engine bore}
  \item \textit{CA}_{0-10} \quad \text{Crank angle of 0–10\% mass burned}
  \item \textit{CA}_{10-90} \quad \text{Crank angle of 10–90\% mass burned}
  \item \textit{C}_d \quad \text{Flow discharge coefficient}
  \item \textit{CR}_{gc} \quad \text{Combined geometric compression ratio}
  \item \textit{c}_p \quad \text{Constant pressure specific heat capacity}
  \item \textit{c}_V \quad \text{Constant volume specific heat capacity}
  \item \textit{D} \quad \text{Mass diffusivity}
  \item \textit{Da} \quad \text{Damköhler number}
  \item \textit{d}_f \quad \text{Fringe spacing}
  \item \textit{d}_{mv} \quad \text{Measurement volume diameter}
  \item \textit{f}_D \quad \text{Doppler frequency}
  \item \textit{f}_s \quad \text{Frequency shift}
  \item \textit{h} \quad \text{Specific enthalpy}
  \item \textit{h}_c \quad \text{Cylinder clearance height}
  \item \tilde{h}_o, \tilde{h}_f \quad \text{Sensible, formation molar enthalpy}
  \item \textit{K} \quad \text{Karlovitz strain parameter}
  \item \textit{Ka} \quad \text{Karlovitz number}
  \item \textit{k} \quad \text{Thermal conductivity}
  \item \textit{k}_e \quad \text{Turbulent kinetic energy}
  \item \textit{Le} \quad \text{Lewis number}
  \item \textit{l}_k \quad \text{Kolmogorov length scale}
  \item \textit{l}_o \quad \text{Integral length scale}
  \item \textit{\dot{m}} \quad \text{Mass flow rate}
  \item \textit{N}_F \quad \text{Number of fringes in the measurement volume}
  \item \textit{N}_s \quad \text{Number of moving fringes to pass a stationary particle}
  \item \textit{Nu} \quad \text{Nusselt number}
  \item \textit{n} \quad \text{Polytropic index, refractive index, number of moles}
  \item \textit{n}_c, \textit{n}_e \quad \text{Polytropic index of compression, expansion}
  \item \textit{\dot{n}}_e \quad \text{Molar exhaust flow rate}
  \item \textit{Pr} \quad \text{Prandtl number}
  \item \textit{p}, \textit{p}_f, \textit{p}_m \quad \text{Pressure, firing pressure, motoring pressure}
\end{itemize}
### Notation

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\bar{p}_x)</td>
<td>Mean crossover passage pressure</td>
</tr>
<tr>
<td>(p_{max})</td>
<td>Maximum combustion pressure</td>
</tr>
<tr>
<td>(\dot{Q})</td>
<td>Heat-transfer rate</td>
</tr>
<tr>
<td>(Q_{LHV})</td>
<td>Fuel lower heating value</td>
</tr>
<tr>
<td>(R)</td>
<td>Radius</td>
</tr>
<tr>
<td>(Re, Re_T)</td>
<td>Reynolds number, turbulent Reynolds number</td>
</tr>
<tr>
<td>(S_L)</td>
<td>Laminar flame propagation speed</td>
</tr>
<tr>
<td>(\bar{S}_p)</td>
<td>Mean piston speed</td>
</tr>
<tr>
<td>(T)</td>
<td>Temperature</td>
</tr>
<tr>
<td>(T_w)</td>
<td>Wall temperature</td>
</tr>
<tr>
<td>(t)</td>
<td>Time, thickness</td>
</tr>
<tr>
<td>(t_k)</td>
<td>Kolmogorov eddy turnover time</td>
</tr>
<tr>
<td>(u_k)</td>
<td>Kolmogorov eddy velocity</td>
</tr>
<tr>
<td>(u_L)</td>
<td>Laminar burning velocity</td>
</tr>
<tr>
<td>(u_o)</td>
<td>Integral eddy velocity</td>
</tr>
<tr>
<td>(u_T)</td>
<td>Turbulent burning velocity</td>
</tr>
<tr>
<td>(\bar{u})</td>
<td>Mean fluid velocity</td>
</tr>
<tr>
<td>(u_{⊥p})</td>
<td>Particle velocity perpendicular to fringes</td>
</tr>
<tr>
<td>(u')</td>
<td>RMS velocity fluctuations, (u)-component</td>
</tr>
<tr>
<td>(V)</td>
<td>Volume</td>
</tr>
<tr>
<td>(V_d)</td>
<td>Cylinder displacement volume</td>
</tr>
<tr>
<td>(v')</td>
<td>RMS velocity fluctuations, (v)-component</td>
</tr>
<tr>
<td>(\dot{W})</td>
<td>Work per unit time</td>
</tr>
<tr>
<td>(\bar{x}_i)</td>
<td>Wet mole fraction of species (i)</td>
</tr>
<tr>
<td>(x_i^*)</td>
<td>Dry mole fraction of species (i)</td>
</tr>
<tr>
<td>(x_{mv})</td>
<td>Measurement volume shift through planar window</td>
</tr>
<tr>
<td>(\alpha)</td>
<td>Thermal diffusivity, laser beam half-angle</td>
</tr>
<tr>
<td>(\gamma)</td>
<td>Ratio of specific heat capacities</td>
</tr>
<tr>
<td>(\delta_L)</td>
<td>Laminar flame thickness</td>
</tr>
<tr>
<td>(\delta_{L,ph})</td>
<td>Preheat layer thickness of laminar flame</td>
</tr>
<tr>
<td>(\delta_R)</td>
<td>Reaction layer thickness of laminar flame</td>
</tr>
<tr>
<td>(\epsilon)</td>
<td>Turbulence dissipation rate</td>
</tr>
<tr>
<td>(\theta)</td>
<td>Crank angle</td>
</tr>
<tr>
<td>(\theta_{ign})</td>
<td>Crank angle for spark ignition</td>
</tr>
<tr>
<td>(\lambda_0)</td>
<td>Wavelength</td>
</tr>
<tr>
<td>(\mu)</td>
<td>Dynamic viscosity</td>
</tr>
<tr>
<td>(\nu)</td>
<td>Kinematic viscosity</td>
</tr>
<tr>
<td>(\rho)</td>
<td>Density</td>
</tr>
<tr>
<td>(\rho_b)</td>
<td>Burned gas density</td>
</tr>
<tr>
<td>(\rho_u)</td>
<td>Unburned gas density</td>
</tr>
<tr>
<td>(\sigma)</td>
<td>Standard Deviation</td>
</tr>
<tr>
<td>(\phi)</td>
<td>Air/fuel equivalence ratio</td>
</tr>
<tr>
<td>(\Psi)</td>
<td>Ratio of (\delta_L) to (\delta_R)</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

1.1 Overview

Natural gas (NG) offers many advantages over other fuels commonly used for internal combustion engines. These include: a high resistance to auto-ignition, wide flammability limits, stability under environmental extremes, low evaporative emissions, non-corrosive, and low levels of harmful exhaust gases. Natural gas is also globally abundant, making it relatively inexpensive compared with heavily refined fuel oils, such as diesel and gasoline. Despite the broad acceptance of these facts, the North American automotive sector is still largely powered by oil-based fuels. As of 2014, it is estimated that only 112,000 NG vehicles are on the road in the United States, compared to the 14.8 million that exist world-wide [38]. In part, commodity pricing plays a role in the resistance to change. A recent reduction in oil prices led to a 6.5% decrease in the sale of NG vehicles in 2014 [105]. Distribution is another factor, with only 864 public compressed natural gas (CNG) refuelling stations currently operating in the United States [141], and a mere 80 in Canada [31]. But economic and infrastructure issues aside, one of the primary deterrents for engine manufacturers to switch to NG is the notable reduction in performance that follows. When a conventional, gasoline spark ignition (SI) engine is converted to run on NG, the power output is generally reduced by 10–15% [35].

The objective of this dissertation is to address the underlying performance issues associated with burning natural gas in a SI engine. The problem was approached from the point of view that the engine design should meet the requirements of the fuel in terms of delivery, mixing, and combustion. This is in contrast to much research that has been carried out by
converting an engine designed for gasoline or diesel to operate on NG. As such, this work has led to the use of a non-conventional engine architecture, known as a \textit{split-cycle} engine.

A thorough presentation on split-cycle engine operation and its potential benefits will be covered in Chapter 2, but for now it is sufficient that the reader understands its basic operating premise. The term “split-cycle” refers to the four-stroke process of a conventional engine being divided between two adjoining cylinders. One cylinder performs the intake and compression of the gas, the second cylinder receives the compressed gas for combustion (expansion) and subsequent exhausting of the products. The exchange process from one cylinder to the other is a source for generating fluid turbulence, which is beneficial for both air/fuel mixing and fast combustion. Since no commercially available split-cycle engines currently exist, the design and construction of the engine used in this research constitutes a significant part of the dissertation.

Before the application of split-cycle architecture is discussed, it is necessary to understand the cause of NG engine deficiency. The performance reduction primarily stems from two properties of the fuel: the density, and the laminar flame speed.

At normal temperature and pressure (NTP)$^1$, NG is in vapour form with a density that is roughly 60\% that of air, and only 14\% that of gasoline vapour. When NG is injected into the intake manifold it displaces a considerable portion of air that would otherwise enter the engine cylinder, resulting in a reduction to the engine’s volumetric efficiency. There is a slight offsetting advantage from the fact that the stoichiometric (gravimetric) air/fuel ratio of NG is approximately 1.2 times that of gasoline, meaning less fuel is required per unit mass of air. However, this is of little significance given the large discrepancy in density. To put this into perspective, consider an engine cylinder with a 1 litre displacement. A stoichiometric air-fuel ratio would require approximately 1.6\% of the cylinder volume to be filled with gasoline, versus 9.5\% for natural gas. In practice, dynamic flow effects and the absence of latent heat, which is usually present from the evaporation of liquid fuels, further impede cylinder filling. A study performed by Evans and Blaszczyk [53] found a 12\% reduction in brake mean effective pressure (BMEP) when converting a port-injected gasoline engine to run on NG, which they attributed solely to a reduced volumetric efficiency. The reduction has been found to range anywhere from 10\% to 15\%, depending on the operating conditions [28, 126].

With advances in injector technology, the displacement of intake air can be circumvented by injecting the fuel directly into the engine cylinder, referred to as \textit{direct injection} (DI), provided it is done after the intake valve closes. However, due to the small momentum of

$^1$ NTP = 293 K, 101.325 kPa
the gas jet, mixing between the fuel plume and the air is difficult. Therefore, a trade-off exists: injection before intake valve closure (IVC) ensures mixture uniformity, since the turbulence level is high during this period and the mixing time is longer; and conversely, injection after IVC improves volumetric efficiency, but is more likely to generate a stratified mixture [15, 150].

The second property of NG that presents an issue is the laminar burning velocity ($u_L$): defined as the flame velocity relative to the unburned reactants [117]. Natural gas is composed almost entirely of methane (CH$_4$), approximately 96% by volume, 93% by mass$^2$, with the remaining constituents including ethane, propane and butane, as well as small quantities of carbon dioxide, nitrogen and various trace impurities [53]. Of all the hydrocarbon species, methane has the lowest laminar burning velocity, $u_L \approx 0.37$ m/s [142], translating into a reduced rate of reaction for NG combustion. Despite the fact that engines operate in a turbulent combustion regime, the laminar flamelet concept stipulates that a turbulent flame is an aggregate of many asymptotically thin laminar flames [113]. As a consequence, the laminar flame speed remains an important aspect for combustion in engines.

Experimental studies have shown that the combustion duration does indeed increase when fuelling is switched from gasoline to natural gas [53, 60, 76]. To compensate for the increased burn duration, spark timing must be advanced to maintain the location of peak cylinder pressure in the vicinity of piston top dead centre (TDC). When converting an engine from gasoline to NG, without making any changes to the compression ratio, spark timing advances of between 2°CA and 10°CA are required to retain mean-best-torque (MBT) conditions [53]. The inherent problem with advanced combustion phasing is the rise in cylinder pressure before TDC, which requires more compression work and increases heat transfer from the combustion chamber. In a study performed by Jones and Evans [76], the reduction in overall efficiency, caused by the aforementioned parameters, was found to be 5%. For these reasons, it is important to maintain a high rate of combustion when using NG in a SI engine.

It is well known that the rate of combustion can be enhanced by increasing the level of in-cylinder turbulence immediately prior to, and throughout, combustion [39, 61, 70, 94]. In fact, turbulent flame speeds in engines are an order of magnitude higher than their respective laminar flame speed [17], which allows combustion to finish within the allotted time frame. The purpose of this research was to investigate whether the turbulent flow-field inherently produced within the combustion chamber of the split-cycle engine is sufficient to provide a rate of combustion higher than that in conventional SI engines. Since the increment in turbulent flame speed is known to decay with increasing turbulence intensity [22], it was

$^2$ For North America. Concentrations vary globally.
decidedly important to quantify the level of turbulence within the split-cycle engine, thus providing a reference for the pressure-derived combustion rates. Velocity measurements have been made to assess the turbulence characteristics in a physically simulated split-cycle combustion chamber, and the results were used to provide insight into the combustion regime and its limitations.

The use of split-cycle architecture also provided a new option for the physical location of the fuel injector: inside the passage that connects the two engine cylinders together. By locating the injector here, fuel can be introduced after the intake valve has closed—thus not affecting volumetric efficiency—but unlike direct injection, the fuel injector is not subjected to the high temperatures of combustion. Furthermore, by using a novel injection timing, there is still ample amount of time for fuel/air mixing prior to combustion. In essence, this constitutes the best of both configurations.

1.2 Big-Picture Implications

It is reasonable to wonder if the reduced engine performance caused by NG is an acceptable trade-off to benefit from lower fuel costs and reduced exhaust emissions. However, one must further consider the impact the fuel density has on storage and transportation. Figure 1.1 shows a volumetric energy-density comparison between common fuel types, normalized using gasoline at NTP conditions. It can be seen that compressed natural gas (CNG) requires approximately four times the volume of gasoline, and five times the volume of diesel, to produce an equivalent energy content—even when pressurized to 248 bar, which is the standard for many CNG fuel systems [38]. The high storage pressure requires the use of cylindrical pressure vessels, constructed of steel or composite material, which limits conformability to the vehicle chassis and adds considerable weight, meaning the reduced engine performance will be exacerbated by a heavier fuelling system. In addition, packaging constraints are unlikely to allow the fuel tank to be enlarged by 400% for CNG, so reduced range is almost inevitable. Consider, for example, the 2015 Honda Civic: the CNG-powered model has a total range of 311 km (193 miles) per tank, compared to the gasoline-powered model, which achieves 702 km (436 miles) per tank [140]. Thus, it is obvious that the range of NG vehicles should be improved, and better engine efficiency is one means of doing so.

Furthermore, on-going research towards higher-density storage methods for NG decreases the margin of energy-density disparity. Liquefied natural gas (LNG) systems are now available for heavy truck applications [69], which puts the on-board volumetric energy-density
### Energy Density Comparison of Various Fuels

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Normalized Volumetric Energy Density (kJ/m$^3$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gasoline</td>
<td>1</td>
</tr>
<tr>
<td>Light Diesel</td>
<td>1.1</td>
</tr>
<tr>
<td>Heavy Diesel</td>
<td>1.15</td>
</tr>
<tr>
<td>B100 Bio-Diesel</td>
<td>1.08</td>
</tr>
<tr>
<td>Ethanol</td>
<td>0.68</td>
</tr>
<tr>
<td>LPG</td>
<td>0.58</td>
</tr>
<tr>
<td>CNG</td>
<td>0.8</td>
</tr>
<tr>
<td>LNG</td>
<td>0.26</td>
</tr>
<tr>
<td>CGH ($\dagger\dagger$)</td>
<td>0.23</td>
</tr>
</tbody>
</table>

*All standard liquid fuel densities calculated at NTP conditions.

†20 bar, 80% liquid  ‡248 bar, 300 K  §1.3 bar, 111 K  ††680 bar, 300 K

**Figure 1.1:** Energy density comparison of various fuels.

Gasoline is much closer to gasoline, as shown in Figure 1.1. Adsorbed natural gas (ANG)\(^3\) is another promising alternative that allows higher densities to be achieved at a relatively low pressure, without the cryogenic storage complexity of LNG. In either case, the fuel enters the engine in a vaporized state, thus the combustion and emission characteristics discussed within this work can be considered applicable to all NG engines, regardless of how the fuel is stored.

Of greater consequence is the potential for greenhouse gas (GHG) savings associated with using NG in place of gasoline or diesel. The low carbon-to-hydrogen ratio of methane (the primary constituent in NG) means approximately 20% less CO\(_2\) is produced per unit energy of fuel when compared with gasoline\(^4\). This net reduction in CO\(_2\) is not trivial, since regulatory agencies in Canada and the United States have implemented fleet-wide CO\(_2\) emission regulations for all on-road, light-duty vehicles beginning with the 2012 model year. By 2016 the fleet-wide average CO\(_2\) emissions must be below 155 g/km, and further reduced to 100 g/km by model year 2025 [49]. To put this into perspective, the global

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\(^3\) ANG uses a porous carbon medium to store the fuel in its adsorbed state, at a pressure of 35 bar, but with an energy density similar to CNG [27].

\(^4\) Based on calculations using CH\(_4\)/air and iso-octane/air mixtures.
historical trends of light-duty vehicle CO₂ emissions and enacted future regulations⁵ are shown in Figure 1.2. It can be seen that over the next 10 years the total CO₂ emitted by light-duty vehicle fleets in North America is mandated to reduce by approximately 35%.

In light of these future regulations, the inherently lower CO₂ produced from NG combustion makes it an increasingly attractive alternative fuel. If the deficiencies associated with its use in internal combustion engines can be reduced, the potential for mass-market adoption in North America can be improved. Despite the focus on automotive implications, it is important to note that this research also applies to other reciprocating-engine based applications, such as stationary power generation.

### 1.3 Research Objectives

The primary objectives of this research were as follows:

1) To determine experimentally, if split-cycle engine architecture could help alleviate the short-comings of natural gas-fuelled, SI engines. Particular emphasis was placed on enhancing the rate of combustion without compromising the exhaust gas emissions.

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⁵ Enacted regulations are sanctioned by federal governing bodies. In Canada, on-road vehicle emissions are regulated by Environment Canada under the Canadian Environmental Protection Act of 1999 [108].
2) To investigate whether a novel location and timing of the fuel injector, afforded by using split-cycle engine architecture, could improve volumetric efficiency and air/fuel mixing in comparison to conventional NG engines (either port- or direct-injected).

3) To provide some quantifying evidence for the first research objective in regards to the levels of turbulence generated inside the split-cycle engine, and their predicted impacted on combustion.

To accomplish the primary objectives, it was necessary to establish some secondary objectives. These were to:

a) Design, manufacture, and assemble a suitable split-cycle engine.

b) Construct an engine dynamometer test platform and instrument for data collection.

c) Program the software necessary for data acquisition and control of the engine.

d) Generate a 1-D computer simulated engine model to determine the initial valve timing for the engine, and examine how valve timing affects performance.

e) Build a test apparatus that mimics the in-cylinder flow-field conditions of the split-cycle engine's combustion chamber, including optical access for measuring equipment.

1.4 Organization of Thesis

The organization of the remainder of this document is as follows:

Chapter 2: explains in detail how the split-cycle engine operates, along with some additional benefits to its configuration. A review of the performance and emissions characteristics of NG engines is provided in conjunction with the current research that is being explored to improve them.

Chapter 3: details the design of the split-cycle engine with an emphasis on the development of a fast-acting valvetrain.

Chapter 4: covers the construction of the engine test platform, including the data acquisition and engine control systems.

Chapter 5: provides a brief overview of the methodology and metrics used to process and analyse the experimental engine data.
Chapter 6: presents the experimental results obtained from the split-cycle engine. This chapter is interleaved with discussion to provide clarity and relevance of the information at hand. General operating characteristics and limitations are also presented in this chapter.

Chapter 7: presents a 1-D gas dynamic simulation model of the split-cycle engine. Results from the model are compared with empirical data obtained in Chapter 6. The implications of valve timing and crossover passage sizing are also investigated.

Chapter 8: addresses item 3) from the primary objectives list in Section 1.3 by empirically investigating the flow field downstream from a reverse poppet valve under conditions similar to that in the split-cycle engine. The measurements taken in this chapter are used to predict the turbulence properties and corresponding combustion regime present in the split-cycle engine.

Chapter 9: summarizes the work as a whole, and concisely outlines the conclusions and recommendations for future work.
Chapter 2

Background

2.1 The Split-Cycle Engine: An Introduction

2.1.1 Operating Principles

As its name implies, the split-cycle engine separates the standard four-stroke cycle between two interconnected cylinders: one for intake and compression, and the other for combustion and exhaust. Each process requires one stroke of the piston (180°CA), thus a cycle is complete in one crankshaft revolution. In this manner, the number of power strokes per revolution in the split-cycle engine is the same as a conventional four-stroke engine, provided it has an identical number of cylinders. An illustration of the split-cycle process is shown in Figure 2.1, along with a basic description of the cycle process given in Table 2.1. The left-hand side and right-hand side cylinders shown in Figure 2.1 are referred to as the compression cylinder and the expansion cylinder, respectively, in Table 2.1. Nearing the end of the compression stroke, the air in the compression cylinder is transferred into a small intermediary volume that connects the two cylinders together; this will hereafter be referred to as the crossover passage. Note that the configuration of the crossover passage can vary for different split-cycle engines; the information provided in this chapter refers to the type of passage used in this work. Further information on the selection of this particular configuration can be found in Section 2.1.5.

It can be seen from Figure 2.1 that a slight difference in piston phasing between the two cylinders is present, with the expansion cylinder leading the compression cylinder. This allows the expansion cylinder to reach top dead centre (TDC) on its exhaust stroke, and then recede slightly, before the compression cylinder reaches TDC on its compression stroke.
During this time, both crossover valves are opened, simultaneously allowing air in and out of the crossover passage. As such, the mass inside the crossover passage remains constant (ideally), and therefore approximately isobaric. This is somewhat analogous to a tube filled with a single row of marbles; as a marble is inserted in one end, another marble is pushed out the opposite end. The magnitude of the crossover pressure is a function of the geometric compression ratio and the crossover volume itself (see Chapter 7), and can be expected to fall within the range of a typical post-compression cylinder pressure.

In current practice, fuel is injected in one of two ways: at the backside of the crossover outlet valve while it is open [115], or directly into the expansion cylinder [103]. In both cases, the injection timing and/or valve timing must be carefully selected to prevent short-circuiting of raw fuel into the exhaust stream. From the time the exhaust valve closes near TDC, air and fuel must be brought into the expansion cylinder, mixed, and ignited in quick succession before the piston recedes too far on its expansion stroke. The fact that combustion is initiated after TDC is one of the key differences between a conventional SI engine and the split-cycle engine. Once combustion has been initiated by the spark plug, expansion of the hot gases drives the expansion piston down to bottom dead centre (BDC).
The compression piston follows on its intake stroke, after closing of the crossover inlet valve at piston TDC. Once both pistons reach BDC, the subsequent compression and exhaust strokes indicates the start of the next cycle.

From a thermodynamic approach, the idealized split-cycle engine process can be represented on a pressure-volume diagram according to Figure 2.2, where the left- and right-hand plots represent the compression and expansion cylinders, respectively. A description of each process path is given below. Note that the actual engine process is given first, with the idealized, closed-system process given afterwards in parenthesis.

![Pressure-volume diagrams](image)

**Figure 2.2:** Ideal pressure-volume diagrams of a split-cycle engine; compression cylinder (left), expansion cylinder (right). $p_i$ and $p_x$ are the intake port and crossover passage pressures, respectively.

**Compression Cylinder**

1. (1-2) Polytropic compression process (isentropic compression)
2. (2-3) Fluid transfer into crossover passage (isobaric heat rejection)
3. (3-4) Polytropic expansion of residual air (isentropic expansion)
4. (4-1) Intake process (isobaric heat addition)

**Expansion Cylinder**

1. (1-2) Exhaust process (isobaric heat rejection)
2. (2-3) Transfer of pressurized fluid into cylinder (isochoric heat addition)
3. (3-4) Constant volume portion of combustion (isochoric heat addition)
4. (4-5) Constant pressure portion of combustion (isobaric heat addition)
5. (5-6) Polytropic expansion process (isentropic expansion)
6. (6-1) Blow-down process (isochoric heat rejection)
In reality, the transfer process, (2-3) in both diagrams, will not occur under isobaric or isochoric conditions due to the dynamics of the engine. Also, process (3-4) in the compression cylinder may or may not fully re-expand to atmospheric conditions before the intake valve is opened, depending on the intake valve timing. Point 3, in actuality, is unlikely to have the same magnitude for both diagrams, due to heat transfer in the crossover passage. And since cylinder filling requires a finite amount of time, the alignment of points (2-3-4) in the right-hand figure will be impossible to achieve in practice. The same can be said for points (3-4-5), which will be heavily rounded due to heat transfer and the finite duration of combustion, during which time the piston never stops moving.

2.1.2 Split-Cycle Challenges

The foreseeable challenges associated with split-cycle engine architecture originate from the additional heat and mass losses associated with transferring the fluid from one cylinder to the other. To the best of the author’s knowledge, no literature that is specific to split-cycle engines currently exists on the magnitude of these losses. However, it can be expected that the increased surface area of the crossover passage walls, in conjunction with the additional time between compression and combustion, will lead to higher heat losses from the compressed gas. The high flow velocity past the crossover outlet valve may also contribute to higher than normal heat transfer.

Flow losses are expected to primarily depend on the engine’s ability to transfer the entire contents of the compression cylinder into the crossover passage during each cycle. It is for this reason that the split-cycle developed in this work has adopted the use of reverse poppet valves, which is discussed further in Section 3.5. At engine speeds beyond those tested in this work, where the flow time required to fill the expansion cylinder approaches the crossover outlet valve opening time, frictional flow losses may also become an issue.

Mechanical friction losses in a well-designed split-cycle engine are not expected to be significantly more than those of a conventional spark-ignition engine. Between split-cycle and conventional engine architectures, the same number of cylinders exists for an equal number of combustion cycles per revolution. The total number of valves is also the same; however, the spring pressure required for a fast-acting reverse poppet valve is expected to be higher than a conventional valve, resulting in some additional valvetrain friction. It is worthwhile to note that the split-cycle developed in this work has not been optimized to reduce mechanical friction losses. Recommendations for future engine improvements in regards to friction, flow, and heat transfer losses can be found in Section 9.4.
2.1.3 Split-Cycle Advantages

There are numerous advantages to using split-cycle engine architecture. Some are inherent in the basic operating premise of the engine, while others must be sought after through additional means. The latter will be discussed briefly here, but were not utilized in this work, as it will be explicitly stated. The following advantages are in no particular order.

In-Cylinder Turbulence

As mentioned in Chapter 1, in-cylinder turbulence is an effective way of increasing the rate of combustion in a SI engine. It also promotes extensive fluid mixing, which is beneficial for charge homogeneity. The standard methods for turbulence generation in engines, as well as the interaction between turbulence and combustion, will be addressed in Section 2.4. For now, it is important that the reader understand three aspects concerning turbulence and engines:

i) The size and intensity of turbulence are both important. Generally speaking, small turbulent eddies combined with high velocity fluctuations (i.e. intensity) is most effective at increasing the rate of combustion [134].

ii) Turbulence is a naturally decaying phenomenon, and as such requires constant energy input to be sustained.

iii) The magnitude of turbulent velocity fluctuations ($u'$) in the combustion chamber of a conventional engine, at the time of ignition, is linearly related to the engine speed in accordance with Equation (2.1) [70], where $\bar{S}_p$ is the mean piston of the engine.

$$u' \approx \frac{1}{2} \cdot \bar{S}_p$$

(2.1)

The split-cycle engine addresses all three aspects in a positive manner. When the crossover valve is designated to open, the pressure in the crossover passage is considerably higher than the cylinder pressure (e.g. $\sim 20:1$). The ensuing flow will therefore be choked, resulting in very high discharge velocities and presumably velocity fluctuations.

Since the flow occurs when the piston is in the vicinity of TDC, the clearance height of the combustion chamber will constrain the size of the turbulent eddies to be very small. And by initiating combustion immediately after the flow period has ended (or even slightly before), there is little time for turbulence decay.
Lastly, because the crossover passage remains at high pressure regardless of engine speed, the resulting discharge flow into the combustion chamber will always be choked for a finite period of time. Consequently, the turbulence intensity is largely decoupled from the mean piston speed, meaning high intensities can be achieved at low engine speeds.

**Polytropic Compression Index**

A polytropic process, by definition, abides by the pressure-volume relationship: \( pV^n = \text{constant} \). For the special case of isentropic compression, the polytropic index, \( n \), is equal to the ratio of specific heat capacities for constant pressure, \( c_p \), and constant volume, \( c_V \), processes, denoted as \( \gamma \):

\[
\gamma = \frac{c_p}{c_V}
\]

(2.2)

Since \( \gamma \) is a property of the fluid, its value differs between pure air and an air-fuel mixture. Take for example, air and a stoichiometric methane-air mixture, which have \( \gamma \) values of 1.40 and 1.35, respectively. If both gases are compressed in the cylinder of an engine that has a compression ratio of 10:1, and assuming the pressure at the start of compression is atmospheric, the final pressure is approximately 3 bar higher for the pure air case, as shown in Figure 2.3. It is therefore beneficial to introduce the fuel after the compression process has been completed—a characteristic of conventional diesel engines. However, diesel engines rely on non-premixed combustion, unlike SI engines that generally require mixture homogeneity for combustion stability and low emission of exhaust gas pollutants. Since the working fluid in a split-cycle engine is transferred to a second cylinder after compression, there is an opportunity to inject the fuel post-compression and still have a premixed charge.

![Figure 2.3: Effect of polytropic index on isentropic compression of an ideal gas.](image)
**Chapter 2: Background and Hypothesis**

Thermal Segregation

The air entering the cylinder of a conventional four-stroke engine is at a relatively low temperature compared with its new environment, which was subject to combustion temperatures in excess of 2000 K approximately 360 °CA beforehand. As the air is heated by the surrounding surfaces and residual gases, its specific volume increases and prevents additional air from entering the cylinder. The problem is compounded as heat conducted through the intake port and manifold causes additional charge heating, which can ultimately account for up 50% of engine’s volumetric inefficiency [70].

The split-cycle engine has the distinct advantage that combustion does not occur in the compression cylinder, and therefore can be maintained at a lower average temperature. The cooler, denser charge that results can be expected to improve volumetric efficiency, and also increases the knock margin, which allows for a higher compression ratio to be used. Segregation of the water jackets and even separate cooling systems could be used to maintain differing optimal temperatures for each cylinder. The effect of segregated engine cooling, however, is beyond the scope of this work.

Atkinson Cycle

The term *blow-down* is often used to describe the sudden drop in cylinder pressure when the exhaust valve opens. Its magnitude is indicative of the amount of under-utilized expansion work that remains in the cylinder. In other words, additional work could be extracted if the exhaust valve remained closed and the combustion gases continued to be expanded, until the cylinder pressure reached atmospheric conditions. One means of accomplishing this task is known as the Atkinson cycle, shown as an extension to the ideal Otto cycle in Figure 2.4. The integral area shaded in grey is the additional work available when the standard power stroke (3–4) is further expanded to point (4*). In theory, the blow-down process (4–5) can be reduced to zero when point (4*) converges to (5*).

In actuality, implementation of the Atkinson cycle in a conventional engine is physically challenging since the compression and expansion ratios are mechanically linked; increasing one means increasing the other. Currently, Atkinson cycles are achieved by using late intake valve closure—at the cost of volumetric efficiency—or a complex variable-compression-ratio mechanism, like those shown in Wos et al. [148].

By using separate cylinders, the split-cycle engine decouples the compression and expansion ratios allowing the approximated Atkinson cycle to be more easily implemented through
differing cylinder displacement volumes. Branyon and Simpson [23] performed 1-D simulations on the effect of downsizing the compression cylinder in a boosted split-cycle engine configuration as a means of achieving full expansion of the exhaust gases. Their results show low values of brake specific fuel consumption (BSFC) were achieved by using higher boost pressures and decreasing the stroke length of the compression cylinder. However, no comparison was made to the same engine with equal compression and expansion ratios.

In the present work, no attempt at using an Atkinson cycle was made in order to maintain simplicity. The same bore/stroke combination was employed for both cylinders, and the difference between the compression and expansion ratio is a consequence of small piston recesses required for the spark plug and exhaust valve. The geometric compression/expansion ratios are listed in Table 3.1 of Chapter 3.

**Regenerative Technologies**

The split-cycle engine configuration is also well-suited for implementing regenerative technologies, such as air-hybridization and regenerative heat transfer. The former refers to the storage of compressed air during periods of surplus energy, and subsequently utilizing that air for combustion when the energy demand returns. Several patents have been filed in regards to the use of a secondary reservoir connected to the crossover passage of a split-cycle engine [97, 98, 123]. The basic idea is that the crossover outlet valve can be disabled and air can be compressed into the storage tank when combustion is not needed (e.g. during vehicle braking). When the requirement for combustion returns, the compression cylinder can be disabled and the pressurized air for combustion would come from the storage tank.
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Regenerative heat transfer refers to using the exhaust gas for heating of the crossover passage. Provided autoignition and engine knock can be suppressed, this is a viable option for recapturing otherwise wasted energy.

To the extent of the author’s knowledge, neither of the aforementioned concepts have been physically investigated, and are well beyond the scope of this work.

2.1.4 Relevant Literature

The concept of a split-cycle engine dates back to circa 1902, with patents showing two cylinders connected together in varying configurations, discussion of differing cylinder sizes, and even mention of regenerative air storage [8, 146]. Despite its early inception, very few peer-reviewed articles have been published on the topic of split-cycle engines until recently.

Most applicable is the work done by Phillips et al. [115], who conducted 1-D/3-D numerical simulations of a split-cycle engine and evaluated the emissions and performance characteristics of a physical split-cycle engine prototype. Their research focused on the use of gasoline as the primary fuel, in both naturally aspirated and turbocharged applications. Computational fluid dynamic (CFD) simulations were reported to show a high level of turbulent kinetic energy (TKE) in the combustion chamber, although no actual values were given. Empirical combustion durations (CA$_{10-90}$) were found to be on the order of 23°CA for the naturally aspirated version, and 12°CA for the turbocharged version, which can be considered relatively quick [70]. Minimum brake specific emissions of nitrogen oxides (NO$_x$), unburned hydrocarbons (UHCs), and carbon monoxide (CO) were found to be approximately 2.0, 4.5, and 25 g/kWh for stoichiometric operation, respectively. These values were all measured at low load, <4 bar BMEP, and with the exception of CO, align well with typical values for current SI engines. The abnormal CO values were explained to be the result of charge stratification, caused by inadequate air/fuel mixing. The injector placement in their engine had the fuel spray impinging on the back side of the crossover outlet valve—similar to how port injected engines spray fuel on the back side of the intake valve—and dosing occurred while the valve was in the open position. Consequently, the fuel had a very limited amount of time to mix with the air, resulting in poor homogeneity.

Another study by Musu et al. [103] introduced the concept of “homogeneous charge progressive combustion”, where pre-compressed air is gradually fed into the combustion chamber as a means of controlling the heat release rate. Diesel fuel is injected into the connecting passage, which is only valved on the compressor side, and combustion relies on auto-ignition
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of the fuel. Their numerical work shows an absence of the characteristic ‘double-hump’ pressure trace, indicating a more pre-mixed type of burn. While no turbulence characteristics are given, it can be hypothesized that the lack of diffusion combustion is caused by excellent and rapid mixing of the fuel spray, which is most likely caused by the high-pressure flow discharging from one cylinder to the other. Relatively high indicated mean effective pressure (IMEP) values, on the order of 20 bar, were also realized in their CFD simulations.

On a much larger scale, Coney et al. [36] outline the details of 3 MW reciprocating-piston, power generation engine that utilizes a split-cycle configuration. By segregating the compression cylinder from combustion, in-cylinder water injection could be used to approach quasi-isothermal compression of the combustion air, which significantly reduced the compression work and allowed for waste heat recovery from the exhaust later in the cycle. No details of combustion were provided.

2.1.5 Selection of Split-Cycle Configuration

The process of transferring the working fluid from one cylinder to the other in a split-cycle engine has been attempted in different ways: through a passage with a relatively large volume and valved on either end [115]; through a small passage valved on the inlet side only [103]; and through a valved orifice with minimal volume (opposed piston design) [136].

In order to meet the research objectives outlined in Chapter 1, a crossover passage valved at either end was selected for this work. Having valves on both the inlet and outlet of the crossover passage allowed the gas inside to remain at high pressure, providing an extended interim period between gas compression and combustion. By injecting the fuel directly into the crossover passage, it was hypothesized that a well-mixed charge could be created upstream from the combustion chamber without affecting the intake air flow rate. While any configuration utilizing fuel injection after the compression stroke would have satisfied the volumetric efficiency aspect, the poor mixing shown by Phillips et al. [115] led the author to use a valved passage to increase the mixing time. The design of the crossover passage, combined with a novel fuel injector placement (see Section 3.4.7), were used as additional means to promote mixing. The turbulence required for combustion was expected to be high regardless of the crossover configuration; however, having a valved passage at constant pressure provides additional control of the flow process relative to the start of combustion.
2.2 Emissions from Spark-Ignition Engines

Nitrogen oxides (NO\textsubscript{x}), carbon monoxide (CO), carbon dioxide (CO\textsubscript{2}), particulate matter (PM), and unburned hydrocarbons (UHC) are all regulated emissions produced by internal combustion engines. This section is intended to provide a brief overview of the mechanisms through which these emissions are generated, and assess their significance in relation to NG engines. The topic of engine emissions is very extensive, and the information provided here is by no means comprehensive.

2.2.1 Carbon Dioxide (CO\textsubscript{2})

It was mentioned in Chapter 1 that the low carbon-to-hydrogen ratio of methane (CH\textsubscript{4}) effectively reduces the CO\textsubscript{2} output from natural gas engines. This is most easily illustrated using a simple example. If CH\textsubscript{4} and C\textsubscript{8}H\textsubscript{18} (iso-octane) are taken to represent NG and gasoline, respectively, the balanced stoichiometric chemical equations can be written as:

\[
\text{CH}_4 + 2(\text{O}_2 + 3.76\ \text{N}_2) \rightarrow \text{CO}_2 + 2\text{H}_2\text{O} + 7.52\ \text{N}_2 \\
1 \text{ mol CH}_4 \rightarrow 1 \text{ mol CO}_2 \\
1 \text{ mol} \times (12 + 4) \text{ g/mol} \rightarrow 1 \text{ mol} \times (12 + 32) \text{ g/mol} \\
1 \text{ g CH}_4 \rightarrow 2.75 \text{ g CO}_2
\]

\[
\text{C}_8\text{H}_{18} + 12.5(\text{O}_2 + 3.76\ \text{N}_2) \rightarrow 8\text{CO}_2 + 9\text{H}_2\text{O} + 47\ \text{N}_2 \\
1 \text{ mol C}_8\text{H}_{18} \rightarrow 8 \text{ mol CO}_2 \\
1 \text{ mol} \times (12 \cdot 8 + 18) \text{ g/mol} \rightarrow 8 \text{ mol} \times (12 + 32) \text{ g/mol} \\
1 \text{ g C}_8\text{H}_{18} \rightarrow 3.09 \text{ g CO}_2
\]

By solving for the mass of CO\textsubscript{2} in the products, and normalizing it by the fuel mass, it can be seen that the complete combustion of methane will theoretically produce 11\% less CO\textsubscript{2} in comparison to iso-octane. This margin is further increased to 21\% when normalized by each fuel’s heating value: 50 MJ/kg for NG, and 44.3 MJ/kg for iso-octane. Since CO\textsubscript{2} is a product of complete combustion (a highly desirable attribute), the only means of reducing its output is to burn less fuel, or use a fuel with a lower C:H ratio, as shown above.

Experimental measurements by Atibeh et al. [9] showed a 5–25\% reduction in CO\textsubscript{2} per unit work, when they switched their bi-fuel engine from gasoline to natural gas. The study went a step further, using the empirical engine data to simulate a New European Driving Cycle.
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(NEDC), and found the CO₂ emissions for NG were 17% lower than gasoline. The simulation findings can be considered approximate, however, since considerable interpolation was used to generate the virtual engine “map”, and cold starting was neglected.

A similar study by Douailler et al. [47] compared the output of a diesel engine to a similar sized DI NG engine on the same virtual NEDC drive cycle. They reported 27% less CO₂ emissions from the NG engine compared with the diesel engine. To be fair, the NG engine had a smaller displacement (1.5 L vs. 2.0 L), but for a given load, still produced 22–25% less CO₂ than the diesel engine, and 30% less than a 2.0 L gasoline engine.

2.2.2 Oxides of Nitrogen (NOₓ)

The term NOₓ collectively refers to both nitric oxide (NO) and nitrogen dioxide (NO₂), the former being the predominant oxide produced inside the combustion chamber of an engine [70]. Once released into the atmosphere NO will quickly oxidize to NO₂, which becomes a significant source of smog and a precursor to acid rain in the presence of ultraviolet radiation and atmospheric hydrocarbons [117, 131]. Due to the adverse health effects of NO₂, federal regulatory agencies, such as the EPA, dictate the acceptable level of NOₓ emissions based on a standardized drive cycle [120]. For example, light-duty, Tier-3 emissions are set to reduce the combined sum of non-methane organic gases (NMOG) and NOₓ from today’s fleet average of 100 mg/km to 19 mg/km by year 2025 [50].

The bulk formation of NO comes from the oxidation of molecular nitrogen through one of three primary routes:

1. Post-flame or thermal NO
2. Flame or prompt NO
3. Oxidation of nitrogen-containing elements in the fuel

For a low nitrogen-content fuel, like NG, the primary means of NO formation is from high temperatures (thermal NO) and can be described by three governing reactions, known collectively as the extended Zeldovich mechanism [70, 117, 131]:

\[
\begin{align*}
O + N₂ & \longleftrightarrow NO + N \quad (2.3) \\
N + O₂ & \longleftrightarrow NO + O \quad (2.4) \\
N + OH & \longleftrightarrow NO + H \quad (2.5)
\end{align*}
\]
The forward reaction of (2.3) is a limiting-step reaction that provides monatomic nitrogen for (2.4) and (2.5). The predicted equilibrium formation rate of NO in (2.3) is calculated from:

\[
\frac{d[NO]}{dt} = k^+[O][N_2]
\]  

(2.6)

where the reaction rate constant, \(k^+\), is an exponential function of the Arrhenius form:

\[
k^+ = AT^\beta e^{-E/R_0 T}
\]  

(2.7)

and \(A = 1.6 \times 10^{13} \text{ cm}^3/\text{mol} \cdot \text{s}^{-1}\), \(\beta = 0\), and \(-E/R_0 = 38000\) K [131].

If \(k^+\) is plotted as a function of temperature, as shown in Figure 2.5, it can be seen that the reaction rate increases rapidly beyond \(\sim 1900\) K. Under stoichiometric conditions, the majority of fuels have adiabatic flame temperatures above \(2000\) K [117], and will therefore favour NO\(_x\) production. Combustion strategies that achieve temperatures below this NO\(_x\) threshold are referred to as low temperature combustion (LTC).

![Figure 2.5: Dependence of NO\(_x\) formation on temperature.](image)

Inside the combustion chamber of an SI engine, NO\(_x\) formation begins with the start of combustion and rises with the on-going heat release. As the pressure and temperature begin to decrease during the expansion stroke, the NO\(_x\) levels freeze and remain well above the predicted equilibrium conditions of the exhaust [70]. The highest levels of NO\(_x\) are found in the earliest burned region of the combustion chamber, which is subject to a continual heat flux from the propagating flame [90]. This suggests the importance of not only the
peak value, but also of the time-history of the burned-gas temperature in the search for lower NO\textsubscript{x} emissions.

A typical remedy for reducing NO\textsubscript{x} in SI engines is through charge dilution; either with excess air (lean air/fuel ratio) or exhaust gas recirculation (EGR). Both methods effectively increase the specific heat capacity and thermal (non-reacting) mass of the combustion chamber, reducing the peak combustion temperatures and thus NO\textsubscript{x} emissions. At the same time, the lower temperatures decrease the reaction rate and so it is important to have a fast-burning, robust combustion system, which allows higher dilution rates to be achieved.

Korakianitis et al. [87] summarized the level of nitrogen oxide emissions from NG engines as generally being higher than those from gasoline engines. The reason being high compression ratios are often employed to utilize the greater octane rating of NG (RON\textsuperscript{1} ≈ 120), which results in a higher peak cylinder pressure/temperature. Korakianitis also stipulated that many of the NG engines producing high NO\textsubscript{x} levels could significantly benefit from proper tuning. A study done in Brazil examined the before and after emissions of gasoline vehicles retrofitted to run on NG, and is perhaps a testament to Korakianitis’ theory; average NO\textsubscript{x} levels were found to increase by 171\%, despite the use of sanctioned hardware [46].

When an engine was switched from gasoline to NG (with no compression ratio change), and adjustments in spark timing were made to maintain MBT conditions, Evans [53] demonstrated that NO\textsubscript{x} emissions remained roughly the same. A similar study by Geok et al. [60] showed NO\textsubscript{x} emissions from NG engines to be both higher and lower than their gasoline counterpart over a range of engine speeds, but trends in the data are questionable and unexplained.

### 2.2.3 Carbon Monoxide (CO)

A complete combustion reaction between any hydrocarbon and air will ultimately produce carbon dioxide (CO\textsubscript{2}) and water vapour (H\textsubscript{2}O). An important intermediary step in this global reaction, is the oxidization of carbon monoxide (CO) with hydroxide radicals (OH) [91], shown in (2.8). The production of OH radicals is strongly temperature dependant [145]. For this reason, excessively-diluted mixtures (e.g. a very high EGR rate) can result in sufficiently low temperatures within the reaction zone to increase CO emissions [70].

$$\text{CO} + \text{OH} \rightarrow \text{CO}_2 + \text{H} \quad (2.8)$$

\textsuperscript{1} Research Octane Number
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However, under normal operating conditions, and reasonable levels of dilution, CO formation is primarily caused by insufficient oxygen available for OH radical formation. It is for this reason that CO emissions are most predominant for rich, or heavily stratified, air/fuel mixtures [70]. Under lean conditions, CO emissions are generally low, although they can be substantially higher than the levels predicted by equilibrium kinetics [90]. This is likely due to flame quenching on the cold surfaces of the combustion chamber [70].

In practice, several studies have shown that CO emissions are reduced on the order of 50% when fuelling is switched from gasoline to NG [46, 53, 124]. Typical values range from 10–25 g/kWh at stoichiometric conditions, to 2–5 g/kWh at lean conditions [10, 53, 95, 150].

2.2.4 Unburned Hydrocarbons (UHCs)

Unburned hydrocarbons (UHCs) are the result of incomplete combustion in SI engines and amount to roughly 1–2.5% of the fuel flowing into the engine [70]. The mechanisms that cause these emissions are: boundary layer and crevice volume quenching, oil layer absorption/desorption, exhaust valve leakage, short-circuiting, and fuel deposits. According to Cheng et al. [33], approximately 9% of fuel injected into the engine escapes normal combustion, over one-half of which is caused by the inability of the flame to propagate into crevice volumes. A breakdown of the fuel escape paths is shown in Table 2.2. Most of these HCs will still be oxidized at a later time (e.g. during the expansion stroke).

<table>
<thead>
<tr>
<th>UHC Pathway</th>
<th>% of Fuel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oil Layers</td>
<td>1%</td>
</tr>
<tr>
<td>Deposits</td>
<td>1%</td>
</tr>
<tr>
<td>Short-circuit</td>
<td>1.2%</td>
</tr>
<tr>
<td>Quenching</td>
<td>0.5%</td>
</tr>
<tr>
<td>Crevices</td>
<td>5.2%</td>
</tr>
<tr>
<td>Valve Leakage</td>
<td>0.1%</td>
</tr>
</tbody>
</table>

Table 2.2: Breakdown of unburned hydrocarbon pathways [33].

The split-cycle engine has a distinct advantage in oil layer HCs since fuel is not introduced into the cylinder until the piston is near TDC, leaving very little cylinder wall exposed to the fuel. Despite the fact that the oil layer absorption only represents 1% of the incoming fuel, the subsequent desorption constitutes 16% of the THC emissions leaving the engine [33]. The low levels of combustion chamber carbon build-up associated with NG fuelling [55] is also expected to reduce HCs originating from these deposits.
Hydrocarbon emissions are segregated into two categories: methane hydrocarbons and non-methane hydrocarbons (NMHC). The former typically accounts for 80–90% of total hydrocarbon (THC) emissions from a NG engine [53]. Although methane is a strong GHG—about 21 times the global warming potential of CO\textsubscript{2} over a 100 year period [38]—it has low photochemical reactivity, which is the primary means of smog production. In other words, the contribution of methane emissions to smog is negligible in comparison to non-methane hydrocarbons, which only constitute 10–20% of THC emissions from a NG engine.

Because HC emissions are sensitive to both the operating conditions and the engine configuration, it is difficult to predict if HC emissions for a NG engine will be less than those for a comparable gasoline engine. The direct comparison made by Evans [53] showed that HC emissions decreased by 50% at full load conditions when operating on NG, but remained roughly equivalent to the gasoline-fuelled HC emissions at low load. This trend has been confirmed by others, who show decreasing HC levels at higher loads for NG fuelling [10, 35]. The same studies also indicate HC levels increase with greater EGR fractions. These findings would seem to indicate a HC sensitivity with combustion temperature, which would make sense given the high activation energy of methane. For stoichiometric conditions, HC emissions from SI engines fuelled with NG were found to range from 1 g/kWh to 12 g/kWh [10, 35, 53, 95, 122].

2.2.5 Particulate Matter and Toxic Emissions

Natural gas exhaust emissions contain no significant amount of toxins, such as particulate matter (PM), benzene, or 1,3-butadiene [124].

2.2.6 Emission Reduction Strategies

The reduction of engine-produced emissions is generally performed in one of two ways: prevention at the source (i.e. in-cylinder reduction), and/or after-treatment of the exhaust stream. This section will briefly outline the fundamental strategies commonly employed in both categories as they pertain to SI engines, and their relevance to split-cycle and NG combustion. Again, the topic of emissions reduction is very broad, and only the very basic elements are covered here.

Based on the information provided in Sections 2.2.1 to 2.2.5, it can be surmised that in order to obtain low engine-out emission levels the peak combustion temperatures should be low (i.e. less than \(\sim 2000\)K), the air/fuel stoichiometry should be accurately controlled,
flame quench regions should be minimized, and mixture homogeneity should be achieved.\textsuperscript{2} Air/fuel metering and mixing are a function of engine design, as is the minimization of flame-quenching crevice volumes. The reduction of peak temperature, as it was mentioned in Section 2.2.2, is primarily accomplished through mixture dilution, either via exhaust gas recirculation (EGR) or lean air/fuel mixtures. Both of these methods affect more than just NO\textsubscript{x} emissions, and thus will now be discussed in more detail.

Figure 2.6 shows the dependence of NO\textsubscript{x}, CO, and UHC emissions on the equivalence ratio for a typical SI engine. Under lean conditions ($\phi < 1.0$), both CO and UHC can be minimized since there is a surplus of O\textsubscript{2} to ensure their oxidation. NO\textsubscript{x}, however, initially increases under lean conditions, reaching a peak around $\phi \approx 0.9$, and then decreasing thereafter. Despite combustion temperatures being highest around stoichiometric conditions ($\phi \approx 1.0$), the maximum rate of NO\textsubscript{x} formation occurs on the lean side due to the greater availability of O\textsubscript{2}. Increasingly lean conditions then lower NO\textsubscript{x} output by reducing the flame temperature. It is therefore beneficial to operate as lean as possible in order to reduce NO\textsubscript{x} and CO, provided UHCs do not increase significantly from incomplete combustion.

Exhaust gas recirculation refers to a fraction of the combustion products being re-introduced back into the cylinder alongside the fresh mixture. The composition of EGR is therefore mostly N\textsubscript{2}, CO\textsubscript{2}, and H\textsubscript{2}O, which gives it a relatively high heat capacity and makes it

\textsuperscript{2} Mixture stratification is a design feature of certain systems, but beyond the scope of this discussion.
largely inert to combustion. As such, the fraction of EGR in the cylinder acts to increase the thermal mass, effectively reducing the peak temperature of combustion. Exhaust gas recirculation can be accomplished by trapping the exhaust gases directly in the cylinder (often referred to as residuals), or externally recirculating them back into the intake manifold. The latter has the advantage that the EGR gases can be cooled in the interim, which further suppresses NO\textsubscript{x} emissions. Provisions were implemented for external EGR use in this work (see Section 4.2.4), but to date no experiments have been conducted.

Dilution from EGR and/or lean mixtures has the undesirable effect of reducing the burn rate. Consequently, the application of such strategies is limited by combustion stability, which deteriorates as the charge is made leaner or the EGR fraction becomes larger. This is depicted in Figure 2.6 by the rise in UHC towards the lean end of the spectrum, which is indicative of incomplete combustion. For a spark-ignition engine operating at part load, EGR in the 15–30\% range is generally tolerable, and can reduce NO\textsubscript{x} emissions by up to 60\% [70, 120]. Similar to the lean-burn scenario, EGR in excessive amounts results in high cyclic variation caused by incomplete combustion [131]. The exact amount of EGR or lean air/fuel ratio that can be tolerated depends on numerous factors, such as fuel type, engine load, and combustion chamber design. However, it is well known that a faster burning combustion chamber can withstand higher amounts of dilution, allowing for increased EGR rates and leaner mixtures to be used [70, 131]. As such, fast-burn engines have a higher potential to achieve low emissions via charge dilution. This was another motivating factor for the use of split-cycle engine architecture at the inception of this dissertation.

In contrast to in-cylinder emissions control, treatment of the exhaust gases downstream from the engine is still the most commonly used form of pollution reduction in vehicles. For an SI engine, after-treatment generally involves the use of a three-way catalytic converter (TWCC), which consists of a ceramic or metallic monolithic substrate that is covered in a catalytic washcoat and placed in the exhaust stream of the engine. The washcoat consists of precious metals—most commonly platinum, rhodium, and palladium—embedded into a highly porous carrier material such as $\gamma$-Al\textsubscript{2}O\textsubscript{3} [54], and promotes the reactions shown in (2.9) to (2.12) [86].

\[\text{CO} + \frac{1}{2} \text{O}_2 \rightarrow \text{CO}_2 \] (2.9)

\[C_xH_y + (x + \frac{y}{2})\text{O}_2 \rightarrow x\text{CO}_2 + \frac{y}{2}\text{H}_2\text{O} \] (2.10)

\[2\text{CO} + 2\text{NO} \rightarrow 2\text{CO}_2 + \text{N}_2 \] (2.11)

\[C_xH_y + (2x + \frac{y}{2})\text{NO} \rightarrow x\text{CO}_2 + \frac{y}{2}\text{H}_2\text{O} + (x + \frac{y}{4})\text{N}_2 \] (2.12)
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It can be seen that these reactions effectively convert CO, HC, and NO into CO$_2$, H$_2$O, and N$_2$. Provided the air fuel ratio is tightly controlled around stoichiometric conditions, oscillating between rich and lean within the range of $\phi = 1.00 \pm 0.02$ at 0.5–1 Hz, emission reduction efficiencies greater than 90% can be achieved [70, 131]. The requirement for HC species in the exhaust stream (Reactions (2.10) and (2.12)) prohibit the use of the TWCC under lean-burn conditions.

Interestingly, a few experimental studies performed on natural gas engines have concluded that high rates of EGR in conjunction with a TWCC were superior in reducing NO$_x$ and HC emissions versus the lean-burn strategy [109, 122]. In this work, no exhaust catalyst was used and the emissions measurements presented are engine-out values. The decision to investigate both stoichiometric and lean mixtures was based on the fact that both methods are practical for achieving low emissions, and were uncharted in a NG split-cycle engine.

2.3 Natural Gas Engine Technology

Various approaches to solving the technical issues associated with the combustion of natural gas in reciprocating engines have been explored. This section is intended to review the most prominent of these approaches, both historic and current, and discuss the potential advantages and disadvantages of each.

2.3.1 Combustion Chamber Design

Some of the earlier work devoted to improving the performance and emissions characteristics of NG engines focused on combustion chamber design. Johansson and Olsson [75] evaluated the performance of nine different bowl-in-piston (BIP) chamber geometries and concluded that a higher squish-to-bore area ratio yielded the greatest turbulence intensity during the bulk phase of combustion or 10–90% mass fraction burned (MFB). This greater turbulence correlated well with higher heat release rates and retarded mean-best-torque (MBT) timing, indicating these chambers had a much faster burn. In a secondary study by the same authors [109], it was discovered that the fast-burning chambers from [75] were also the ones with the lowest indicated fuel conversion efficiency. The authors concluded that the higher heat transfer rates associated with the increased in-cylinder turbulence was to blame. As one might expect, increased cylinder turbulence greatly affects the convective heat transfer coefficient and thus a trade-off of between turbulent-induced rapid combustion and higher heat losses exists. The same study also showed that the combustion chamber surface area
becomes increasingly important under lean-burn conditions, when the emissions levels are low and the effects of wall quenching are more pronounced.

Of particular interest in the aforementioned studies are the results from the flat or pancake style combustion chamber, which had the highest indicated efficiency and lowest emissions levels, but a poor heat release rate due to the absence of squish-generated turbulence. Based on the bowl-in-piston geometry used in the study, this is perhaps a good indication that NG engines utilizing chamber geometry designed for compression ignition (CI) spray combustion are not ideal for the propagation of a spark ignited flame. It also affirms that the pancake style chamber is beneficial for reducing surface area and crevice volumes, which increase heat transfer and HC emissions, respectively, but requires some extraneous source for turbulence generation. This is relevant to split-cycle combustion chamber design, since a minimal cylinder clearance volume (i.e. pancake chamber) is desirable to help fully expel residuals, and turbulence generation is largely independent of chamber geometry.

In a similar study, Yu et al. [149] evaluated the turbulent kinetic energy (TKE) of two bowl-in-piston combustion chambers through CFD simulation and tested both designs in a physical engine. The sole difference in TKE between the two designs occurred at TDC, and higher amounts of TKE reduced the overall combustion duration and brake specific fuel consumption (BSFC). However, the early flame development time was longer at higher levels of TKE, which indicates difficult conditions for flame kernel development. The interaction between combustion and fluid motion will be discussed in Section 2.4.2, but suffice it to say that this supports the notion of a practical limit for turbulence intensity in SI engines.

2.3.2 Hydrogen Supplementation

One method of improving the combustion rates in NG engines is to supplement the fuel with hydrogen (H$_2$), which has a laminar burning velocity roughly one order of magnitude larger than NG ($u_{L,H_2} \approx 3$ m/s). Mixtures containing anywhere from 5% to 55% hydrogen by volume, with a balance of natural gas or pure methane, have been investigated [16, 78, 95, 122, 138, 143, 152]. The higher laminar flame speed of the mixture allows the ignition timing to be retarded, which decreases heat transfer and compression losses. Hydrogen also has a high energy density (120 MJ/kg), and smaller quenching distance than NG [143].

A review of the aforementioned literary sources revealed that the largest gains to be had from H$_2$ supplementation are for engines that do not possess a fast-burn combustion chamber. Baratta et al. [16] investigated the effects of H$_2$ fuel blending on a 1.4 L turbocharged SI NG engine that was already capable of achieving relatively short 10–90% MFB durations; on the
order of 30°CA. When H$_2$ was introduced in fractions up to 25%, virtually no change in the bulk combustion duration was found over a wide range of engine loads. In slower burning engines, like the one used by Wang et al.[143], where 10–90% MFB durations ranged from 40–65°CA, the addition of 35% H$_2$ reduced the combustion durations by 5–10°CA. The faster combustion rates allowed MBT timing to be retarded, improving thermal efficiency several percent at low load and a negligible amount at high load. Thus, the effectiveness of H$_2$ supplementation is heavily dependent on the existing burn rates of each unique engine.

However, for very lean air/fuel ratios, the use of hydrogen has been shown by Tunestål et al. [138] to better maintain the high rates of combustion obtained at richer mixtures. At $\phi = 0.56$ and for 15% H$_2$, the authors report a 10–90% MFB duration reduction by approximately 10°CA. Furthermore, Ma et al. [95] proved that leaner air/fuel mixtures can be utilized with H$_2$ supplementation when they extended the lean operating limit of a NG engine from $\phi = 0.58$ to $\phi = 0.42$ with a 50/50 H$_2$/NG mixture. Another study by Saanum et al. [122] showed a similar result for high levels of EGR, where combustion rates and emissions were generally improved with the addition of H$_2$. It is therefore evident that one of the main benefits to using H$_2$-blended NG is an increase in combustion stability for very dilute mixtures, which allows extended operating limits to be achieved.

H$_2$ supplementation has also been found to affect the emissions characteristics of NG engines. In general, when H$_2$ is introduced into the fuel, HC emissions decrease, CO emissions remain largely the same, and NO$_x$ emissions either increase or stay the same [16, 95, 122, 138, 143]. The reduction in HC emissions is likely a combination of the higher flame temperature and reduced quench distance of H$_2$. The former provides a greater overall temperature for HC oxidation, and the latter decreases the amount of flame-quenched HCs. The increase in NO$_x$ is mostly seen in the slow burning engines that realize substantial gains in peak cylinder pressure/temperature with faster combustion. For lean-burn operation, it has been shown that a trade-off exists between HC and NO$_x$; increasingly lean mixtures reduce the combustion temperature and NO$_x$ emissions, but this simultaneously leads to progressively incomplete combustion and HC emissions. The addition of H$_2$ to the fuel has been shown to reduce the severity of this trade-off [95, 138].

Despite several advantages to using hydrogen supplementation in NG engines, there are evidently some challenges that remain to be solved. First, there seems to be no clear indication as to the optimal ratio between H$_2$ and NG, which will likely also depend on the composition of NG available. Second, the gains in efficiency and emissions for a fast-burn engine are modest, and thus the return on investment is questionable. And third,
the hydrogen needs to be sourced or produced by some means, which is sure to generate additional costs that are beyond the scope of this assessment.

2.3.3 Direct Injection

Recent improvements in gaseous fuel injector technology have made direct injection (DI) of natural gas more feasible. By injecting the fuel directly into the cylinder during the compression stroke, the intake air does not get displaced and the volumetric efficiency lost to port-injection can be regained. However, various studies have shown that NG injected directly into an engine cylinder does not readily mix with the air [15, 47, 63], which makes it challenging to achieve mixture homogeneity. An experimental investigation by Zeng et al. [150] assessed the combustion and emissions characteristics of a DI NG engine under various injection timings. They discovered that injecting too early (while the intake valve remained open) was detrimental to volumetric efficiency. Conversely, injecting too late, and the air/fuel mixing was inadequate, resulting in long burn durations and high HC emissions.

Direct injection also offers the benefits of utilizing stratified charge combustion, which uses spray techniques or wall-guided flows to generate a locally rich mixture in the vicinity of the spark plug. This enables ignition of a globally lean mixture that would otherwise be incombustible. Generally, two injections are required: one to create a lean homogeneous mixture during the compression stroke, followed by a second injection around TDC to enable ignition. This method of combustion is often referred to as gas-jet ignition. Shiga et al. [128] were able to obtain ultra-lean combustion in a rapid-compression-machine with an overall equivalence ratio of $\phi = 0.02$ by utilizing the fuel injection-generated turbulence to improve the overall reaction rate. The combustion efficiency at this lean limit was very poor, however, due to bulk quenching of the flame. Ali et al. [6] implemented the same method in an engine, and were able to extend the lean operating limit down to $\phi = 0.35$. Timing of the first injection pulse (i.e. the pre-mixed charge) was found to strongly correlate with cyclic variation, implying homogeneity issues.

It is obvious that direct injection of natural gas imposes significant challenges associated with air/fuel mixing. Provided adequate homogeneity can be accomplished, DI of NG is a promising technology for improving engine performance. However, the low lubricity of the fuel, combined with no evaporative cooling during injection, is likely to affect injector longevity. The author expects that durability will be one the greatest obstacles in creating a production DI NG fuel injector.
2.3.4 Pre-Chamber Gas Jet Ignition

A variation of the gas-jet ignition method, known as *turbulent jet ignition*, has recently been introduced in the literature [11, 20, 135]. In this system, fuel is injected into a small pre-chamber where it is ignited by means of a spark plug or glow plug. The reacting mixture then emerges from the pre-chamber into the main chamber through a circular array of orifices that connect the two chambers together. The homogeneous mixture in the main chamber is bulk-ignited by these turbulent reacting jets, resulting in a very rapid burn. A rendering of a possible turbulent jet ignition pre-chamber configuration is shown in Figure 2.7. One disadvantage to this method is that it requires the use of two fuel injectors, one for each chamber, which adds to the cost and complexity of the fuel system.

![Figure 2.7: Rendering of turbulent jet ignition concept, adapted from [135].](image)

Experimental studies by Toulson et al. [135], and Attard and Blaxhill [11] have shown that the lean limit of gasoline can be extended from $\phi \approx 0.65$ to $\phi < 0.5$, with near zero NO$_x$ emissions, using the turbulent jet ignition method. However, additional research remains to be done regarding high load operation ($\phi = 1$), and the applicability to NG engines.

2.3.5 Summary

It is quite evident that the methods being explored, in the search for improved characteristics of performance and emissions from NG engines, are significantly different. However, all of these approaches essentially focus on two things: manipulation of the fuel properties, through H$_2$ addition or mixture stratification, and increasing the in-cylinder turbulence intensity. The latter has been shown in this section to increase the burn rate, reduce emissions, and extend the flammability limits under dilute conditions. It is for these reasons
that combustion turbulence is the fundamental basis of this research. The relevant theory relating to the interaction between combustion and turbulence will now be discussed.

2.4 Turbulence and Combustion

Turbulent fluid motion is random in nature and thus difficult to precisely define. It does, however, possess certain characteristics that provide insight into the physical attributes and interactions of the flow. First, turbulence is a property of the flow and not the fluid. It is disordered, chaotic, and involves extensive fluid mixing. Its origin stems from flow instabilities and the turbulent flow field is characterized by three-dimensional fluctuating vorticity, distributed continuously but irregularly. Turbulent flows operate in a dissipative, cascading manner, where the kinetic energy of large scale motion is continuously broken down by viscous effects into smaller vortical motions and eventually consumed as a rise of the fluid’s internal energy. In other words, turbulent flow cannot be sustained without a continual abstraction of energy from the mean flow. This is of particular relevance to engines where the bulk flow is intermittent and the desired effects of turbulence are required at the end of the compression stroke, during a period of quasi-static flow.

In-cylinder fluid motion strongly affects the performance and emission characteristics of an internal combustion engine. Were it not for turbulent fluid motion, mixing of the air/fuel charge would be poor and the combustion process would be slow and inefficient. It is well documented that increased levels of turbulence in engines enhance the rate at which the reactants are consumed during combustion [39, 70]. High intensity turbulence also effectively reduces cycle-to-cycle variations by enhancing the early flame development period, which reduces the opportunity for flame kernel quenching caused by large scale, bulk fluid motion [74, 119]. The purpose of this section is to provide the reader with some basic information regarding the interaction between combustion and turbulence in engines. The conventional methods used to generate in-cylinder turbulence will also be discussed.

2.4.1 Turbulent Scales

A turbulent flow field is characterized by a spectrum of eddies: tubular, vortical structures of varying size that reside within the mean flow. The classification of turbulent eddies is based on a length and a velocity scale, which is represented by the diameter and angular velocity of the eddies, respectively. Alternatively, and more pertinent in the context of combustion, the velocity scale may be substituted by a time scale that represents the turnover time
of an eddy. On the large end of the size spectrum, a characteristic dimension known as the integral length scale, $l_o$, can be defined. The integral scale represents larger eddies that possess the highest amounts of kinetic energy. Their size is generally dictated by the size of the flow field confinement. For example, in engines the integral length scale has been deduced from velocity measurements and found to be on the order of one-sixth of the clearance height at TDC [56, 94]. At the other end of the spectrum, the smallest eddies are classified by the Kolmogorov scale, $l_k$, beyond which smaller eddies cannot exist as they are damped out by viscous dissipation. The dissipative nature of turbulence means that the kinetic energy of the eddies in the integral scale is continuously passed down to the Kolmogorov-sized eddies at a constant rate of dissipation, $\epsilon$. Dimensional analysis leads to following relations for Kolmogorov time $t_k$, length $l_k$, and velocity $u_k$; where $\nu$ is the kinematic viscosity of the fluid [91]:

$$t_k \approx \left( \frac{\nu}{\epsilon} \right)^{1/2} \quad l_k \approx \left( \frac{\nu^3}{\epsilon} \right)^{1/4} \quad u_k \approx (\nu \epsilon)^{1/4}$$ (2.13)

From the equations shown in (2.13), it is apparent that a larger rate of cascading energy will produce smaller and faster (more intense) eddies.

Again through dimensional analysis, an expression can be derived for the rate of energy transfer in terms of the turbulent kinetic energy, $k_e$, and the integral length scale, $l_o$. The equation is further refined by identifying the characteristic velocity fluctuations to be associated with integral scale, such that $k_e \approx 3/2(u'_o)^2$:

$$\epsilon \approx \frac{k_e^{3/2}}{l_o} \approx \frac{u'^3}{l_o}$$ (2.14)

Equation (2.14) can then be combined with the equation for $l_k$ from (2.13) to yield:

$$\frac{l_o}{l_k} \approx \left( \frac{u'_o l_o}{\nu} \right)^{3/4} = Re_T^{3/4}$$ (2.15)

where $Re_T$ is the turbulent Reynolds number based on the integral scales. From Equation (2.15) it can be seen that a large Reynolds flow increases the size gap between the largest and smallest eddies. Since the integral size is essentially fixed, constrained by a characteristic dimension of the system, the higher turbulence intensity associated with large $Re$ flows will therefore act to reduce the size of the Kolmogorov eddies. This is of significant importance in relation to combustion due to the physico-chemical effects turbulence has on a reacting mixture, specifically in terms of the size of the eddies in relation to the chemically reacting volume.
### 2.4.2 Interaction Between Combustion and Turbulence

Inside the combustion chamber of a spark ignition engine, the compressed, premixed air/fuel charge is ignited by a high voltage arc generated by the spark plug. The localized high temperature of the spark initiates a self-sustaining, exothermic combustion reaction in the form of a thin, turbulent flame [70]. The flame propagates through the air/fuel mixture in a spherical manner until it is eventually quenched at the combustion chamber walls. During this process, the interaction between the flame and its surroundings has a profound effect on the rate at which combustion progresses.

For a laminar flame, the velocity at which the flame front advances normal and relative to the unburned gas is known as the laminar burning velocity, $u_L$. This is a fundamental property of the unburned gas mixture and should not be confused with the laminar flame speed, $S_L$, which is the rate of flame propagation relative to a fixed frame of reference. The former is proportional to temperature, inversely proportional to pressure, and generally highest for a slightly rich air/fuel ratio [117]. For spherically burning flame with a large radius (neglecting flame thickness and stretch) the relationship between the laminar flame speed and laminar burning velocity is given by [61]:

$$S_L = \frac{\rho_u}{\rho_b} u_L$$  \hspace{1cm} (2.16)

where $\rho_b$ and $\rho_u$ are the burned and unburned gas densities, respectively. This ratio of densities accounts for the fact that the combustion products are at a significantly higher temperature than the reactants and the difference in density is pushing the flame front outward. Take for example an unburned gas temperature and pressure of 600 K and 20 bar, respectively. If the burned gas temperature is $\sim 2000$ K, the density ratio is approximately 4:1. The laminar burning velocity can be estimated using the empirical relationship derived by Metghalchi and Keck [101]:

$$u_L = u_{L,0} \left( \frac{T_u}{T_0} \right) ^{\alpha} \left( \frac{p}{p_0} \right) ^{\beta}$$  \hspace{1cm} (2.17)

where $\alpha = 2.18 - 0.8(\phi - 1)$, $\beta = -0.16 + 0.22(\phi - 1)$, $\phi$ is the air/fuel equivalence ratio, and $u_{L,0}$ is the laminar burning velocity at reference temperature, $T_0$, and pressure, $p_0$. A value of $u_{L,0} \approx 0.35$ m/s for methane at 1 atm and 298 K is given by Law [91]. It should be noted that Equation 2.17 was based on experiments using methanol, iso-octane, and indolene, but is adequate for the purpose of this example. For the conditions of $T = 600$ K and $p = 20$ bar, Equation (2.17) yields $u_L \approx 1$ m/s; therefore, $S_L = 4$ m/s. For an engine bore with a radius of 50 mm, a laminar flame travelling at 4 m/s would require 12.5 ms to
reach the bore wall from centre. At an engine speed of 3000 RPM this would constitute a crank angle duration longer than the expansion stroke, which is obviously impractical. It is for this reason why turbulence is essential for combustion in engines.

Turbulence effectively increases the flame propagation speed up to 50 times the laminar value [117]. It does so by significantly wrinkling and distorting the laminar flame, increasing the overall surface area of the flame sheet. At the same time, turbulence increases the rates of molecular mixing, which accelerates the conduction of heat and the diffusion of radicals out from the reaction zone. As a result, the mass burning rate is accelerated, allowing combustion of a fixed mass to occur in a shorter amount of time. The turbulent burning velocity, \( u_T \), can be used to describe the rate of reactant consumption, analogous to \( u_L \).

Ting et al. [134] show the increase in turbulent burning velocity to be proportional to the turbulent velocity fluctuations, \( u' \), under relatively small levels of normalized turbulence \( (u'/u_L < 8) \). For higher levels of \( u' \), studies by Abdel-Gayed et al. [2, 3] and Kobayashi et al. [85] show a non-linear decay in turbulent burning velocity with increasing \( u' \). In other words, there appears to be diminishing returns for increased turbulence intensity, although the literary data reviewed in [3] shows considerable scatter. In any case, it is generally accepted that the turbulent burning velocity can be expressed in the form:

\[
\frac{u_T}{u_L} = 1 + C \left( \frac{u'}{u_L} \right)^n
\]  

(2.18)

where \( n \) are \( C \) are constants fitted from experimental data. It is expected that \( C \) will depend on the relative size between the turbulent eddies and the flame thickness \( (l_o/\delta_L) \) [114]. Therefore, it can be said that the turbulent burning velocity, and thus mass burning rate, is dependent on both the intensity and size of the turbulence. The latter will now be discussed.

**The Importance of Length Scale**

The initial flame kernel that develops from the spark plug is relatively small \( (R < 1 \text{ mm}) \), approximately spherical in shape, and most likely laminar [71]. A gradual transition to a turbulent flame occurs as the local flow begins to distort the thin reaction layer of the flame kernel. This initial mass burning period, generally taken arbitrarily as 1, 5, or 10% of the total mass, is referred to as the flame development period. Although seemingly inconsequential, this flame development period can account for up to 50% of the overall 0-90% mass burn duration [70].
Because the flame kernel is initially quite small relative to the smallest turbulent eddies that are likely to be present in the combustion chamber, the turbulence acts to convect the flame kernel rather than wrinkle it. Images of relatively unperturbed flame kernels being displaced from the spark plug electrodes have been shown in combustion chamber images, such as those given by Tabaczynski [132] and Heywood [71]. The importance of eddy size on flame kernel development was also noted by Keck et al. [81] who reported that the transition from early flame development into rapid combustion did not occur until the flame radius was of the same order as the integral length scale. This notion was further expanded by Ting et al. [134] who showed that for a given flame size and turbulence intensity, the smaller scale turbulence is significantly more effective at enhancing the turbulent burning velocity. Therefore, it is clearly advantageous to have small scale turbulence during the initial flame development period. However, smaller scale turbulence also decays faster (see Equation (2.14)) and therefore may not be sustainable for the entirety of combustion. Studies by Johannson [73] and Shiyong et al. [129] looked at the correlations between mass burning rate and the measured turbulent frequency spectrum. Both studies indicate the most beneficial turbulent frequencies for flame development are in the approximate range of $4$–$7$ kHz, which were roughly calculated by the respective authors to be between $0.5$ mm and $1.5$ mm. Thus, if the integral length scale is around $1/6$ of the clearance height, $h_c$, at TDC, one could expect values of $h_c$ on the order of $3$–$9$ mm to minimize the flame development time. Interestingly, a numerical simulation performed by Reddy and Abraham [119] showed that for an integral length scale that is smaller than the flame kernel diameter, wrinkling of the flame kernel does occur. Additionally, provided the turbulence intensity is sufficiently high (i.e. $u'/u_L > \sim 5$), the flame kernel can become stretched and torn. At a certain point with increasing intensity, their simulation revealed the flame kernel was quenched before it could develop into a propagating flame. Thus, it appears decreasing the size and increasing the intensity of turbulence is beneficial, but a practical limit is very likely to exist.

As the flame kernel continues to grow in size, the spectrum of eddy sizes that are able to wrinkle its surface progressively increases. Eventually, the flame radius is sufficiently large that the entire range of length scales are influential. The larger scales continue to wrinkle and corrugate the flame, while the smaller scales act to broaden the reaction zone through enhanced molecular transport.

It is important to note that up to this point there has been an implicit assumption that the construction of a turbulent flame is simply that of a heavily wrinkled laminar flame. In reality, there are several regimes of turbulent flames that depend on the size and velocity of the turbulent eddies. For now, it is sufficient that the reader understand that both size
and intensity are important factors pertaining to turbulence in engines. A discussion of turbulent flame regimes will be given in Chapter 8.

2.4.3 Generating In-Cylinder Turbulence

Typically, in-cylinder turbulence is generated by one of two different methods:

1. Organized flow through the intake port that is in the form of axial swirl or transverse tumble. This bulk fluid motion eventually breaks down into smaller scale turbulence throughout the compression stroke.

2. Having small gaps between the cylinder head and select regions of the piston face, known as squish volumes. The compressed gas is displaced laterally from these small gaps as the piston approaches TDC and impinges into the main combustion chamber.

The first method relies on specific intake port designs that guide the incoming charge into an organized flow pattern within the cylinder. Helical and tangential intake ports, or staggered intake valve timing (for multi-valve engines), are used to produce a swirling flow. While inclined intake ports are generally used to produce a tumbling flow pattern. Regardless of the design, the purpose is to prolong the dissipation of the flow’s kinetic energy by conserving its angular momentum so that it breaks down into small scale turbulence closer to the end of the compression stroke. Numerous empirical studies have been performed regarding the measurement of fluid velocity within the combustion chamber of an engine. A few examples include the works of Li et al. [92] and Kang and Beck [77] for tumble flow, as well as Hall and Bracco [67] and Witze et al. [147] for swirl flow designs. Despite the high initial velocities that can be generated by these port configurations (e.g. up to 30 m/s [67]), all of the aforementioned studies showed considerable turbulence decay by the end of the compression stroke. In fact, for all cases, the turbulent velocity fluctuations ranged from 0.5–3 m/s at TDC. Hall [67] and Kang [77] show these values to be around one-half the mean piston speed or less. No mention of engine geometry or piston speed was provided in the other works.

It is expected that the higher flow velocities required for tumble and swirl ports will incur some penalty in flow friction. Squish areas are advantageous in this respect since they do not rely on intake flow for turbulence generation. However, smaller valves may be required in order to physically produce the squish areas, which is generally counter-productive in terms of volumetric efficiency. In any case, measured velocity fluctuations for bowl-in-piston (BIP) combustion chambers with large squish areas (up to 74% of the bore) have been
presented by Tunestål et al. [138] and Johansson and Olsson [75]. The highest turbulent velocity fluctuations were again found to be on the order of 3 m/s for the largest fractional squish areas. Furthermore, the peak turbulence is shown to occur approximately 10°CA after TDC, which was well into the main burning phase of combustion. This is an inherent problem with generating turbulence using squish volumes.

2.4.4 Summary

The rate of combustion in engines is largely dictated by the level of turbulence present within the combustion chamber. Both the size and the intensity (velocity) of the turbulence is important, especially when the flame radius is small. High turbulent velocity fluctuations are desirable in order to heavily wrinkle the flame front and maximize the turbulent burning velocity. At the same time, small length scales, which are a function of the combustion chamber geometry and turbulence intensity, are also important for accelerating the early flame development period. The traditional methods of turbulence generation are seemingly incapable of producing velocity fluctuations in excess of one-half the mean piston speed, and the mechanisms of generation lack proper temporal alignment with the ignition timing.
Chapter 3

Split-Cycle Engine Design

3.1 Engine Overview

The split-cycle engine developed in this work is based on an extensively modified Kubota Z482 indirect injection (IDI) diesel engine. The Kubota engine is liquid cooled and has a vertical, in-line, twin cylinder layout. Each cylinder displaces approximately 241 cm$^3$, and has a square$^1$ bore-to-stroke configuration. The close proximity of the two cylinders, the ruggedness of the overall design (e.g. forged crankshaft, integrated main bearings, etc.), and the low cylinder clearance volume at TDC made the Kubota engine an excellent candidate for modifying into a split-cycle engine.

The front cylinder of the engine was designated to perform the intake and compression strokes, while the rear cylinder provides expansion and exhaust. Methane fuel is introduced by means of electronic fuel injection into an intermediary volume in the cylinder head, which connects the two cylinders together. The air/fuel mixture is then ignited within a pancake style combustion chamber by a single, centrally mounted spark plug.

Modifications to the crankshaft, connecting rods, pistons, valvetrain, timing drive system, and many peripheral components were necessary to accommodate the unique physical and operational characteristics of a split-cycle engine. In addition, the author has designed and developed a new two-piece cylinder head to facilitate fluid transfer from one cylinder to the other. Part of the existing Kubota pushrod valvetrain was retained for actuation of the intake and exhaust valves, although extensive modification of the cam and rocker shafts was required. An entirely separate camtrain, utilizing an overhead camshaft (OHC) and reverse

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$^1$ Bore and stroke are approximately equal: Bore = 67 mm, Stroke = 68 mm
poppet valves (RPVs), was also developed and integrated into the new cylinder head. This additional valvetrain governs fluid transfer in and out of the crossover passage, and the fundamental operating nature of the engine requires these valves to have an uncharacteristically short duration (~35°CA), resulting in large accelerative forces. To help generate a cam profile with acceptable kinematics, MATLAB® was used to develop a profile optimization tool, which will be discussed briefly in Section 3.5.3. A multi-body dynamic simulation of the OHC valvetrain was performed as part of the design process (see Section 3.5.5), and predicted a safe operating speed for the camshaft to be approximately 1500 RPM. Since the split-cycle camtrain has a 1:1 ratio with the crankshaft, the maximum engine speed used in this work was limited by the valvetrain design.

Due to the limited information available regarding split-cycle engine design, the author elected to incorporate as much versatility as practicable; such as, mechanically variable piston and cam phasing, and an adjustable and modular valve spring preload mechanism for the RPVs. The timing drive was also changed from an internal gear type to an external belt drive for easier access and quicker adjustment. Not all of the adjustability aspects were utilized in this work, but implemented with future research in mind (e.g. piston phasing).

An overview of the split-cycle engine specifications are given in Table 3.1. Note the abnormally high geometric compression and expansion ratios, which are calculated values based on engine geometry. Because the reverse poppet valves are open near TDC and fluid transfer is occurring, the actual compression and expansion of the gases is considerably less. The difference between the two ratios is from small recesses machined into the expansion cylinder piston (~560 mm$^3$) to accommodate the exhaust valve and spark plug at TDC.

<table>
<thead>
<tr>
<th>Table 3.1: Split-Cycle engine specifications.</th>
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<tbody>
<tr>
<td>Bore</td>
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<td>Stroke</td>
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<tr>
<td>Rod Length</td>
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<tr>
<td>Cylinder Displacement</td>
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<tr>
<td>Geometric Compression Ratio</td>
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<td>Geometric Expansion Ratio</td>
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<tr>
<td>Piston Offset Angle</td>
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<tr>
<td>Intake / Exhaust Valve OD$^1$</td>
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<tr>
<td>Crossover Valve OD$^2$</td>
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<tr>
<td>Spark plug</td>
</tr>
</tbody>
</table>

$^1$ Conventional poppet valve, inward opening.

$^2$ Reverse poppet valve (RPV), outward opening.
Figure 3.1 shows a CAD rendering of the split-cycle engine designed in this work and modelled using CATIA software. The entire engine assembly measures approximately 35 cm wide by 33 cm long by 60 cm tall. New (previously non-existent) part drawings can be found in Appendix A, along with a select number of drawings pertaining to extensively modified components.

Figure 3.1: CAD rendering of split-cycle engine.
3.2 Cranktrain

The cranktrain components were entirely sourced from the Kubota engine and then modified
to suit the requirements of the split-cycle engine. This involved re-orientating the journals
of the crankshaft, followed by balancing of the rotating assembly. Small alterations to the
pistons and the connecting rods were also required for clearance purposes.

3.2.1 Crankshaft Modification

Offset piston phasing in a split-cycle engine is desirable so that when a portion of fluid is
released from the crossover passage into the expansion cylinder, it is simultaneously replaced
by the compression cylinder. In this manner, the crossover passage remains at elevated
pressure, and re-compression by the expansion cylinder is not necessary. At the time when
the engine was being designed, no literature regarding the effects of piston phasing in a
split-cycle engine could be found, and the author did not wish to rely on the 1-D engine
model (see Chapter 7) before it was validated experimentally. Therefore, the initial piston
offset phasing was selected based on the work of Phillips et al. [115], who reported successful
split-cycle engine trials using an offset of 20°CA. A graphical depiction of the offset is given
in Figure 3.2, where the crankshaft rotates in a clock-wise manner when viewed from the
front of the engine, resulting in the expansion cylinder leading the compression cylinder.

1 = Compression
2 = Expansion

Figure 3.2: Depiction of crankshaft journal offset.

The high cost of manufacturing a crankshaft, combined with the uncertainty regarding ideal
piston phasing, motivated the author to implement a non-permanent solution for offsetting
the crankshaft journals. At a plane perpendicular to the rotational axis and bisecting the
middle bearing journal, the Kubota crankshaft was cut in two pieces and then rejoined using a three-sided polygon pin, as shown in Figure 3.3.

![Polygon Pin](image)

**Figure 3.3:** Exploded-view CAD rendering of split-cycle crankshaft.

The use of the polygon shape was based on three necessary requirements:

(i) The offset between the crankshaft journals must be incapable of angular change during engine operation.

(ii) The crankshaft must retain its original dimensions, and all main bearing journals must remain concentric and cylindrical when re-assembled.

(iii) Assembly and accurate indexing of the crankshaft should be simple.

The polygon shape achieves all of these criteria: the triangular geometry binds under attempted rotation, prohibiting any relative motion between the two crankshaft halves; the use of a pinned joint allowed for retention of the original journals and main bearings; and the geometric shape of the pin provides easy and accurate phasing of the two connecting rod journals. Adjustment of the piston phasing requires offset-machining one half of a new polygon pin to the desired orientation. The use of a splined joint was also considered, as it meets the above criteria and would not require a new pin for each adjustment, but the cost of machining a spline with a 5°CA increment resolution was prohibitive.

A finite element analysis (FEA) model was created in CATIA to verify the strength of the modified crankshaft. The torsional yield strength of the 40 mm main bearing journal is capable of transmitting over 5000 Nm of torque; approximately 200 times the rated value of the Kubota engine. Thus, the FEA model was intended more for the press-fit stresses than the torsional loads, although an axial torque of 250 Nm was still applied for good measure. The resulting pin geometry can be found in Appendix A.
The polygon pin was machined from Aermet310®, a steel alloy with a tensile yield strength of 1875 MPa. While considerably stronger than necessary, the alloy was primarily chosen for its higher hardness (HRC 38) and fracture toughness (71 MPa$\sqrt{m}$) over a 4340 steel alloy—a common crankshaft material. Both the pin bore and the pin were machined by Carlyn Manufacturing in Windsor, ON, using a die-sink electrical discharge machining (EDM) process. The accuracy of this method was required to maintain the journal bearing alignment, and an accurate press-fit of approximately 0.012 mm. The piston timing in the assembled engine was verified to be within $\pm 0.5^\circ$CA of the nominal 20°CA offset, although the measurement was limited by the accuracy of the instruments used, and the author expects the actual error is somewhat less.

**Balance of Rotating Assembly**

The Kubota crankshaft uses an in-line cylinder arrangement, with the crankshaft journals offset by 180°. Counterweights positioned axially on the far side of, and radially opposite to, the connecting rod journals produce a coupling moment to oppose the one created by the journals. In other words, this configuration yields no end-over-end crankshaft moment and the primary inertia forces are balanced. The secondary forces, a result of the sinusoidal connecting rod motion, still exist.

Re-orientation of the connecting rod journals for the split-cycle engine resulted in both connecting rod journals being located on the same side of the crankshaft, offset by 20°. In this configuration, the primary forces are no longer cancelling in the vertical plane. Thus, balancing of the cranktrain is fundamentally similar to that of a single cylinder engine, where the mass of the counterweights cannot fully balance the reciprocating forces of the pistons, resulting in a lateral unbalance when the crank is at mid-stroke.

The inertial forces generated, in the axial direction of the cylinder, by the piston and a portion of the connecting rod can be evaluated for a single cylinder using Equation (3.1).

$$F_v = MR\omega^2 \left( \cos \theta + \frac{R}{l} \cos(2\theta) \right)$$  \hspace{1cm} (3.1)

Where $M$ is the lumped reciprocating mass, $R$ is the throw of the crankshaft, $\omega$ is the angular velocity of the crankshaft, $l$ is the connecting rod length, and $\theta$ is the angular position of the crankshaft journal.

The right hand side of Equation (3.1) are the first two terms of a Fourier series that represent the first and second harmonics of the system. The first term is known as the *primary*
unbalance, since it has a frequency proportional to the crankshaft speed. The second term or secondary unbalance oscillates with a frequency that is two times that of the crankshaft speed. Higher-order modes were neglected due to their relatively small contribution to engine vibrations.

Multiple cylinders can be evaluated by using the expression for $\theta = \theta_1 + \phi$, where $\theta_1$ is the crank angle of any selected reference connecting rod journal starting from 0°CA at TDC, and $\phi$ is the relative angle between the reference journal and that of cylinder $i$. Using the trigonometric identity for $\cos(\theta + \phi) = \cos(\theta) \cos(\phi) - \sin(\theta) \sin(\phi)$, and noting that $\phi$ is a constant, Equation (3.1) can be re-written for multiple cylinders as

$$F_v = M \omega^2 \left( R \left[ \cos(\theta_1) \sum \cos(\phi) - \sin(\theta_1) \sin(\phi) \right] + \frac{R^2}{l} \left[ \cos(2\theta_1) \sum \cos(2\phi) - \sin(2\theta_1) \sin(2\phi) \right] \right) \quad (3.2)$$

By including the forces of the crankshaft counter-balance masses ($F_b = M_b R_b \omega^2$), Equation (3.2) provides the primary and secondary unbalanced forces in the vertical direction of the rotating assembly.

Similarly, the transverse forces can be calculated from the first term of Equation (3.2), except the sine of the crank angle is used. The second term is dropped since the secondary inertia forces stem from the reciprocating mass only.

$$F_t = M R \omega^2 \left[ \sin \theta_1 \sum \cos \phi + \cos \theta_1 \sum \sin \phi \right] \quad (3.3)$$

A balance factor can be defined as the proportion of counter-balance mass in relation to the opposing reciprocating mass. Equations (3.2) and (3.3) were evaluated for a range of balance factors from zero to one. The minimum resultant magnitude was found to be with a balance factor of 0.62. Therefore, the crankshaft was balanced using approximately 60% of its reciprocating mass. The resulting out-of-balance forces calculated for an engine speed of 1500 RPM are shown in Figure 3.4.

A neutral or zero balance was not achieved in the rotating assembly of this engine. This is evident by examination of Figure 3.5, which shows the resultant force unbalance plotted on polar coordinates for an engine speed of 1500 RPM and a balance factor of 0.62. Complete force cancellation is nearly realized at 170°CA, when the primary forces are mostly offset by secondary inertia forces. However, the relatively low magnitude of the unbalanced forces throughout the remainder of the cycle, versus the complexity and cost of adding counter-balance shafts, was a decidedly acceptable trade-off for this application.
Chapter 3: Split-Cycle Engine Design

Figure 3.4: Primary and secondary unbalanced forces of rotating and reciprocating mass in the split-cycle engine.

Figure 3.5: Magnitude of unbalanced force (in Newtons) of the split-cycle cranktrain.
To physically obtain the desired balance factor, the connecting rod journals were drilled hollow and excess material surrounding each journal was removed to reduce the journal-side mass. Opposite from the journals, the counterweight masses were increased using Tungsten slugs ($\rho = 19.25 \text{ g/cm}^3$), pressed into holes machined in the counterweights. The crankshaft was then dynamically balanced (verified to 5000 RPM) using simulated piston/connecting rod masses. Balancing was performed by Mountain Machine in Tecumseh, Ontario. Figure 3.6 shows the stock Kubota crankshaft versus the rotated and modified version for the split-cycle engine. Note that the photograph of the split-cycle crankshaft was taken before the tungsten slugs were added to the counterweights.

![Kubota Z482](image1)

![Split-Cycle](image2)

**Figure 3.6:** Original Kubota crankshaft (*top*) compared to the modified split-cycle version (*bottom*). Note the modified unit is shown before additional mass was added to the counterweights.

### 3.2.2 Piston and Connecting Rod

The Kubota connecting rods were reused without any significant modifications. Minor removal of material from the piston-end bearing journal boss, as shown in Figure 3.7, was required for clearance around the additional crankshaft counterweight when the piston is at bottom dead centre (BDC).
Indirect injection (IDI) engines, such as the Kubota Z482, use a small chamber located in the cylinder head to initiate combustion and therefore have a very minimal cylinder clearance volume at TDC. The Kubota pistons actually protrude above the block deck at TDC, leaving approximately 0.50 mm piston-to-cylinder head clearance when the head gasket is installed. This was ideal for the split-cycle engine in order to minimize the residual gas left in each cylinder at TDC; as such, no change to the piston crown height was made.

The original piston recesses were filled in, by means of welding, to further minimize the clearance volume of each cylinder. During the welding process, the cast aluminum pistons were partially submerged in water to prevent distortion. Afterwards, the crown of the compression cylinder piston was machined flat, and minor recesses were machined into the crown of the expansion cylinder piston to provide clearance for the exhaust valve and the spark plug. A schematic of the expansion cylinder piston is shown in Figure 3.8. The small bowl shape allows the spark plug electrode to protrude slightly from the cylinder head surface, preventing shrouding and possible flame kernel quenching. The use of reverse poppet valves (RPVs) avoids the requirement of additional piston recesses.

![Piston-pin journal of connecting rod; original (left), modified (right)](image)

**Figure 3.7:** Piston-pin journal of connecting rod; original (left), modified (right).

![Schematic of piston-top profile for the expansion cylinder.](image)

**Figure 3.8:** Schematic of piston-top profile for the expansion cylinder.
3.3 Engine Block

A brand new original equipment (OE) Kubota Z482 engine block was used without any modification. The integrated mechanical fuel injection pump was not required and was removed to reduce parasitic losses. Also, the oil line running to the cylinder head was blocked since a separate system was implemented for the new cylinder head (see Section 4.2.2).

3.4 Cylinder Head

To enable fluid transfer from one cylinder to the other, along with the implementation of RPVs, it was decidedly easier to manufacture a new cylinder head versus modifying the existing Kubota one. Several prototypes were created, eventually leading to a two-piece, billet-machined cylinder head.

3.4.1 Design Methodology

Ideally, the design of a cylinder head would begin with general packaging constraints (e.g. bore size, number of cylinders etc.) and proceed by positioning critical components in the combustion chamber such as the spark plug and valves. The general shape of the combustion chamber is often dictated by the number and size of the valves required for a specific engine application and/or performance target. The design can then proceed in an iterative manner with the lower-level components (e.g. bolt holes, cooling passages etc.) until an optimal design is achieved. The design hierarchy used in this work, however, was burdened by the use of an existing engine block. Head bolt hole locations took immediate precedence over some of the more important components and the entire design was constrained by the retention of these bolt holes. Furthermore, the decision was made to retain the intake and exhaust valve locations in order to reuse the major valvetrain components (e.g. camshaft, pushrods, rocker arms) from the Kubota engine, thereby minimizing costs. The author also opted for a single intake/exhaust valve per cylinder in order to simplify the design.

Based on the aforementioned constraints, packaging priority was allotted to the intake, crossover and exhaust port geometries, as well as the placement of the spark plug, pressure transducers, and fuel injector. Cooling and oil passages were given lowest priority and implemented last. Design revisions were made until packaging and assembly were mutually feasible, and all components were practicably manufacturable, particularly the cylinder head, which was machined from a solid billet to avoid the cost and complexity of casting.
3.4.2 Port, Valve, and Chamber Configuration

A disk or pancake style combustion chamber was essentially dictated by the decision to maintain a portion of the original Kubota valvetrain components and their subsequent geometry. However, as it was discussed in Section 2.4.3, the split-cycle engine does not require squish areas to generate turbulence. Therefore, there was no reason to employ a bowl-in-piston or bathtub style combustion chamber, which could potentially incur higher HC emissions. The commonly-used pent-roof combustion chamber was also unnecessary for the split-cycle engine, since the high gas pressure ratios do not require the large valve diameters for increased flow that this design is intended to achieve.

The split-cycle engine was designed to use both inward-opening conventional poppet valves for the intake and exhaust ports, as well as outward-opening reverse poppet valves (RPVs) for the inlet and outlet to the crossover passage. The RPVs were used to accommodate opening of the crossover valves in the vicinity of TDC without necessitating a clearance volume in each piston crown, thereby minimizing the residual gas trapped in each cylinder. It was anticipated that this would help maintain an acceptable volumetric efficiency.

The locations of the intake and exhaust valves were largely predetermined by the Kubota valvetrain geometry. However, the ports were relocated from the sides of the cylinder head to opposing ends. This was done to minimize heat transfer from the expansion cylinder to the intake charge, and to create a more streamlined flow path for each port. The valve and port configuration is shown in Figure 3.9; the intake port faces the front of the engine.

The shallow entry angle of the Kubota intake port is designed to generate turbulence through a swirling motion (see Section 2.4.3). Because the split-cycle engine does not use the intake/compression cylinder for combustion, turbulence generation is not required and may also be detrimental to the volumetric efficiency. To better streamline the path of air entering the engine, the intake port was raised vertically by extending the inlet partially into the upper cylinder head (discussed in Section 3.4.3), as shown in Figure 3.10. This also enabled the intake port to be machined in two relatively simple setups on a 3-axis computer numerically controlled (CNC) mill.

Each valve translates vertically through a bronze valve guide and a corresponding Viton metal-reinforced seal attached to the top of the guide. A valve stem-to-guide diametrical clearance of 0.025–0.050 mm was used for all valves. The guides and seals were purchased from Ferrea Racing Components in Ft. Lauderdale, FL. A complete list of part numbers can be found in Appendix B.
Figure 3.9: Split-cycle valve and port configuration.

Figure 3.10: Cross-sectional view of engine intake port: imitation of Kubota intake port for one-piece head design (left), extended port using two-piece head design (right).
3.4.3 Construction and Materials

Three prototype cylinder heads were manufactured (to varying stages of completeness) during the development of the final product used to conduct the experimental trials presented in this dissertation. All of these prototypes were a one-part design manufactured from a billet of 6061-T6 aluminum and had the cooling passages machined from the outer walls of the cylinder head. Two important difficulties were discovered from these original prototypes:

1. Adequate cooling passages are difficult to machine from exterior surfaces due to limited straight-line tooling access.

2. Providing a hardened valve seat for a RPV in an aluminum cylinder head is no simple task. Typical press-fit seats are likely to be pushed out by the valve, and threaded seat inserts are bulky and costly to machine. A modular crossover passage with integral valve seats was also attempted, but proved to be difficult to adequately manufacture.

It was primarily these two factors that lead to the design of a two-part cylinder head, shown in Figure 3.11. The two-part design allowed for the crossover passage and cooling jackets to be machined from the top side of the head, enabling integral valve seats to be used. Furthermore, intricate cooling passages were able to be machined in closer proximity to temperature-sensitive components, like the spark plug and pressure sensors. The upper cylinder head acts to seal these passages when the engine is assembled, and it also supports the RPV camtrain. The two parts of the cylinder head are clamped together by the head bolts, which run from the upper head, pass down through the lower head, and thread into the block. A 1 mm thick, grade 110 copper sheet was used to fabricate a gasket between the two parts of the cylinder head. Hyolmar® universal blue, non-setting joint compound was applied to both sides of the gasket to ensure sealing and suppress galvanic corrosion.

Material Selection

Selection of the lower cylinder head material was primarily based on the properties of thermal conductivity and machinability. The decision to use integral valve seats for the RPVs also meant the material had to have good impact and tensile strengths, as well as the ability to be case-hardened. Three materials were considered for selection: ISO grade GJV-450 compacted graphite iron (CGI), ASTM A536 grade 65-45-12 ductile iron, and ASTM A48 class 40 (G2) grey iron. A comparison of their mechanical properties is provided in Table 3.2, culminated from various sources [7, 42, 43, 66].
Figure 3.11: Two-piece cylinder head with copper gasket.
**Table 3.2:** Comparison of cylinder head material choices.

<table>
<thead>
<tr>
<th>Material</th>
<th>CGI GJV-450</th>
<th>Ductile Iron 65-45-12</th>
<th>Grey Iron G2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ultimate Tensile Strength (MPa)</td>
<td>450</td>
<td>448</td>
<td>300</td>
</tr>
<tr>
<td>Tensile Yield Strength (MPa)</td>
<td>315</td>
<td>310</td>
<td>300</td>
</tr>
<tr>
<td>Elongation (%)</td>
<td>1-2.5</td>
<td>12</td>
<td>1</td>
</tr>
<tr>
<td>Thermal Expansion (µm/m°C)</td>
<td>12</td>
<td>12</td>
<td>12</td>
</tr>
<tr>
<td>Thermal Conductivity (W/m-K)</td>
<td>36</td>
<td>32.3</td>
<td>39</td>
</tr>
<tr>
<td>Average Hardness (BHN)</td>
<td>240</td>
<td>180</td>
<td>241</td>
</tr>
<tr>
<td>Heat Treatable</td>
<td>Yes</td>
<td>Yes</td>
<td>Yes</td>
</tr>
</tbody>
</table>

1 G2 grey iron is considered a brittle material.

Originally, compacted graphite iron (CGI) was chosen as the desired material for the lower cylinder head, as it possess similar strength and ductility to ductile iron while maintaining higher values of thermal conductivity (similar to G2 iron). In essence, the best of both non-CGI materials. However, an economic source of pre-cast billets could not be located. Therefore, Versa-Bar® 65-45-12 ductile cast iron was used for the lower cylinder head. The product is manufactured using a continuous casting process, claimed by the manufacturer to minimize porosity and in-homogeneities, both of which can lead to cracking in high-stress applications [7]. A slightly lower thermal conductivity of the ductile iron, in comparison to the G2, was deemed an acceptable trade-off for higher strength and impact toughness.

The upper cylinder head was machined from a 6061-T6 aluminum billet. Ease of machining, high strength and excellent thermal conductivity were the three primary factors used in its selection.

### 3.4.4 Cooling Passages

Cooling of the cylinder head was accomplished by incorporating water passages into the lower cylinder head, as shown in Figure 3.12 (rendered in blue). Machining of the passages was done on a 3-axis mill, and thus direct-line tooling access was a design requirement. Three large cooling pockets were milled from the top surface and then interconnected via holes drilled from the outside surfaces of the cylinder head. The drill holes were eventually sealed to the outside using threaded pipe plugs. The spark plug, pressure transducers, and exhaust valve were given priority when laying out the cooling passages, but it was also necessary to ensure the channels could be connected with existing water jackets in the engine block. The latter enabled the Kubota head gasket to be reused.
The high thermal conductivity of the upper cylinder head, combined with its direct contact to the cooling passages, also helps to remove excess heat from the cooling system—effectively working as a large heat sink. This was also a deciding factor when selecting the gasket material to seal the two head components. The pure copper gasket meets the requirement of good thermal conductivity and mechanical strength, while remaining physically compliant to surface irregularities. Although the cooling passages designed for this cylinder head have worked well under a variety of engine conditions, there is undoubtedly room for their improvement. A cast cylinder head would have allowed more evenly distributed cooling to be achieved, but would also have been considerably more difficult and costly to produce.

### 3.4.5 Reverse Poppet Valve Seats

Instead of using valve seat inserts, the reverse poppet valve seats were machined directly into the cylinder head material. This eliminated the packaging and seat retention difficulties previously discussed in this section. To improve the wear characteristics of the valve seat surface, the region was flame hardened to HRC 50 and then tempered back to HRC 35-40 in order to prevent brittle fracturing. The nodular graphite spheres present in ductile iron provides some inherent lubrication for the valve seat, which also helps prevent wear [144].
To promote a good mechanical seal between the valve and its seat, the chamfer of the seat was decreased by one degree (from 45° to 44°), as shown in Figure 3.13. This provides a ‘wedging’ action as the valve pushes against its seat and supposedly helps achieve good annular contact [144]. With the cylinder head assembled, both crossover valves were lapped using Loctite® Clover lapping compound. The head was then disassembled, thoroughly cleaned, reassembled, and pressure checked to ensure proper valve sealing.

![Figure 3.13: Angular difference between the reverse poppet valve and its seat.](image)

### 3.4.6 Spark Plug and Transducer Locations

In spark ignition engines, *knock* can be defined as the autoignition of end-gas before it is consumed by the travelling flame front. It generally occurs due a lengthy combustion duration (e.g. slow burn, large bore), or due to ignition from another source (e.g. the hot edge of a protruding head gasket). By locating the spark plug adjacent to the exhaust valve, which is typically the hottest part of the combustion chamber, and centred in the bore, the propensity of knock can be greatly reduced [131]. This was the basic criteria used for laying out the spark plug and exhaust valve in the split-cycle engine.

It is worthwhile to note that in the work of Phillips [115], two spark plugs were used in their split-cycle engine. They note a performance advantage when using two spark plugs and one exhaust valve, compared with a single spark plug and two exhaust valves. However, the vast majority of production engines rely on a single spark plug, and in the interest of doing burn rate comparisons with current literature, a single spark plug configuration was used.

Figure 3.14 shows the layout of major cylinder head components within each engine cylinder. The location of the pressure transducer in the combustion chamber is at the furthest possible distance from the spark plug in order to delay its exposure to the flame front. The high temperature of the flame causes thermal distortion of the sensor’s diaphragm,
which introduces a slight error in its measurement, commonly known as \textit{thermal drift} [121]. By locating the transducer near the cylinder wall, the sensing element is subject to the high combustion and post-combustion temperatures for the least amount of time. The installation depth is also important to prevent pipe oscillations from occurring between the measuring diaphragm and the combustion chamber. All transducers were installed in accordance with the recommended specifications given by Kistler [83]. A detailed schematic of the installation is given in Section 4.4.3 of Chapter 4.

![Diagram of cylinder head components](image)

\textbf{Figure 3.14:} Layout of cylinder head components as viewed from the cylinder bore. All dimensions in mm.

### 3.4.7 Crossover Passage and Fuel Injector

Accounting for the two valves and their guides, the volume of the crossover passage is approximately 0.122 L, which equates to roughly one-half of the cylinder displacement volume. However, because the crossover passage remains at high pressure, only 10–15\% of its mass is exchanged per cycle in actual practice. This mass ratio was somewhat arbitrarily selected, since no literature currently exists discussing the optimal size of a crossover passage. Nevertheless, the relatively large crossover volume was done intentionally to provide considerable residence time for air/fuel mixing. The size of the passage was ultimately determined by packaging constraints imposed by higher priority components, such as the head bolt holes and the valves. The creation of a homogeneous charge inside the crossover passage is one of the novelties of this engine, and was done to avoid the issues of poor mixture homogeneity discovered by Phillips [115] (see Section 2.1.4). The use of a high octane fuel, such as methane or NG, is a desirable in this case to avoid autoignition in the crossover passage.
The fuel injector was positioned approximately midway between the two crossover valves, as shown in Figure 3.15, where it injects directly at the convex curvature of the opposing wall. The intent is to break-up the solid gaseous jet as it emerges into the crossover passage, and accelerate the mixing process. It can be seen that the spray cone is off-axis from the injector body due to the spray-guided type of injector that was used. The cone is angled back towards the inlet crossover valve to prevent a rich pocket of fuel from forming near the outlet port, which could easily make its way into the combustion chamber. This is in direct contrast to the split-cycle engine developed by Phillips [115], where fuelling is directed to the backside of the crossover outlet valve. The ability to relocate the injector is afforded by the high octane rating of the fuel. Also, since the fuel is gaseous, there is no need to be concerned about condensation on the passage walls.

Fuel is delivered into the crossover passage by a Siemens VDO gasoline DI, single hole fuel injector, part number: 94860523002. This injector was selected due to its compact size and its M12x1.5 mm, 30° JIS bull-nose fuel connection, which does not require the use of a clamp-down style fuel rail. The injector is held in place by two clamping washers, bolted down to a pair of stand-offs threaded into the cylinder head. An annular Teflon® ring, located on the snout of the fuel injector, provides a seal for the high pressure gases in the crossover passage. Since the injector was designed to be used with a liquid, it is expected that long-term use with a gaseous fuel may cause damage due to the lower fuel lubricity. Injector flow rates were monitored throughout the testing regime for signs of malfunction. One of the benefits to injecting fuel in the crossover passage, versus in-cylinder, is that injector leakage has no

![Figure 3.15: Graphical representation of the location and approximate direction of the fuel spray inside the crossover passage. View is from the top side of the cylinder head.](image-url)
affect on emissions, provided the correct air/fuel ratio is maintained. A photograph of the injector installed in the cylinder head is shown in Figure 3.16.

![Fuel Injector and Stand-off Restraint](image)

**Figure 3.16:** Photograph of fuel injector installed on engine.

### 3.5 Reverse Poppet, Overhead Cam (OHC) Valvetrain

#### 3.5.1 Overview

The operation of the split-cycle engine is such that the crossover valves must be opened when both pistons are in the vicinity of their respective TDC positions. Furthermore, it is desirable to minimize the clearance volume of each cylinder in order to fully expel the gases at the end of the cycle, thereby maximizing the volumetric efficiency potential of the engine. With conventional, inward opening valves, these two characteristics are concurrently incompatible. Recesses in the piston crown, required to prevent mechanical contact between the piston and valve, are counter-productive for minimizing the clearance volume. It was for this reason that reverse poppet valves (RPVs), which open outward from the cylinder, were implemented to control the flow in and out of the crossover passage.

This section covers the design and development of the RPV valvetrain. To familiarize the reader with its basic operation and component nomenclature, a section view of the valvetrain is shown in Figure 3.17. The RPV is pulled upwards from the cylinder by a pivoting rocker arm, which is actuated by the lobe of an overhead camshaft (OHC). A coil spring located above the valve provides the necessary control and closure force. With the exception of the valve guide, valve seal, and bearings, all components were designed by the author.
Chapter 3: Split-Cycle Engine Design

Figure 3.17: Cross-sectional rendering of OHC valvetrain components.

The design constraints are outlined in Section 3.5.2, followed by the development of the valve lift profile in Section 3.5.3. The process and simulation model used to generate the physical valvetrain geometry are presented in Sections 3.5.4 and 3.5.5, respectively. Details of the individual components are covered in Sections 3.5.7 to 3.5.10.

3.5.2 Design Constraints

Because a new cylinder head was being manufactured to accommodate the OHC valvetrain, the primary design constraints were those imposed by the operation of the RPVs. In other words, the characteristics of the valve lift profile were selected first, and then used to dictate the remainder of the design. The only other constraint, was that imposed by the existing (pushrod) valvetrain in terms of component packaging.
The initial determination of the crossover valve durations began with the crossover inlet valve. The basic operating premise of the split-cycle engine dictates the crossover inlet valve should nominally close at TDC. Furthermore, the valve’s opening time should occur when the cylinder pressure is approximately equal to the crossover passage pressure. The conditions of the latter are limited by the autoignition temperature of the fuel. Thus, working backwards from this point, the autoignition temperature of methane at elevated pressure (e.g. \( \sim 660 \text{ K at 25 bar} \)) was used to iteratively solve for the allowable in-cylinder pressure at the end of compression. For polytropic indices ranging from \( n = 1.3 \) to \( n = 1.4 \), the limiting pressure for autoignition ranged from 18 bar to 31 bar. Based on the engine geometry, the opening time of the crossover inlet valve was then determined from Figure 3.18. As one would expect, the figure shows that increased heat transfer during compression (i.e. a lower value of \( n \)) allows higher pressures to be achieved before reaching autoignition. The variation of \( n \) translates into a range for the opening time of the crossover inlet valve, from 35°CA BTDC to 25°CA BTDC. To err on the side of caution, the design duration of the crossover inlet valve was set to 35°CA (excluding the opening and closing ramp periods). The limits shown in Figure 3.18 are also conservative, considering additional heat losses will occur inside the crossover passage.

![Figure 3.18: Polytropic compression curve showing theoretical timing of pressure equalization between cylinder and crossover passage.](image)

The crossover outlet valve duration was set to equal that of the crossover inlet valve. This was done based on the expectation that different durations might lead to a build-up of pressure in the crossover passage. It also minimized the required number of cam lobe designs, accelerating the design process.
As a consequence of the short opening duration, the desired peak valve lift was constrained to 3.5 mm. This value was based on similar fast-acting valvetrain designs [65, 99]. Table 3.3 summarizes the design constraints for the valvetrain. The ramp heights and velocities were values recommended by Wang [144]. For reference, a typical lift value for a conventional valvetrain is on the order of 8–10 mm.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Lift</td>
<td>3.5</td>
<td>mm</td>
</tr>
<tr>
<td>Lift Duration(^1)</td>
<td>35</td>
<td>deg</td>
</tr>
<tr>
<td>Opening Ramp Height</td>
<td>0.3</td>
<td>mm</td>
</tr>
<tr>
<td>Closing Ramp Height</td>
<td>0.3</td>
<td>mm</td>
</tr>
<tr>
<td>Opening Ramp Velocity</td>
<td>0.031</td>
<td>mm/deg</td>
</tr>
<tr>
<td>Closing Ramp Velocity</td>
<td>0.031</td>
<td>mm/deg</td>
</tr>
</tbody>
</table>

\(^1\) Excluding ramps

### 3.5.3 Valve Lift Profile

The short duration of the RPVs was a considerable design challenge in regards to valvetrain kinematics. To put it into perspective, consider a conventional valve with a peak lift of 10 mm over a 180° period. A crude approximation of the valve acceleration can be found by dividing the lift by the square of the duration, which yields $3 \times 10^{-4}$ mm/deg\(^2\). By comparison, the pseudo-acceleration of the split-cycle valve is $3 \times 10^{-3}$ mm/deg\(^2\)—one order of magnitude larger. Furthermore, it has been shown by Blair et al. [19] that changes on the order of ±0.01 mm in the valve lift profile, can alter the subsequent acceleration by as much as 40%. In other words, the design of a valve lift profile is no trivial task.

In order to produce a kinematically feasible valve lift profile, the author has developed a profile generation and optimization tool using MATLAB\(^\text{®}\). Only a brief overview of the tool will be provided here; however, additional details can be found in Cameron and Minaker [29]. Essentially, the MATLAB\(^\text{®}\) tool uses a built-in optimization algorithm to alter the polynomial coefficients that make up the various segments of a valve acceleration profile. The user starts by implementing the design constraints, such as valve lift and overall duration, as well as an initial guess of the polynomial coefficients. The MATLAB\(^\text{®}\) script solves the kinematic equations for lift, velocity, acceleration, and jerk, and assesses their
characteristics against a user-defined set of metrics, known as the penalty function. Each metric in the penalty function is weighted, allowing the user to fine-tune the desired output. The MATLAB® tool enabled the author to successfully create a valve lift profile meeting the specifications given in Table 3.3 without a considerable amount of trial and error. The final lift profile selected for use in the split-cycle engine is shown in Figure 3.19. A reduction in peak lift, from 3.5 mm to 3.43 mm, can be seen and is a result of the smoothing algorithm used. The overall and ramp-to-ramp durations are 54.4 degrees and 36 degrees, respectively.

![Figure 3.19: Lift curve for reverse poppet valves.](image)

### 3.5.4 Valvetrain Geometry

To physically produce the valve lift motion shown in Figure 3.19, the lift profile was transformed into a cam lobe contour. The cam profile is dependent on the valvetrain geometry, which in turn is limited by certain aspects of the profile. Thus, generation of the cam lobe and valvetrain geometry was an iterative process. A schematic of the finalized geometry used in the split-cycle engine is shown in Figure 3.20.

Two primary factors had to be considered when generating the camshaft lobe:

1. The minimum concave radius of curvature on the lobe’s surface, governed by the radius of the roller-follower. In this case the roller size of 8.875 mm was predetermined based on availability. Therefore, to prevent undercutting of the roller (i.e. multiple lines of contact between the roller and cam), all concave regions of the cam profile were required to have a radius larger than 8.875 mm.
Figure 3.20: Geometry of overhead camshaft valvetrain.
2. The pressure angle, $\phi'$, defined as the angle between the velocity vector of the follower $(V_R)$ and the axis of force-transmission between the cam lobe and follower. In other words, $\phi'$ is the angle between the follower's velocity vector and normal force, as depicted in Figure 3.20. The pressure angle is a representation of the force magnitude being applied in the slip direction of the roller. A high value can result in unnecessary frictional losses, and it is recommended by Norton [107] to maintain a pressure angle within the range of $\pm 30$–35 degrees.

To generate the cam lobe profile from the translational valve motion, the rocker arm geometry was used to determine the Cartesian coordinates of the follower axis over the course of the valve lift profile. These coordinates were subsequently converted into the pitch curve of the cam lobe. The cam profile was then generated by offsetting the pitch curve using Equations (3.4) and (3.5), where $f(\theta)$ and $g(\theta)$ are the Cartesian coordinates of the pitch curve, and $R_f$ is the follower radius. Higher-order numerical differentiation was used to minimize the error between the two curves.

$$x(f, g) = f(\theta) \pm \frac{(R_f) \left( \frac{dg}{d\theta} \right)}{\sqrt{\left( \frac{df}{d\theta} \right)^2 + \left( \frac{dg}{d\theta} \right)^2}}$$

$$y(f, g) = g(\theta) \pm \frac{(R_f) \left( \frac{df}{d\theta} \right)}{\sqrt{\left( \frac{df}{d\theta} \right)^2 + \left( \frac{dg}{d\theta} \right)^2}}$$

To evaluate the pressure angle over the duration of the cam lobe, the triangle OBA from Figure 3.20 was first used to calculate the centreline distance between the cam and the rocker arm pivot, $l_{OB}$. From the law of cosines, where $\beta = 105^\circ$ as recommended by Wang [144]:

$$l_{OB} = \sqrt{l_{AB}^2 + l_{OB}^2 - 2 l_{AB} l_{OB} \cos \beta}$$

(3.6)

Similarly, the angle between lengths $l_{OB}$ and $l_{OA}$ can then be solved from Equation (3.7):

$$\Psi = \cos^{-1} \left[ \frac{l_{OB}^2 + l_{OA}^2 - l_{AB}^2}{2 l_{OA} l_{OB}} \right]$$

(3.7)

Combining Equations (3.6) and (3.7), the pressure angle is then defined by Equations (3.8) and (3.9) for the rise and fall portions of the profile, respectively, where $d\phi'/d\theta$ is the angular velocity of the roller in rad/rad for each crank angle being evaluated.
\[ \phi' = \frac{\pi}{2} - \sin^{-1}\left(\frac{l_{OB}\sin\delta'}{l_{OA}}\right) + \tan^{-1}\left[\frac{l_{OA}^2}{l_{AB}l_{OB}\sin\delta'\frac{d\delta'}{d\theta}} - \frac{l_{OB}^2 - l_{OA}^2 - l_{BC}^2}{2l_{OA}l_{OB}\sin\Psi}\right]^{-1} \] (3.8)

\[ \phi' = -\frac{\pi}{2} + \sin^{-1}\left(\frac{l_{OB}\sin\delta'}{l_{OA}}\right) + \tan^{-1}\left[\frac{l_{OA}^2}{l_{AB}l_{OB}\sin\delta'\frac{d\delta'}{d\theta}} + \frac{l_{OB}^2 - l_{OA}^2 - l_{AB}^2}{2l_{OA}l_{OB}\sin\Psi}\right]^{-1} \] (3.9)

In an iterative process using Equations (3.6) to (3.9), the base circle radius of the cam was incrementally increased until an acceptable curvature and pressure angle were obtained. This was done in conjunction with changes to the overall valvetrain geometry, which also affects the pressure angle. The final lobe design has a 40 mm base circle radius, a minimum (concave) radius of curvature of 19.8 mm and a maximum pressure angle of 29.8°. A larger base circle would favourably decrease the pressure angle; however, as the radius increased packaging became more difficult. A schematic of the resulting cam contour and pitch profile are plotted using polar coordinates in Figure 3.21.

![Figure 3.21: Camshaft profile and pitch curve shown on a polar coordinate system.](image-url)
3.5.5 Dynamic Simulation

As part of the validation process for the valvetrain design, a multi-body dynamic model was created using LMS Virtual.Lab.Motion software. Figure 3.22 shows the key elements of the model that heavily influence the dynamic response of the valvetrain. The primary purpose of the simulation was to dynamically evaluate the valvetrain over a range of engine speeds. In doing so, realistic operating behavior of the valvetrain could be assessed. In particular, the model was used to check for conditions of valve float\(^2\) and valve bounce\(^3\), both of which are undesirable phenomena that can damage the engine.

![Figure 3.22: LMS Virtual.Lab.Motion dynamic valvetrain model.](image)

The mass, inertia, and stiffness of each component shown in Figure 3.22 is automatically determined by the software program based on material and geometry specifications. All bearings were considered to be rigid supports with one degree of freedom and a fixed damping value. In an effort to maximize the model’s fidelity, tessellated contact elements were used to transmit forces between unaffixed components, allowing them to separate if needed (e.g., valve float). Furthermore, the frequency response of the valve spring was also made realistic by inclusion of a flexible spring element, composed of thirty-six individual bodies,

---

\(^2\) Valve float is the loss of driving contact between the camshaft and the valve.

\(^3\) Valve bounce is when the valve does not stay seated upon initial closure and partially re-opens.
and ten beam elements per body. The mode shapes of the spring were superimposed onto the static coil deflections in order to create a dynamically accurate response. Inter-coil contact was also accounted for, which provides damping to the spring and is an important phenomenon in regards to spring surge\footnote{Spring surge is the oscillation of the spring's coils at their resonant frequencies or mode shapes.} and, consequently, valve float. Frictional damping at the translating interface between the valve guide and the valve was also employed.

Figures 3.23 and 3.24, show the simulated lift and acceleration profiles, respectively, for the selected valvetrain design operating at 1500 RPM. The dashed lines indicate the kinematic design curves, and it is evident from Figure 3.23 that the valve lift follows very closely to the designed profile; the slight difference is caused by the valve lash present in the dynamic model, deflection of the non-rigid contact elements, and truncation errors in the solver.

![Figure 3.23: Comparison of kinematic and dynamically simulated valve lift.](image)

The simulated acceleration profile, Figure 3.24, shows a ‘stepped’ characteristic that is speculated to be caused by the differential algebraic equation (DAE) method used to solve the equations of motion. The contact elements used in the model are essentially very stiff springs that prevent penetrating contact between two bodies while still allowing them to separate. Because of the high material stiffness, the contact elements create a \emph{stiff problem}, meaning their frequencies are several orders of magnitude larger than other frequencies in the model, namely the flexible valve spring bodies. These vastly differing frequencies make it difficult for the numeric solver to properly select a calculation time step. This was observed when the user-allotted minimum time step was set too large and the solver could not converge. The solver was also very sensitive to the parameters used for each contact
element. For these reasons, it is believed the “stepped” curve shown in Figure 3.24 is a manifestation of the solver. The sharp peaks at the beginning and end of the simulated profile are from the valve lash and will likely be much lower in the actual engine due to additional damping by the oil-soaked surfaces.

It is obvious the dynamic valvetrain model contains significant deficiencies and could use improvement. However, despite the lack of fluency in Figure 3.24, the magnitude and trend of the simulated acceleration profile correlate well with the kinematic design curve. As such, the model proved to be useful for providing an approximation of the engine speed at which valve float and/or bounce starts occurring. By monitoring the contact forces as the simulated engine speed was increased in 100 RPM intervals, valve float was determined to be problematic at approximately 1700 RPM. This number is based on a maximum spring preload of 435 N. Given the lack of confidence in the model, the engine speed has been conservatively capped at 1500 RPM in order to provide a margin of safety. At 1500 RPM, the minimum spring preload force required to prevent valve float is approximately 305 N. Valve bounce was not detected for any engine speed evaluated up to 1800 RPM, likely due to the lengthy closing ramp of the camshaft profile.

**Design Process Overview**

A flow-diagram showing an overview of the complete valvetrain design process is given in Figure 3.25.
Chapter 3: Split-Cycle Engine Design

Figure 3.25: Design process used to develop the reverse poppet valvetrain.
3.5.6 Camshaft

Design and Manufacture

With the exception of some high-cost solutions, cam timing is generally fixed for any given engine application. The cam lobes are ground directly into the camshaft, and the phasing between lobes is therefore non-adjustable. In the case of the split-cycle engine, it was desirable to be able to vary the timing between the inlet and outlet valves of the crossover passage, since it was not clear at the beginning what the optimal phasing of each should be. To accomplish this task, the camshaft and its lobes were made as separate pieces. The cam lobes were then bolted onto an axial flange machined into the base shaft, and the friction force between the lobe and flange provides the necessary means of torque transmission. The bolt holes in the lobe are slotted, which allows for 30 degrees of radial (phase) adjustment. The inner diameters of the lobes were machined by wire EDM to the exact size of the shaft, within 5.1 \( \mu \text{m} \), ensuring concentricity between the shaft and base circle of the lobe. While this solution does not allow for on-the-fly timing adjustments, it does satisfy the prerequisite for a flexible timing requirement without significant cost and/or complexity. It also provides a cost-effective route for changing valve lift profiles, should it be required. The upper cylinder head was designed so that the cam is easily accessible and can be removed/installed without dismantling or re-adjusting any other valvetrain components.

Alloy 1018 hot rolled steel was used for the material of the OHC shaft base, primarily since it is easily hardened and machinable. The shaft was manufactured using an OD grinding process by Lindham Industries in Windsor, ON. The bearing surfaces and lobe mounting locations, shown in Figure 3.26, were hardened to a minimum of HRC 50. Complete dimensional drawings of the shaft can be found in Appendix A.

The camshaft lobes were made from 100Cr6 (SAE 52100) alloyed steel, a material commonly used to manufacture bearing races. This material has an allowable rolling contact stress limit of 4.2 GPa [68] making it ideally suited for high contact stress applications, such as those encountered in a cam-follower system. For reference, the maximum static contact stress on the cam lobe was calculated to be approximately 750 MPa, occurring at 3.2°CA before the lobe centreline. To prevent rapid surface wear, both overhead cam lobes were heat-treat hardened to HRC 58. This was done before final machining of the lobe profile, which ensured geometric tolerances were not compromised from thermal distortion sustained during the hardening process.
The final machining of the lobe profile was done by wire EDM. In order to minimize valvetrain friction, it is recommended by Wang [144] that the cam lobe surface roughness not exceed 1.2 \( \mu m \). Some trial and error iterations were necessary to achieve the desired surface finish and geometric tolerance using the EDM manufacturing process. Figure 3.27 shows the progression of machining: from left is a blank profile cut from aluminum to verify the geometric shape, followed by the same piece cut from 100Cr6 (SAE 52100) steel to determine the number of wire passes necessary to achieve an adequate surface finish; the remaining three pieces show the final lobe as the contact surface was smoothed by hand with progressively finer abrasive, from 1200 grit to a micro-polishing compound.

**Bearings and Lubrication**

The overhead camshaft rides on three pressure-fed, hydrodynamic bearings manufactured by Mahle Clevite, part number: CB-1479P. The oil flow is supplied by an external oiling system (discussed in Section 4.2.2) that pumps oil through passages drilled into the upper cylinder head. Roller thrust bearings accompanied by steel shims provide low-friction retention in the axial direction and a means of adjusting the camshaft end-play, set to a value between
0.13–0.25 mm. The bearing caps are held in place using bolts with self-piloting shanks that double as locating dowels. This was done to reduce the overall size of the bearing caps, thereby minimizing their packaging space.

The camshaft bore was the last machining operation to be performed on the upper cylinder head. The entire bore was machined in a single setup with the bearing caps bolted in place to ensure roundness and concentricity of each journal. A custom ground-to-size reamer was purchased to achieve the finish diameter of 41.013±0.006 mm. The caps were then marked for location and orientation before removal, and should not be mixed up during future assembly if proper bearing clearances are to be maintained. An exploded-view CAD rendering of the OHC bearing and lubrication system is shown in Figure 3.28. Not shown are two oil squirter nozzles mounted in the valve cover that supply the cam lobes with a constant stream of lubricating oil. Also, the front thrust bearing assembly is not visible, but mirrors that of the rear.

3.5.7 Reverse Poppet Valve

The reverse poppet valves were designed by the author and manufactured by Ferrea Racing Components in Ft. Lauderdale, Florida. The valves are made of a proprietary stainless steel that was recommended by the manufacturer for use with a low-lubricity fuel like methane.
Each valve has a head diameter of 25 mm and a stem diameter of 6 mm that is undercut\(^5\) to 5.3 mm for weight reduction. A photograph of the valve is shown in Figure 3.29.

![Figure 3.29: Photograph of the reverse poppet valve designed in the present work.](image)

Opening of the valve occurs as the result of the rocker arm motion acting on a flanged nut that is threaded onto the stem of the valve. The flanged nut is fixed and prevented from turning by a jam nut. Use of a threaded valve stem allows for the position of the flanged nut to be adjusted vertically in order to set the proper valvetrain lash.\(^6\) The M\(6 \times 0.75\) thread was ground into the valve stem by H&K Thread Grinding Co. of Madison Heights, Michigan. Complete dimensions of the valve are given in Appendix A.

### 3.5.8 Reverse Poppet Valve Spring

Valve springs serve two purposes: 1) to maintain a closure force on the valve in its static state, and 2) to oppose the inertial forces present during the negative acceleration period over the nose of the cam. The crossover valves present a unique challenge since they are subject to high fluid pressure forces acting on both sides of the valve head. During the majority of the cycle, the pressure maintained in the crossover passage acts to hold the reverse poppet valves closed against their seats. Consequently, the limiting spring force is that which prevents valve float at the maximum engine speed design limit. From the dynamic simulation, discussed in Section 3.5.5, the minimum spring force required (excluding gas forces) is approximately 305 N. However, the pressure of combustion imparts a force on the valve in the opposite direction and therefore, depending on the crossover pressure, an additional spring force may be required to keep the valve closed.

---

\(^5\) Undercut refers to the removal of material along the valve stem near the head of the valve.

\(^6\) Lash is a small mechanical clearance required for thermal expansion of the system components.
Design Methodology

The design of the RPV spring was a challenge due to the unknown pressure profiles of the crossover passage and expansion cylinder ahead of having the engine operational. Both of these parameters are affected by numerous variables such as valve timing, volumetric efficiency, seal leakage, air-fuel ratio, heat transfer, and combustion duration. In the absence of literature with specific similarity to this work, accurate prediction of such parameters would require a high-fidelity engine model to be used. With no such model available, an adjustable and modular valve spring configuration was implemented to provide flexibility around the operating conditions.

In order to quickly evaluate and compare different spring designs, a MATLAB® script (valve_spring_design.m in Appendix C) was developed by the author to programmat-ically select and display a list of valid spring designs, based on a number of user-input parameters and fundamental spring design guidelines. In this manner, quick design changes were possible as the valvetrain evolved. The MATLAB® script uses a series of graphical user interface (GUI) selection boxes to prompt the user to make a priori design decisions such as: spring material, end conditions, shot-peening, presetting, factor of safety, and the spring forming process (cold or hot). The user must also select one or more pre-defined wire diameters. Next, the hard-limit constraints are input, which allows the upper and lower geometric limits, the number of cycles for fatigue, and the desired spring frequency ratio to be specified. Once the design variables are input into the program, the script then calculates a comprehensive set of spring parameters based on the formulae found in Budynas and Nisbett [26]. For each wire diameter that was selected, a set of parameters is calculated for a range of spring indices\(^7\), ranging in value from 4 to 12, in increments of 0.5. The result is a series of 2-D arrays representing a single output variable as a function of the spring index, \(c\), and the wire diameter, \(d\). Within each array, coordinates of the cells that fall meet the design requirements are extracted, and a counter is used to monitor the number of times a single set of coordinates is returned. The cell corresponding to the highest count is likely to be the one with the most practicable design, as it meets the largest number of constraints. All spring configurations that meet the minimum design constraints are then output to a table, in order of decreasing count number, for user review and selection. An overview of the automated spring selection process is given in Figure 3.30.

To minimize costs, a standard catalogue spring that closely matched the design specifications was purchased from Diamond Wire Company, part number: DWC-148JJ-12. The

\(^7\) The spring index, \(c\), is the dimensionless ratio of spring OD to wire diameter, \(d\). Values below 4 and above 12 are difficult and costly to manufacture.
specifications of the spring are given in Table 3.4. A relatively large coil pitch (free length vs. solid height) was used to allow for a greater range of spring preload to be applied.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>Cr-Si</td>
<td>-</td>
</tr>
<tr>
<td>Free Length</td>
<td>50.8</td>
<td>mm</td>
</tr>
<tr>
<td>Outer Diameter</td>
<td>22.0</td>
<td>mm</td>
</tr>
<tr>
<td>Inner Diameter</td>
<td>14.5</td>
<td>mm</td>
</tr>
<tr>
<td>Wire Diameter</td>
<td>3.75</td>
<td>mm</td>
</tr>
<tr>
<td>Solid Height</td>
<td>31.9</td>
<td>mm</td>
</tr>
<tr>
<td>Natural Frequency</td>
<td>580</td>
<td>Hz</td>
</tr>
<tr>
<td>Spring Rate</td>
<td>42.8</td>
<td>N/mm</td>
</tr>
</tbody>
</table>

Table 3.4: Reverse poppet valve spring specifications.

Valve Forces and Spring Preload

As previously mentioned, the net force acting on the valve ($F_{net}$) is the summation of the pressure and spring preload forces:

$$F_{net}(p_c, p_x) = A_f p_c - A_b p_x - F_{sp} \tag{3.10}$$

where $p_c$ is the magnitude of cylinder pressure, $p_x$ is the magnitude of crossover pressure, $A_f$ is the front (cylinder) side area of the valve head, $A_b$ is the back (crossover) side area of the valve head, and $F_{sp}$ is the preload force provided by the valve spring.
Nominally, the front-side area of the valve is greater than the backside area \((A_b/A_f=1.42)\). However, the sealing surface does not consume the entire face of the valve, and thus the actual area ratio is less than 1.42. The difference was believed to be negligible at the time of selecting a valve spring, but this was later determined not to be the case. The consequences and limitations that this error imposed are discussed in Section 6.4.2.

Regardless of the actual ratio value, the fact that the pressure-exposed areas of the valve are not equal, from front-to-back, means the pressure differential required for force equilibrium is dependent on the magnitude of the pressures. In other words, as the backside (crossover) pressure is increased, the differential pressure across the valve required for zero net force also increases. Thus, a higher crossover passage pressure allows for a linearly greater overshoot in cylinder pressure. This is shown in Figure 3.31, using the nominal ratio of \(A_b/A_f=1.42\), for the condition of no spring, and for the range of possible spring preloads. The minimum preload is based on the required spring cover for dynamic stability, and the maximum corresponds with the closure force of the spring outlined in Table 3.4. An updated version of Figure 3.31 is given in Section 6.4.2, based on experimental observations.

![Figure 3.31: Maximum allowable pressure difference \((\Delta p = p_c - p_x)\) as a function of the absolute pressure in the crossover passage, \(p_x\).](image)

**Preload Adjustment Mechanism**

The reverse poppet valve springs are located atop of each valve, inside threaded bores of a separate steel structure. This structure, referred to as the *valve spring bridge*, is fastened within the upper cylinder head and designed to be modular in order to facilitate easier
installation of the valvetrain components. Furthermore, the modularity of the valve spring bridge is a provision to accommodate future spring changes (e.g. pneumatic valve springs), which may be a necessity if engine speeds in excess of 1500 rpm are to be explored [99]. The preload on the spring is increased by tightening the upper spring stop down into the bridge, effectively compressing the spring against the RPV. To prevent the adjustment stop from turning while the engine is operating, a secondary threaded plug (referred to as the adjuster lock) is tightened against the stop. The adjuster lock has a bored-through centre, providing access to the internal M10 hex drive of the preload adjustment plug. Figure 3.32 shows a CAD rendering of the valve spring bridge location within the cylinder head, as well as an enhanced breakout view of the spring preload adjustment mechanism.

\[\text{Figure 3.32: CAD rendering of the valve spring bridge positioned in the upper cylinder head (left), and a detailed breakout view of the preload adjustment mechanism (right).}\]

### 3.5.9 Reverse Poppet Rocker Arm

The reverse poppet valvetrain is essentially a standard rocker-follower OHC configuration that has been inverted, so the rocker arm pulls upward (instead of pushing downward) on the valve stem. To accomplish this, the rocker arm has a forked end, referred to as the tappet side, that straddles the valve stem, allowing it to lift equally on both sides. The pivot axis of the arm is also located below the centreline of the camshaft (see Figure 3.17).

For the purpose of minimizing operational friction, the rocker arm was designed as a full roller type, meaning needle bearings were incorporated in both the pivot and follower. The pivot bearing that was used is a catalogue part purchased from Jesel Inc., part number: BRG-20610. The follower bearing assembly was re-purposed from a Chrysler 3.0 L V6 engine.
valvetrain with the exception of the bearing’s axle, which had to be discarded since it was originally designed by Chrysler to be a single-use part. To form a new bearing axle, a piece of 8620 steel round-stock was case hardened to HRC 60, ground to size, and micro-polished. The axle was swaged into the rocker arm after assembling the bearing.

The geometric constraints on the rocker arm were largely governed by the available bearings and the allotted packaging space. To reduce weight and increase stiffness, the rocker arm was made as short as possible, limited by the clearance between valve spring and cam lobe. A rocker ratio of 1:1 was used since the valve lift height is small (3.5 mm) compared to the base circle radius of the cam (40 mm), rendering rocker arm lift multiplication undesirable and unnecessary. Achieving a light weight rocker arm design was crucial due to the relatively large inertial forces present in the valvetrain. In order to minimize these forces, the rocker arm’s mass moment of inertia was reduced by approximately 31 % from initial conception to the final product. The generative structural analysis (GSA) workbench in CATIA was used for finite element stress analysis of each rocker arm design. A force of 2000 N was applied to a mock tappet in contact with the rocker arm, roughly simulating the superposition of the worst-case spring and gas pressure forces. A multi-body contact element was used to emulate the deforming line-contact present between the two bodies. The follower axle axis was held fixed, but a deformable contact element allowed for elongation of the axle bore. Table 3.5 shows the deflection, mass and inertial values of progressive rocker arm designs for a material selection of 4340 steel. Deflection of the needle bearings was neglected.

Further mass reduction in the rocker arm is limited by a trade-off in stiffness and also the necessary wall thickness required to support the interference fit of the pivot bearing. The rocker arm FEA model, combining the effects of the bearing press-fit and the tappet force, is shown in Figure 3.33. The high Hertzian stress at the line of contact between the rocker arm and tappet is evident from the figure. Consequently, both components were coated with diamond-like carbon (DLC), a thin-film coating that has been shown to significantly reduce valvetrain friction and improve wear characteristics in high contact stress applications [58, 79]. Figure 3.34 is a photograph of the coated and fully assembled rocker arms prior to being installed in the engine.

3.5.10 Tappet and Spring Retainer

The tappet and spring retainer undergo translational motion with the valve and were therefore manufactured from Ti-6Al-4V titanium to minimize their mass. A weight savings of
### Table 3.5: Summary of RPV rocker arm design revisions.

<table>
<thead>
<tr>
<th>Part Rev.</th>
<th>Mass (g)</th>
<th>$I_{xx}^\dagger$ (kg mm$^2$)</th>
<th>Max. $\Delta_d^\ddagger$ (mm)</th>
<th>Revision Notes</th>
<th>Image</th>
</tr>
</thead>
<tbody>
<tr>
<td>R1</td>
<td>100.7</td>
<td>26</td>
<td>0.022</td>
<td>Base design.</td>
<td><img src="image1.png" alt="Image" /></td>
</tr>
<tr>
<td>R2</td>
<td>92.4</td>
<td>24</td>
<td>0.024</td>
<td>Tappet-side width reduction.</td>
<td><img src="image2.png" alt="Image" /></td>
</tr>
<tr>
<td>R3</td>
<td>88.8</td>
<td>23</td>
<td>0.025</td>
<td>Lightening pockets added to side of rocker.</td>
<td><img src="image3.png" alt="Image" /></td>
</tr>
<tr>
<td>R4</td>
<td>78.5</td>
<td>20</td>
<td>0.030</td>
<td>Re-contoured top and bottom profile in low-stressed areas.</td>
<td><img src="image4.png" alt="Image" /></td>
</tr>
<tr>
<td>R5</td>
<td>72.3</td>
<td>19</td>
<td>0.031</td>
<td>Radius of tappet contact patch increased to reduce Hertzian stress. Reduction of mid-section width.</td>
<td><img src="image5.png" alt="Image" /></td>
</tr>
<tr>
<td>R6</td>
<td>68.1</td>
<td>18</td>
<td>0.031</td>
<td>Roller-follower slot extended and contoured around pivot bearing.</td>
<td><img src="image6.png" alt="Image" /></td>
</tr>
</tbody>
</table>

$^\dagger$ Moment of inertia around centre pivot of rocker arm.

$^\ddagger$ Maximum change in displacement of valve lift for a load of 2000 N.
Figure 3.33: Finite Element Analysis (FEA) of the rocker arm for a 2 kN tappet force and 25.4 \( \mu \text{m} \) bearing press-fit. Red contours indicate a Von-Mises stress around 400 MPa.

Figure 3.34: Photograph of fully assembled, DLC-coated, RPV rocker arms. Note the small tack welds on the follower bearing axle that were used to prevent the axle from moving during the swaging process.
approximately 6.5 g per valve or 14% of the sprung mass was achieved using titanium compared with low-carbon steel. As previously mentioned, the tappet was coated with a DLC coating to reduce friction. Dimensional drawings of these components can be found in Appendix A.

### 3.6 Pushrod Valvetrain

The intake and exhaust valves of the split-cycle engine are actuated using a pushrod style valvetrain shown in Figure 3.35. The majority of the components, such as the springs, rocker arms, pushrods, tappets, etc., were retained from the Kubota Z482 engine. However, modifications to the camshaft, rocker shaft, and timing drive were required for the split-cycle conversion. Both the intake and exhaust valve have the same 25 mm OD, and special valve seat inserts made by Dura-Bond®, part number: 71169, were installed to accommodate the low lubricity of methane. This section focuses primarily on the physical modifications made to the camshaft and rocker shafts. The timing and duration of the valves was determined using a 1-D engine model that will be covered in Chapter 7.

![Figure 3.35: Split-cycle pushrod valvetrain for used for intake and exhaust valves.](image)
3.6.1 Rocker Arm Shaft Modification

The Kubota engine has a total of four valves: one intake and one exhaust valve per cylinder. In its original form, all four rocker arms are mounted on a single shaft, known as the rocker shaft, and the shaft is fastened to the cylinder head in special mounting blocks. Since the split-cycle engine only requires two of the four valves, and therefore only two rocker arms, the rocker shaft was divided into two parts and mounted into saddles machined directly into the upper cylinder head. This provided more space in the vicinity of the combustion chamber, allowing for a more optimal placement of the spark plug and pressure transducers. The axial position of the rocker arms is maintained using retaining rings locked into grooves that were machined into the rocker shaft. Oil passages drilled in the upper cylinder head provide pressurized lubrication to the rocker shaft’s internal bore, which in turn lubricates the bearing surface of the rocker arm. Figure 3.36 shows a comparison of the original Kubota rocker arm configuration (left) and the modified split-cycle layout (right). Dimensional drawings of the rocker shafts can be found in Appendix A.

![Figure 3.36](image3.36.png)

Figure 3.36: Photograph of original Kubota rocker shaft layout (left), compared with a CAD rendering of the split-cycle modified configuration (right).

3.6.2 Camshaft Modifications

The Kubota camshaft is comprised of four lobes and three bearing journals. The two middle lobes on the stock camshaft were no longer required and were removed. Doing so allowed for the shaft to be cut into two parts and a 3 degree locking taper was fabricated between the second and third journal bearings. The taper joint allows infinite phase adjustment of the intake and exhaust lobes, while maintaining axial alignment of the journal bearings. The two halves of the taper are pulled together using a single bolt, threaded down the centre of the cam in the axial direction. To maintain oil flow to the rear journal bearing, the
centre of the bolt was drilled out. A cross-sectional view of the camshaft taper is shown in Figure 3.37. Disassembly of the camshaft is accomplished by loosening the bolt 2–3 turns and tapping the bolt head with a brass hammer to jar the taper loose.

![Figure 3.37](image-url)

**Figure 3.37:** Cross-sectional CAD rendering of locking taper mechanism machined into camshaft, providing phase adjustment for the intake and exhaust valve opening periods.

In stock form, the Kubota camshaft is driven internally by an oil-lubricated gear train. Since the split-cycle engine operates in a two-stroke manner, the crank-to-cam drive ratio had to be changed from 2:1 to 1:1. It was decidedly easiest to dismantle the internal gear drive and replace the camshaft gear with a shaft extension, thereby allowing the pushrod camshaft to be driven by the same external belt driving the OHC (see Section 3.7). Figure 3.38 shows a photograph of the camshaft in its original and modified forms. The change in drive ratio also required new cam lobes to be developed, which will be covered in the next section.

![Figure 3.38](image-url)

**Figure 3.38:** Pushrod camshaft: Kubota Z482 (*bottom*); modified for split-cycle (*top*).
3.6.3 Camshaft Lobe Profiles

The faster camshaft speed of the split-cycle engine meant new lobe profiles had to be created. This involved determining the required lift/duration characteristics of each valve, designing new lobe profiles, and then machining these profiles into the camshaft. Since the camshaft speed was increased by a factor of two, the new cam lobes essentially had to be twice as large in duration. This was accomplished by building up the lobes via welding and then regrinding to the correct shape; the work for which was performed by Colt Cams in Aldergrove, B.C. The lobe profiles were developed and optimized using a combination of different software products, which will be discussed here briefly and in greater detail in Chapter 7.

Lift Height Determination

The selection of maximum valve lift was based on the available flow area of the port, which is a function of the lift height. The minimum flow cross-section is known as the throat area, $A_t$, and up to a certain lift height corresponds to the aperture opening of the valve. Beyond this height, the port area becomes the limiting factor. The calculation of a valve’s aperture area, known as the curtain area, $A_c$, is also dependent on the lift height. For low lift values, the flow area is perpendicular to the valve seat and can be calculated as the side surface of a cone frustum. This is given in Equation (3.11), where $\phi$ is the seat angle and $d_{is}$ and $d_{os}$ are the inner and outer seats diameters, respectively.

$$A_c = \pi L \cos \phi (d_{is} + L \sin \phi \cos \phi)$$

(3.11)

Beyond a certain lift height, denoted as $L_{lim}$, the conical flow surface at the valve is no longer perpendicular to the valve seat face. This height limit can be calculated from:

$$L_{lim} = \frac{d_{os} - d_{is}}{\sin 2\phi}$$

(3.12)

For lift heights greater than $L_{lim}$, the curtain area is then [18]:

$$A_c = \pi \left( \frac{d_{os} + d_{is}}{2} \right) \sqrt{\left( L - \frac{d_{os} - d_{is}}{2} \tan \phi \right)^2 + \left( \frac{d_{os} - d_{is}}{2} \right)^2}$$

(3.13)

Table 3.6 shows the dimensions, flow areas, and $L_{lim}$ value calculated for the intake and exhaust ports of the split-cycle engine. Both ports share the same valve seat insert and therefore have the same minimum flow area.
Table 3.6: Intake and exhaust port parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner Seat Diameter</td>
<td>(d_{is})</td>
<td>22.5</td>
<td>mm</td>
</tr>
<tr>
<td>Outer Seat Diameter</td>
<td>(d_{os})</td>
<td>26</td>
<td>mm</td>
</tr>
<tr>
<td>Seat Angle</td>
<td>(\phi)</td>
<td>45</td>
<td>deg</td>
</tr>
<tr>
<td>Transition Height</td>
<td>(L_{lim})</td>
<td>3.915</td>
<td>mm</td>
</tr>
<tr>
<td>Port Area</td>
<td>(A_p)</td>
<td>346.4</td>
<td>mm(^2)</td>
</tr>
</tbody>
</table>

Throat Area: \(A_t\)  
- \(A_t = A_c\) for \(A_c < A_p\) 
- \(A_t = A_p\) for \(A_c > A_p\)

Figure 3.39 graphically depicts the curtain flow area calculated from Equations (3.11) and (3.13), in comparison with the port area from Table 3.6. It is shown that lift values above 6 mm offer no increase in flow area. However, a higher lift translates into a greater valve velocity, which results in a quicker opening of the valve, and thus the maximum throat area is achieved for a longer duration. Therefore, 6 mm was constrained as the minimum peak lift value but higher values were also investigated.

![Figure 3.39](image)

Figure 3.39: Intake / exhaust port flow area as a function of lift.

Intake and Exhaust Lobe Profiles

Development of the intake and exhaust lobe profiles was done with the aid of AVL BOOST, a 1-D numerical engine modelling software. The model was coupled with an optimization
software, called modeFRONTIER®, that allowed for a design of experiments (DOE) to be used in the search for optimal intake and exhaust valve timing. Details of both software models and the DOE will be presented in Chapter 7.

Within the numerical model, generic intake and exhaust valve lift curves from the program library were initially used. The opening and closing points of these curves were then manipulated by the optimization software, with the goal of maximizing the net indicated mean effective pressure (IMEP) of the engine. For each timing shift, the valve lift profiles were automatically regenerated in BOOST by maintaining a similar peak valve velocity. Thus, the lift height was allowed to vary but restricted to be no less than 6 mm. The crossover valves, which were already designed at this point, were also included in the model. The optimizer was allowed to vary the phase of the crossover valves, but the duration was fixed. The valve timings that resulted from the optimization trials are given in Table 3.7. The values include ramp durations and are referenced to their respective cylinder TDC.

Table 3.7: Preliminary valve timing based on a numeric engine simulation.

<table>
<thead>
<tr>
<th>Valve</th>
<th>Opening Crank Angle</th>
<th>TDC† Reference</th>
<th>Closing Crank Angle</th>
<th>TDC† Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake</td>
<td>20° ATDC</td>
<td>13.5° ABDC</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Crossover Inlet</td>
<td>42° BTDC</td>
<td>6.5° BTDC</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Crossover Outlet</td>
<td>7.5° BTDC</td>
<td>28° ATDC</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Exhaust</td>
<td>8° BBDC</td>
<td>11° BTDC</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

† TDC refers to the cylinder in which the valve being considered resides.

Based on the durations in Table 3.7 and a target lift value of 6-7 mm, intake and exhaust cam lobe profiles were developed using Lotus Concept Valvetrain software. The lift profiles are shown in Figure 3.40 and the corresponding specifications are given in Table 3.8.

Table 3.8: Intake/exhaust valve lift specifications.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Intake</th>
<th>Exhaust</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lift Height</td>
<td>6.65</td>
<td>6.65</td>
<td>mm</td>
</tr>
<tr>
<td>Ramp Height</td>
<td>0.30</td>
<td>0.35</td>
<td>mm</td>
</tr>
<tr>
<td>Ramp Velocity</td>
<td>0.017</td>
<td>0.020</td>
<td>mm/deg</td>
</tr>
<tr>
<td>Lift Duration†</td>
<td>173</td>
<td>177</td>
<td>°CA</td>
</tr>
<tr>
<td>Symmetric Profile</td>
<td>Yes</td>
<td>Yes</td>
<td>-</td>
</tr>
</tbody>
</table>

† Duration between ramps.
3.7 Timing Belt Drive

Both camshafts are driven by the crankshaft using a synchronous timing belt, located externally on the front of the engine. The belt is a high torque drive (HTD) series manufactured by Gates® Corporation, part number: 1000-5M-15, and is made of synthetic rubber with Kevlar impregnated cords. The synchronous belt was used because it requires no lubrication, minimal tension, and will not slip—a situation that could cause severe engine damage. Figure 3.41 shows a schematic of the timing belt layout, including the degree of belt wrap on each pulley. A minimum belt wrap of 60° was recommended by the belt manufacturer.

The eccentric idler and belt tensioner pulleys serve two purposes:

1. to route the timing belt around various obstacles like the intake runner and main coolant line

2. to provide a fine level of adjustment to the timing drive after the engine has been assembled

Because the belt and pulleys have a finite pitch, the span length between pulleys is unlikely to precisely match that required for accurate valve timing, at least without significant belt compliance. Instead of readjusting the cam lobes after the belt was installed, it was easier to use the idler pulleys as a means of altering the span lengths. A close-up schematic of how the tensioner adjustments work is shown in Figure 3.42.
Figure 3.41: Schematic of synchronous timing belt layout. Coordinates in mm.

Figure 3.42: Photograph of timing belt adjustment mechanisms.
Front Cover Modification

The Kubota engine’s front cover, which houses the geared timing drive and oil pump, was retained for use with the split-cycle engine. The cover was modified to incorporate a support bearing and seal for the extended pushrod camshaft, and the integrated water pump casting was removed and replaced with a 3/4 inch national pipe thread fitting for the cooling water inlet. The water pump was replaced with an external unit (see Section 4.2.1). Figure 3.43 shows a photograph of the original and modified covers side-by-side.

![Figure 3.43: Engine front cover: OE Kubota unit (left), modified for split-cycle (right).](image-url)
Chapter 4

Experimental Setup

Chapter 3 described the design and function of the split-cycle engine and its various internal components. This chapter focuses on the design and construction of a test platform used to operate, control, and measure the performance of the split-cycle engine. Ultimately, the purpose of the test platform is to acquire information regarding the combustion, emissions, and thermodynamic properties of the engine under a variety of different test conditions, such as rotational engine speed, spark timing, air-fuel ratio, throttle position, etc. The physical phenomena are acquired through an assortment of transducers, primarily pressure- and temperature-based, which relay the information to a data acquisition (DAQ) system where it is processed and digitally stored. Programming interfaces written in LabVIEW™ provide the DAQ logistics, while simultaneously giving the user full control of the engine operating parameters. A detailed overview of the test hardware, instrumentation, and programming will be covered in this chapter.

Much of the test apparatus is situated on a steel t-slot plate, measuring approximately 1.59 m x 2.13 m x 0.15 m and weighing an estimated 1500–2000 kg. The plate’s mass serves to dampen the engine vibrations, and prevents the test platform from moving during operation. Control of the engine speed and/or load is accomplished using an electric dynamometer, which is fastened to the t-slot plate and connected to the engine through a synchronous belt drive. Additional engine support systems, such as the air intake, exhaust, lubrication, cooling, and electrical power distribution are also incorporated on the test platform and will be discussed in detail within this chapter. Figure 4.1 provides a basic schematic of the fluid flow and power transmission within the test apparatus. It also shows some of the measurement locations for pressure, temperature, brake torque, and air-fuel ratio. A photograph of the experimental setup is shown in Figure 4.2.
**Figure 4.1:** Plan view schematic of engine dynamometer test bed.
Figure 4.2: Photograph of experimental setup.
4.1 Engine Dynamometer

4.1.1 Speed and Load Control

The test bed is centred around a 15 HP alternating current (AC) motor that functions as a dynamometer. The motor and its associated control unit are capable of either driving the engine (generally referred to as engine motoring) or absorbing power from the engine and dissipating it as heat through a resistor bank. Two operational modes of power absorption exist: constant torque with varying velocity, or constant velocity with varying torque. In this work, all experiments have been carried out under conditions of constant angular velocity, since the engine’s control strategy has not yet been designed to handle transient conditions. The dynamometer drive speed is adjusted through LabVIEW™, using a 0–5 VDC analog output channel.

Engine brake torque is measured with a Lebow in-line rotary torque transducer, model 1605-5K, located in the path of power transmission between the electric motor and the engine (see Figure 4.1). The shafts of the transducer are mated to the drive assembly through a pair of Morse Moreflex® flexible couplers, allowing for slight shaft misalignment and providing isolation and damping from load changes. The rotary torque transducer is an AC device and requires the use of an AC strain gauge conditioner with an excitation frequency of 3.28 kHz at 2.77 VAC. For this purpose, a Daytronic model 5M78 conditioner was selected, which provides the necessary input excitation and has a linear 0–10 VDC output. The conditioner is located in the data acquisition cabinet discussed in Section 4.4. A two point (zero and span) dead weight calibration was performed regularly on the torque measurement hardware. Information on the calibration procedure can be found in Appendix F.

4.1.2 Engine Synchronous Belt Drive

The engine is connected to the dynamometer drive-line using a synchronous belt and pulley system. A direct shaft-coupled system would have been the preferred method of connection, however, the combined in-line length of the engine and dynamometer exceed the length of the t-slot plate. Therefore, the engine and dynamometer are located side-by-side and connected through a Gates GT2 synchronous belt, part number: 1600-8MGT-50. The belt and associated pulleys were designed for the maximum rated horsepower of the dynamometer, including a service factor\(^1\) of 3.0 for use with an internal combustion engine. At an engine speed of 1000 RPM the belt is rated for approximately 40 HP.

\(^1\) The service factor accounts for unsteady loading on the belt drive [59].
On the dynamometer side, the drive belt pulley is keyed to a ground and hardened shaft, which is supported by a pair of pillow block bearings. Two separate aluminum riser blocks, one for the torque transducer and one for the pillow block bearings, were manufactured to provide the correct centreline height of each respective component. The riser blocks and AC motor are all dowelled and fastened to a 25.4 mm thick steel plate, which can be moved relative to the t-slot plate in order to adjust belt tension without disturbing the axial alignment of the motor/transducer/pulley assembly. The pillow blocks were also machined so they fit into the riser block, allowing the bearings to be removed for belt installation and subsequently re-installed without affecting the shaft alignment. Figure 4.3 is an exploded view of the dynamometer components mounted to the plate.

On the engine side, the belt pulley is mounted on a fabricated stub-shaft that is fastened to the engine’s crankshaft flange, where the Kubota flywheel would normally be located. To prevent the belt load from being cantilevered on the stub-shaft, which would consequently increase the load on the rear main bearing of the crankshaft, a support for the stub-shaft was fabricated and attached to the engine block in place of the Kubota flywheel housing. The support is a steel weldment that uses a ball bearing (RBI part number: 1641-2RS, rated for a 13.2 kN load up to 5000 RPM) to support the radial shaft load of the dynamometer belt. Precise alignment of the support bearing with the crankshaft axis was achieved by attaching the weldment to the engine block and aligning the mill spindle to the crankshaft axis before machining the bore of the support bearing. The left side of Figure 4.4 shows the engine block and bearing weldment being machined as an assembly; dowel pins between the two components were used to provide a locating reference for disassembly. The final installation of the bearing support structure on the engine is shown on the right side of Figure 4.4. The hexagonal drive machined into the end of the stub-shaft, protruding from the support bearing, serves a dual purpose: a means of rotating the engine by hand with a 19 mm wrench, and a positive drive engagement for the engine’s flywheel. An exploded view of the stub-shaft and bearing support assembly, along with the corresponding bill of materials, can be found in Appendix A.

Generally, synchronous belts require very minimal pre-tensioning, as they are not friction-driven like a V-belt. However, to prevent harmonic resonance of the belt, an adjustable idler pulley was installed at the belt’s midspan. The tensioner has an aluminum roller mounted on a pair of needle bearings (Koyo part number: HJ-283716) that roll around an eccentrically mounted shaft, which can be rotated using a spanner wrench to adjust the belt tension. The base of the tensioner is rigidly bolted to the t-slot plate. A break down of the tensioner components can be found in Appendix A.
Chapter 4: Experimental Setup

Figure 4.3: Exploded view of the engine dynamometer drive-line.
4.1.3 Flywheel

The stock Kubota flywheel had to be removed in order to mount the belt drive stub-shaft support to the engine block. Therefore, a new flywheel was machined and attached on the end of the stub-shaft by means of an interference fit. The steel flywheel, photographed in Figure 4.5, has a diameter of 101.6 mm and a moment of inertia of 38193 kg mm$^2$, which is approximately 15% greater than the OE flywheel. Three bolt holes on the face of the flywheel are for attaching a protractor wheel that was used in the process of locating cylinder TDC and also for configuring the camshaft timing.

Figure 4.4: Bearing support for dynamometer drive belt pulley: attached to engine block for concentric machining (left); finished installation on engine (right).

Figure 4.5: Photograph of the engine flywheel mounted on the belt drive stub-shaft.
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4.2 Mechanical Engine Subsystems

4.2.1 Engine Cooling System

Engine temperature is regulated using a pressurized, closed-loop cooling system filled with a 50/50 mixture of ethylene glycol and water. The system consists of an electric water pump, water-to-air heat exchanger, mechanical thermostat, electronic fan control, and numerous temperature sensors.

The engine coolant is circulated using a 12 VDC electric water pump manufactured by Meziere Enterprises, part number: WP136S, that, according to the manufacturer, is capable of flowing up to 3.8 L/min under no-load conditions. The pump is mounted on a pedestal external to the engine and operated remotely from the dynamometer control room. Coolant leaving the pump is directed into the front of the engine block, where it flows around the cylinders and then upwards into the cylinder head through holes in the deck surface. The primary outlet is located at the front of the lower cylinder head, and a secondary outlet at the rear can be manually adjusted using a ball valve to bleed-off coolant before it has a chance to fully circulate within the head. This provision was put in place to prevent overheating under high-load conditions. Coolant leaving the cylinder head is directed to the thermostat, which regulates flow through the water-to-air heat exchanger. A heat exchanger bypass line provides minimal fluid circulation through the engine when the thermostat valve is closed, to keep from dead-heading the pump, and also to prevent hot spots from forming within the engine’s cooling jackets. A schematic of the cooling system is shown in Figure 4.6, including the locations of thermocouples used to monitor coolant temperature. Further details of temperature related measurements can be found in Section 4.4.4.

The stock Kubota Z482 thermostat was retained, but the thermostat and its housing were relocated from the cylinder head to an external mounting block, as shown in Figures 4.7 and 4.8. The thermostat actuates mechanically based on the fluid temperature, beginning to open at approximately 70°C and reaching a fully open state at 85°C [88]. The thermostat was located in close proximity to the engine block in order to minimize the temperature differential between the two. Coolant temperatures measured at three different locations inside the cylinder head have shown that the temperature is easily controlled between 70–80°C and quite uniform throughout (within ±2–3°C).

An aluminum water-to-air heat exchanger (HEX) combined with an electric fan reduces the temperature of the coolant before it is returned to the engine. A thermocouple mounted at the outlet of the heat exchanger provides a feedback signal to command the on/off operation
Chapter 4: Experimental Setup

Figure 4.6: Schematic of engine cooling system.

Figure 4.7: Left: Kubota Z482 cooling thermostat in stock location. Right: Thermostat moved to external mounting block to accommodate new cylinder head.
of the fan based on the desired temperature of the coolant returning to the engine. The on/off temperature limits are user-defined in LabVIEW™, which allows a finite amount of control over the engine’s operating temperature.

### 4.2.2 Engine Oiling System

The internal engine components are lubricated by two separate pressure-fed oil systems that share the same lubricating oil. All of the components located inside the engine block (e.g. crankshaft, connecting rods, etc.) utilize the OE Kubota pump, which is gear-driven by the crankshaft. The cylinder head components are oiled by an external system that is driven by an electric motor. Internally, all of the oil drains back into the engine’s wet sump, where a portion is then routed to an external reservoir for the secondary system.

The external oiling system was installed for two reasons: 1) to maintain good oil pressure with addition of an entirely new camtrain, and 2) to enable the camtrain to be primed with oil before operation. A picture of the secondary oiling system is provided in Figure 4.9, and shows, from left-to-right, the 4L aluminum oil reservoir, followed by the pump/filter mounting block, and finally the 1/3 HP Baldor® electric motor. The motor and pump
Figure 4.9: Photograph of external oiling system used for cylinder head valvetrain components. Belt guard removed for photo.

are connected through a synchronous belt drive, with a motor-to-pump pulley ratio of 2:3 resulting in a pump speed of 1150 RPM. The pump is the same one used inside the engine, only it has been modified to include an o-ring on the gerotor shaft to prevent leakage. It is mounted to a custom-made aluminum block, which also houses a filter and an over-pressure bypass mechanism. The oil pressure is adjusted manually with a ball valve that controls the amount of flow recirculating to the reservoir. It was set to 3 bar for all experiments in the present work. The pressurized oil flows into a distribution rail (not shown), where it is then routed into multiple lines that feed individual oil galleries in the cylinder head.

Valvoline Premium Blue GEO 15W40 engine oil was selected as the lubricating oil for the split-cycle engine. The oil is specially formulated for use in natural gas engines and, according to the manufacturer, will produce lower quantities of ash compared to conventional engine oil. Additionally, it has enhanced valve recession protection, which is known to be an issue with low lubricity fuels such as methane [144].

Due to the relatively high contact forces present in the reverse poppet camtrain, it was decidedly important not to rely solely on splash lubrication for the cam lobes. Therefore, oil squirtsers were installed directly overhead of the camshaft lobes (in the valve cover) to provide a steady stream of oil onto the cam lobe surface. This ensures a thin oil film between the lobe and rocker arm is present at all times. A summary of the engine oiling system specifications are given in Table 4.1.
Table 4.1: Engine oiling system specifications.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Oil Pump</td>
<td>Kubota Z482</td>
</tr>
<tr>
<td>Oil Filter</td>
<td>Kubota Z482</td>
</tr>
<tr>
<td>Total Oil Capacity</td>
<td>6 L</td>
</tr>
<tr>
<td>Primary pump speed</td>
<td>Varies w/ engine RPM</td>
</tr>
<tr>
<td>Primary pump pressure</td>
<td>1.5–4 bar</td>
</tr>
<tr>
<td>Secondary pump speed</td>
<td>1150 RPM</td>
</tr>
<tr>
<td>Secondary pump pressure</td>
<td>3 bar</td>
</tr>
<tr>
<td>Oil Type</td>
<td>Valvoline GEO 15W40</td>
</tr>
</tbody>
</table>

4.2.3 Air Intake System

The air intake system for the engine consists of a filtering element, flow measurement device, surge tank, electronically controlled throttle, and an assembly of seamless, welded stainless steel piping. A drawing of the system layout is shown in Figure 4.10.

Intake Air Flow Measurement

The quality and quantity of the air entering the engine is crucial for determining parameters such as the air/fuel ratio, EGR rate, and volumetric efficiency. A Meriam Z50MC2-2F laminar flow element (LFE) connected to a Dywer Instruments differential pressure transmitter, both shown in Figure 4.11, were installed to measure the volume of air entering the engine. The LFE has a factory installed filter element on top, and comes calibrated as a complete unit with an overall uncertainty of ±0.54% in volumetric flow rate at a 95% confidence level. The combined uncertainty of the entire measuring chain is given in Section 4.6. A 159 L galvanized steel tank, located between the LFE and the engine, was used to dampen reverberating pressure waves and steady the air flow measurement.

The operational principle of the LFE is based on the exact solution to the Navier-Stokes equations for steady, incompressible, laminar flow through a straight circular pipe, known as Hagen-Poiseuille flow [102]. The LFE is designed to induce laminar flow through a set of identical and parallel capillary tubes, across which the pressure drop is measured. The linear relationship between the pressure drop, $\Delta p$, and the volumetric flow rate, $\dot{Q}$, is given by Equation (4.1), where $R$ is the capillary tube radius, $\mu$ is the dynamic viscosity of the fluid, and $l$ is the capillary tube length.

$$\dot{Q} = \frac{\pi R^4 \Delta p}{8 \mu l}$$ (4.1)
Chapter 4: Experimental Setup

Figure 4.10: Layout drawing of engine air intake system. Drawn to scale.

Figure 4.11: Photograph of laminar flow element connected to the differential pressure transmitter. The intake air temperature thermocouple is also visible.
The values of $R$ and $l$ are known, and the Dwyer pressure transmitter measures $\Delta p$ across the capillaries. Thus, the only remaining unknown is the dynamic viscosity term. Since the engine aspirates air directly from the test cell, the air properties can vary substantially from the factory calibration conditions of 21.1°C, 101.3 kPa, and 31.3% relative humidity (RH) or dry air. Within this section, these values will be hereafter referred to as the standard conditions and denoted with a naught subscript. The calibration curve provided by the manufacturer of the LFE gives $\dot{Q}$ based on $\mu_0 = 1.8187 \times 10^{-5}$ kg/m·s (for the standard conditions listed above). Therefore, the volume flow rate must be corrected for the changes in $\mu$ associated with differing temperature and RH. At a given temperature, $T$, the dry-gas dynamic viscosity, $\mu_{\text{dry}}$, can be calculated from:

$$\mu_{\text{dry}}(T) = \mu_0 \left( \frac{T}{T_0} \right)^{3/2} \frac{T_0 + S}{T + S}$$

(4.2)

Where $\mu_0$ and $T_0$ are the dynamic viscosity and temperature at the standard conditions, respectively, and $S = 110.4$ K.

The difference between dry-air and wet-air viscosity changes with both temperature and humidity. At the beginning of each engine test, the ambient conditions were used to determine the ratio of $\mu_{\text{wet}}/\mu_{\text{dry}}$ from Figure A-35500 in the LFE operators manual [100]. Since the ratio falls between 0.995 and 1 for all realistic test conditions, any changes that occurred during a single test were considered negligible. The actual volumetric flow rate, $\dot{Q}_a$, can then be calculated from:

$$\dot{Q}_a = \dot{Q} \left( \frac{\mu_0}{\mu_{\text{dry}}} \right) \left( \frac{\mu_{\text{dry}}}{\mu_{\text{wet}}} \right)$$

(4.3)

By applying correction factors for the ambient temperature, $T$, and pressure, $p$, of the test cell, the volumetric flow rate can be standardized to the calibration conditions:

$$\dot{Q}_s = \dot{Q}_a \left( \frac{T_0}{T} \right) \left( \frac{p}{p_0} \right)$$

(4.4)

The mass air flow rate, $\dot{m}$, was then calculated by multiplying $\dot{Q}_s$ by the density at standard conditions, $\rho_0$, and a correction factor for the change in density with RH:

$$\dot{m} = \rho_0 \dot{Q}_s \left( \frac{\rho_{\text{wet}}}{\rho_{\text{dry}}} \right)$$

(4.5)

The density correction factor for humidity ($\rho_{\text{wet}}/\rho_{\text{dry}}$) was taken from Table A-35600 [100]. Similar to the $\mu$ ratio, its value can be expected to vary less than ±0.0005 over the course of a single test and therefore was considered a constant.
The mass air flow rate calculations were performed in real-time using the LabVIEW™ sub-VI shown in Figure 4.12. Inputs to the VI come from various physical measurement channels: the air temperature was measured downstream from the LFE (see Figure 4.10) using a 1/16 inch exposed-tip, K-type thermocouple; the humidity was measured with a model HX71 relative humidity transmitter mounted near the filter element on the LFE; and the ambient pressure was measured with an Omega PX319, 0–2 bar absolute pressure sensor located in the test cell.

![Figure 4.12: LabVIEW™ sub-VI used to correct the intake air measurement under varying air quality conditions.](image)

**Intake Air Throttle**

Spark ignition engines must operate around stoichiometric air-fuel conditions in order to maintain functionality of the three-way exhaust catalyst, which oxidises CO and UHCs during periods of lean operation, and reduces NO\textsubscript{x} under rich conditions. Since the air-fuel ratio must be precisely controlled under various engine operating conditions, such as changes in speed and load, the air and fuel both require metering devices. Much of the developed split-cycle engine fabrication was accomplished by using or modifying existing components from the Z482 Kubota engine. However, since the stock engine operates on diesel fuel (i.e. lean burn), no intake air throttling device is present; therefore, an electronic throttle body was developed to allow precise metering of the air entering the split-cycle engine.
The throttle body, shown in Figure 4.13, uses a standard butterfly valve located inside a stainless steel pipe. The valve is rotated by a Firgelli L12-30-100-12-I linear actuator, connected to the valve through a short radial linkage. A 0–5 VDC analog signal, commanded using LabVIEW™, controls the actuator position. The valve shaft rotates on bronze bushings, and both the shaft and housing incorporate o-ring seals to prevent air leakage, which would affect the mass flow measurement. On the opposite side from the linear actuator, a rotary position sensor is attached to the valve shaft and provides confirmation of the desired throttle position. An exploded diagram of the throttle components along with the corresponding bill of materials is provided in Appendix A.

Figure 4.13: Photograph of the intake air throttle body assembly. Throttle position is adjusted using an electronic linear actuator (shown in the foreground) that is controlled via LabVIEW™.

Figure 4.14 shows the measured throttle position (from the rotary sensor) versus the linear actuator command voltage. It can be seen that a linear relation exists for a command voltage ranging from approximately 1–4.5 VDC, but below 1 VDC and above 4.5 VDC linearity is not preserved. Data points for both sequential opening and closing are presented, and a hysteresis of approximately 1% is shown. Thus, for consistency, all throttle position references should be based on the rotary throttle position sensor, and not from the command voltage. It should also be mentioned that a 0% and a 100% throttle position correspond to a butterfly valve angle of 10° and 90°, respectively, where 90° means the valve is parallel with the intake pipe. The term wide open throttle (WOT) refers to this position, and was used exclusively for the results presented in this work.
Chapter 4: Experimental Setup

4.2.4 Exhaust System

Combustion products exiting the engine are routed through a piped exhaust system, shown schematically in Figure 4.15, before being discharged to the outside environment. All piping is Schedule 40, 304/304L stainless steel, starting from a 25.4 mm nominal diameter at the engine and increasing to 38.1 mm nominal diameter downstream. The exhaust pipe terminates at a 69.7 L expansion tank, which serves to eliminate pressure pulsations for steadier control of EGR to the intake system. A ball valve on the outlet of the tank is a provision for adjusting the EGR rate by varying the exhaust back pressure. The latter is measured by an Omega PX319, 0–2 bar absolute pressure transducer located on the side of the tank. An automotive style flex pipe between the engine and the expansion tank allows for relative movement between the two. Insulation was applied to all the exhaust piping to prevent condensation and minimize heat transfer to the test cell.

Exhaust temperatures reported in this document were measured by a 1/16 inch, K-type thermocouple located inside the exhaust pipe approximately 76 mm downstream from the exhaust port. Gas sampling for emissions is taken from the exhaust flow approximately 200 mm downstream from the exhaust port and is directed to the gas analysers through a flexible, heated, 3/8 inch stainless steel tube. Temperature controllers on the emissions bench maintain the heated sampling line at approximately 190°C to prevent condensation.

Also shown in Figure 4.15 is the location of the exhaust gas oxygen sensor (labelled \(O_2\) bung), which is located approximately 400 mm downstream from the exhaust port.
4.2.5 Fuel System

Fuel Specifications

All testing was conducted using bottled Grade 2.0 methane, which has a purity of 99%. The approximate composition of the gas is shown in Table 4.2.

Table 4.2: Composition of test fuel.

<table>
<thead>
<tr>
<th>Component</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Methane</td>
<td>99%</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>&lt;3000 ppm</td>
</tr>
<tr>
<td>Oxygen</td>
<td>&lt;750 ppm</td>
</tr>
<tr>
<td>Other Hydrocarbons</td>
<td>&lt;1.0%</td>
</tr>
<tr>
<td>Moisture</td>
<td>&lt;10 ppm</td>
</tr>
<tr>
<td>Total Impurities</td>
<td>&lt;1.0%</td>
</tr>
</tbody>
</table>

Delivery and Measurement

A schematic of the split-cycle engine fuel system is shown in Figure 4.16. Methane is supplied to the system from a gas bottle, which is initially pressurized to 138 bar. As the cylinder deletes, a two-stage gas regulator maintains a constant fuel line pressure at approximately 68 bar. The fuel bottle is replaced once its internal pressure reduces to the regulated value. Contaminants are removed by a 40 micron filter before the gas passes through a Sierra Smart-Trak M100L mass flow meter. A 500 cm$^3$ stainless steel surge tank is located between the fuel injector and mass flow meter. Its purpose is to dampen pressure waves caused by the intermittent injection pulses, which helps to stabilize the flow.
measurement. Two Omega® PX429 pressure transducers, capable of measuring pressures up to 70 bar, were installed on either side of the flow meter to monitor its pressure drop. Near the operating limit of the meter, it can become an intermittent flow restriction as the pressure regulator opens and closes over a finite range. The sudden change in differential pressure was used to detect these conditions, although this did not present as an issue during engine testing. The majority of the fuel line is 1/4 inch rigid 304 stainless steel tubing, with a 0.035 inch wall thickness. To allow for engine vibration, the rigid tubing transitions to a flexible, steel over-braided nylon tube in the vicinity of the fuel injector. Fuel temperature measurements via thermocouple were taken on the outlet side of the surge tank to ensure consistency between tests.

![Figure 4.16: Schematic of fuel delivery system.](image)

Measurement of the flow rate is based on the first law of thermodynamics for controlled heat transfer to a working fluid. As the gas enters the flow meter, the bulk of the fluid passes through a laminar flow element, which creates a small pressure drop in the flow. This generates a steady flow of gas through a secondary passage that bypasses the laminar flow element and is heated by a pair of resistance temperature detector (RTD) coils. When heat from the first RTD coil is absorbed by the flowing gas, a temperature (and thus resistance) difference is generated between the two RTD outputs. This temperature differential, $\Delta T$ in Equation (4.6), is linearly proportional to the mass flow rate of the gas over the calibrated range of the instrument.

$$\dot{m} = \frac{\dot{Q}_{in} - \dot{Q}_{out}}{c_p \Delta T}$$  \hspace{1cm} (4.6)$$

The specific heat capacity of the gas $c_p$ and the net heat transfer $\dot{Q}_{in} - \dot{Q}_{out}$ are both constants, input by the manufacturer during instrument calibration. The micro-processor
of the Smart-Trak then generates a 4–20 mA output current proportional to the measured flow rate. The instrument has a maximum calibrated flow rate of 0.418 g/s of methane, with an accuracy of ±1.0% of full scale. Fuel dosing is controlled by the pulse width of the injector, and is discussed in detail in Section 4.4.7.

4.3 Emissions Analysis

The exhaust gas concentrations of NO\textsubscript{x}, O\textsubscript{2}, THC, CH\textsubscript{4}, CO, and CO\textsubscript{2} were measured using an emissions bench built and donated by Ford Motor Company. The bench uses a diaphragm (vacuum) pump to draw a continuous sample of the engine's exhaust gases through a stainless steel sampling line connected to the engine's exhaust pipe (see Figure 4.15 for location). Before reaching the pump, the sample gases are passed through a 4 micron filter to remove unwanted particulates. The sampling line, filter housing, and pump are all heated to 190°C in order to prevent the water vapour in the exhaust gases from condensing. After the pump, the flow is split into two streams: one that goes straight to the HC analyser, and the remainder, which passes through a chiller, coalescing filter, and desiccant dryer to completely remove any water vapour. Thus, it can be said that the HCs are measured \textit{wet}, and the remaining gases are measured \textit{dry}. The THCs were measured on a C\textsubscript{1} basis, and segregation of CH\textsubscript{4} and NMHCs was performed periodically during testing but not continuously recorded like THC. Table 4.3 lists the measuring principle and span gas concentration used for each gas constituent. The bench consists of three separate analysing devices, each made by California Analytical Instruments Inc; their part numbers are also provided in the table.

Table 4.3: Details of emissions analysis system.

<table>
<thead>
<tr>
<th>Species</th>
<th>Span</th>
<th>Range</th>
<th>Measurement Principle</th>
<th>Analyser</th>
</tr>
</thead>
<tbody>
<tr>
<td>O\textsubscript{2}</td>
<td>25 %</td>
<td>25 %</td>
<td>Paramagnetism</td>
<td>CAI 602P-NDIR</td>
</tr>
<tr>
<td>CO</td>
<td>0.5 %</td>
<td>0.5 %</td>
<td>NDIR</td>
<td>CAI 600-HFID</td>
</tr>
<tr>
<td>CO\textsubscript{2}</td>
<td>8 %</td>
<td>15 %</td>
<td>NDIR</td>
<td></td>
</tr>
<tr>
<td>THC/CH\textsubscript{4}</td>
<td>3000 ppm</td>
<td>3000 ppm</td>
<td>HFID</td>
<td>CAI 600-HFID</td>
</tr>
<tr>
<td>NO\textsubscript{x}</td>
<td>1000 ppm</td>
<td>1000 ppm</td>
<td>Chemiluminescence</td>
<td>CAI 605-HCLD</td>
</tr>
</tbody>
</table>

ppm = parts per million
NDIR = non-dispersive infrared
HFID = heated flame ionization detector
4.4 Data Acquisition and Control System Hardware

4.4.1 General

The data acquisition (DAQ) system refers to the chain of measurement from sensor to personal computer (PC). The interface that allows communication between these two forms of hardware is the DAQ device, which conditions and converts the physical signals into digital measurements. Several DAQ devices were used in this work due to the large number of required channels and relatively high data rates. Three PCs were employed to control these devices and provide a means of processing, visualizing, and storing the data.

In addition to acquiring data, electronic control of the engine and dynamometer was also required. Low-level devices such as dynamometer speed, throttle position, and cooling fan operation were controlled with standard analog output (AO) hardware. For engine control (i.e. fuel injection and ignition timing), the fast data rates combined with the requirement for a highly deterministic system, led the author to use a stand-alone hardware platform that utilizes its own processor and real-time operating system. The aforementioned hardware is a CompactRIO system made by National Instruments™ (NI).

A custom instrumentation and electronics cabinet was constructed by the author to house several of the DAQ devices, along with the associated wiring and hardware. The cabinet is shown in Figure 4.17 and a summary of the components is provided afterwards.

![Figure 4.17: Photographs of the DAQ and electronic control cabinet.](image-url)
1) Horiba Mexa-730 air/fuel ratio analyser

2) 12 VDC power supply for Horiba analyser and heated O$_2$ sensor

3) Kistler piezoresistive amplifier type 4618A0 for intake port pressure sensor

4) BNC breakout box for in-cylinder pressure transducers

5) NI-9213 thermocouple DAQ device

6) NI-USB-6356, X-series multi-function DAQ device

7) 9074 CompactRIO controller with 8-slot I/O chassis

8) AC strain gauge conditioner, model 5M78, for brake torque measurements

9) Linear DC power supplies: 5/12/24 VDC

10) RMSS2 TTL digital signal splitter for crank angle encoder

11) Resistor bank for converting 4–20 mA signals into voltage signals

12) USB-6210 multifunction DAQ device (not shown, mounted to side of cabinet)

**CompactRIO (cRIO) System**

The cRIO system contains an on-board 400 MHz real-time processor, capable of operating in a stand-alone manner. This was the primary reason for selecting this platform, since it mitigates the use of a PC-based operating system (e.g. Microsoft Windows®), which could introduce latency or jitter into the control. The 9074 cRIO chassis has 8 slots for I/O modules. Communication and control of these modules is accomplished through a 2 M gate FPGA$^2$, which is configured using LabVIEW™. The FPGA circuitry has an on-board clock rate of 40 MHz allowing digital logic to be accomplished with 25 ns resolution. Data communication from the cRIO to a PC was accomplished through a 10/100 Ethernet connection. The following input/output (I/O) modules were installed in the cRIO:

- NI 9401: 8-channel, 5V/TTL digital I/O module used for measurement of crank angle position and control of engine ignition timing.
- NI 9751: 3-channel, direct injector driver module used for fuel injection control.
- NI 9215: 4-channel, ±10 V, 16-bit simultaneous analog input (AI) module used for fuel line and exhaust pressure measurements.

$^2$ A Field Programmable Gate Array (FPGA) is a reprogrammable silicon chip that uses configurable logic blocks to create a circuit.
Chapter 4: Experimental Setup

USB-6210 Device

The NI USB-6210 DAQ device has 8 differential analog inputs (AIs), an aggregate sampling rate of 250 kS/s, and a 16-bit resolution. It was used to acquire the exhaust gas emissions measurements, which were output from the analysers through 0–5 VDC signals.

USB-6356, X-Series Device

The NI USB-6356 DAQ device was used for all the remaining analog I/O not covered under the cRIO or USB-6210 devices. The 6356 is a high-speed, simultaneous sampling, multifunction DAQ device that can handle 8 differential AIs at 1.25 MS/s/channel with 16-bit resolution. It was primarily selected for this reason, allowing the engine pressures to be simultaneously sampled at very high rates (e.g. 10 S/°CA). The 6356 also has digital input capability, which was used to acquire the crank angle encoder signal and trigger the AIs based on engine position (see Section 4.5.1). Analog outputs were used to control the dynamometer speed, and the throttle position of the engine.

Current-Based Measurements

The majority of the transducers used in the test platform generate a 0–5 VDC or 0–10 VDC output signal linearly representing a range of physical phenomena being measured. However, when available, a preferred 4–20 mA analog signal was used instead to reduce the signal’s susceptibility to electrical noise and eliminate voltage reduction errors caused by resistance in the cable leads. Since all the DAQ devices are configured to accept voltage inputs, the voltage drop across a 250 ohm resistor was used to convert the mA signals to 1–5 VDC outputs. The resistors have an uncertainty of ±0.01% in their nominal resistance and this translates into additional mechanical-unit uncertainties that are shown in Table 4.4. The uncertainty is a function of the input current and therefore only full-scale values are given. The complete design-stage uncertainty for each measurement is given in Section 4.6.

<table>
<thead>
<tr>
<th>Measurement</th>
<th>F.S. Range</th>
<th>Max. Uncertainty†</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air Flow Rate</td>
<td>8.00</td>
<td>±5.12 × 10⁻³</td>
<td>SCFM</td>
</tr>
<tr>
<td>Fuel Flow Rate</td>
<td>0.418</td>
<td>±5.23 × 10⁻⁵</td>
<td>g/s</td>
</tr>
<tr>
<td>Exhaust Pressure</td>
<td>6.08</td>
<td>±6.25 × 10⁻⁴</td>
<td>bar</td>
</tr>
</tbody>
</table>

† Uncertainty at full scale output conditions.
4.4.2 Crank Angle Encoder

Most of the analysis that was performed on the cylinder pressure measurements requires that the instantaneous cylinder volume be known for each concurrent pressure sample. This can be determined by precisely measuring the rotated angle of the crankshaft relative to a datum (e.g. piston TDC) and then inferring the cylinder volume through known geometry.

To accomplish this task, an optical incremental encoder was connected to the snout of the crankshaft. The encoder is a model H25\(^3\), manufactured by BEI Sensors Inc., and was selected because it is capable of withstanding large vibrations (5–2000 Hz at 20 g, up to 50 g for 11 ms) and harsh environmental conditions (up to 70°C at 98 % relative humidity). The H25 encoder requires a 5–28 VDC excitation and provides three transistor-transistor logic (TTL) digital output channels. An aluminum bracket assembly was designed by the author to mount the encoder to the engine, as shown in Figure 4.18. The brackets have slotted mounting holes that allow for 3-axis adjustment of the encoder in order to properly align its shaft with the crankshaft, and are also rigid in order to prevent measurement errors. A flexible, zero-backlash, 3/8 inch ID shaft coupler, made by Gurley Precision Instruments, part number: SCA-06E, connects the encoder shaft to the crankshaft. The coupler allows for a 1° angular misalignment and a 0.2 mm parallel offset between the two shafts.

![Figure 4.18: CAD rendering of optical encoder mounted on the engine.](image)

\(^3\) The complete encoder model number is: XH25D-SS-900-ABZC-28V/V-SM18
Quadrature Encoding: 0.1°CA Sampling

An example of the output signals from the crankshaft encoder are depicted in Figure 4.19. Two counter channels, labelled A and B, each producing 900 pulses per revolution (PPR), are shown to be quarter-cycle offset. An index channel, Z, pulses once per revolution and is used to reset the count of channels A and B every 360° of crankshaft rotation. The Z-index is half-cycle gated with B-low, and it is important to note that the falling edge of channel B takes precedence over the rising Z-index in the data acquisition software. To improve the sampling resolution, both rising and falling edges of each digital line can be observed by the counter device; this is generically known as quadrature. Doing this allowed the line count resolution of the encoder (900 counts/rev/channel or 0.4°CA/sample) to be extended to 3600 counts/rev or 0.1°CA/sample. This also eliminated directional ambiguity, since logic was implemented to ensure every A pulse was followed by a B pulse and vice versa.

![Figure 4.19: Output signals from BEI optical crankshaft encoder.](image)

Signal Splitting

As previously mentioned, two separate DAQ devices (USB-6356 and cRIO) require the signal for crankshaft position. Because the cRIO uses crankshaft position to determine ignition timing, it is crucial the encoder signal is accurate and free from noise spikes. As a precaution, the encoder signal was duplicated using an optically-isolated TTL signal splitter; model RMSS2, made by NorthStar Technologies Inc. This enabled both systems to receive the encoder signal, while effectively decoupling the two DAQ devices and simultaneously protecting them from any voltage spikes that may be incurred from the encoder or its wiring.
Encoder-TDC Alignment

Proper alignment of piston TDC with the encoder index pulse is essential for performing accurate engine calculations. Davis and Patterson [41] have shown that a 1°CA error in TDC-encoder alignment yields a 4–5% error in calculations for net fuel conversion efficiency and mean effective pressure. As follows, three primary methods exist for aligning physical cylinder TDC with the Z-index of the encoder:

i) **Mechanical Alignment:** This method involves setting the engine accurately to the TDC position, typically through the use of a dial indicator on the piston, and then adjusting the encoder until its index pulse is triggered. An oscilloscope is often used to monitor the encoder output during this adjustment procedure. This is the least accurate of all methods because of the finite width of the index pulse and the difficulty associated with manually adjusting the encoder in minute increments.

ii) **Motoring Cylinder Pressure Trace:** By monitoring the pressure trace of a non-firing cylinder, TDC can be more accurately determined. Heat and mass loses during the compression stroke guarantee that the pressure must be equal or lower during the subsequent expansion stroke. Ideally, plotting the logarithmic pressure-volume curve of an engine should yield a sharp point at the location of TDC. However, heat and mass losses will result in the peak pressure actually occurring slightly before TDC. The location of this peak with respect to minimum cylinder volume has been coined the *thermodynamic loss angle* [121], but it can only be accurately determined once the encoder has been properly phased. This technique is limited by the measuring chain fidelity as well as operator interpretation. Regardless, the split-cycle engine configuration renders this method invalid since the crossover valves are open when the piston is in the vicinity of TDC and, consequently, mass transfer is occurring.

iii) **TDC Sensor:** The final and most accurate method to properly phase encoder TDC is by using a dedicated TDC sensor, which measures the capacitance at the end of a probe placed on the surface of the combustion chamber. Capacitance increases as the piston approaches the cylinder head and reaches a maximum at TDC. A programmatic correction can then be applied to the encoder signal within the DAQ software, eliminating the need to make fine, physical adjustments to the encoder orientation. Since this method measures piston position directly while the engine is operational, it can consistently produce accuracies better than ±0.1°CA [121]. The accuracy is subject to the degree of side-to-side piston wobble at TDC and, therefore, thoughtful placement of the sensor is required.
Unfortunately, due to the high cost of a TDC sensor system, the mechanical alignment procedure was the only viable option for the split-cycle engine. The determination of TDC was performed according to the procedure given in Lancaster, Krieger and Lienesch [89], who suggest it to be accurate within $\pm 0.1^\circ$CA when performed correctly. However, due to the half-cycle width of the Z-index pulse (see Figure 4.19), an accuracy of $\pm 0.2^\circ$CA is believed by the author to be more realistic. This can be extrapolated to an approximate error of 1% in the values reported for fuel conversion efficiency and IMEP.

### 4.4.3 Engine Pressure Measurements

An overview of the hardware used to measure the pressure in the engine’s intake port, cylinders, and crossover passage is shown in Table 4.5. The in-cylinder measurements were made with piezoelectric sensors, which are non-absolute and therefore require zero-level correction (see Section 5.1.1). Conversely, the intake port and crossover passage pressure transducers are piezoresistive type and provide absolute measurements. All of the sensor signals are amplified to a 0–10 VDC analog signal before arriving at the DAQ system.

<table>
<thead>
<tr>
<th>Location</th>
<th>Transducer</th>
<th>Range</th>
<th>Cooling</th>
<th>Amplifier</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake Port</td>
<td>Kistler 4043A5</td>
<td>0–5 bar</td>
<td>Uncooled</td>
<td>Kistler 4618A0</td>
</tr>
<tr>
<td>In-Cylinder</td>
<td>Kistler 6052C</td>
<td>0–100 bar</td>
<td>Uncooled</td>
<td>Kistler 5064</td>
</tr>
<tr>
<td>Crossover Passage</td>
<td>Kistler 4005BA</td>
<td>0–50 bar</td>
<td>Water Cooled</td>
<td>Kistler 4665</td>
</tr>
</tbody>
</table>

*Amplifiers 4665 & 5064 are part of the Kistler 2854A132 Signal Conditioning Platform (SCP).

**In-Cylinder Transducer Installation**

Installation of the in-cylinder pressure transducers requires careful placement of the measuring diaphragm with respect to the combustion chamber wall: too close and the transducer may be susceptible to overheating, too far away and pipe oscillations in the connecting passage will give erroneous readings [121]. The transducers were installed according to Figure 4.20, as recommended by the manufacturer (Kistler). Note the abundance of cooling passages surrounding the transducer, which relies on engine coolant to maintain its temperature below 350°C. The mounting bores were machined using a Kistler step drill, part number: 1300A51, followed by honing of the sealing surfaces using a special surface reamer, part number: 1300A79. To prevent distortion of the measuring diaphragm, the transducers were installed with a torque driver to a recommended value of 1.5 Nm [82].
Chapter 4: Experimental Setup

Intake Port Transducer Installation

The intake port pressure transducer was installed in the ceiling of the intake runner, approximately 100 mm upstream from the intake valve. The diaphragm of the sensor was mounted flush with the inner wall of the runner.

Crossover Passage Transducer Installation and Cooling

The Kistler 4005BA piezoresistive sensor used to measure absolute pressure in the crossover passage has a maximum continuous operating temperature of 100 °C. As such, a special cooling adapter, Kistler 7525A2, was installed between the engine and sensor, and a dedicated cooling circuit was built to continuously flow water through the adapter. The closed loop cooling circuit, shown in Figure 4.21, consists of a 12 VDC diaphragm pump, two in-line filters, a water reservoir, and a forced convection water-to-air heat exchanger. The tubing is 1/4 inch stainless steel, except near the transducer where it transitions to flexible nylon.

The filter located downstream from the pump serves two purposes: to remove contaminants that could potentially block the small passages of the cooling adapter, and to help reduce the amplitude of pump-induced pressure waves travelling through the water. These waves can cause signal noise in the transducer measurements known as cooling water crosstalk [121]. To verify this effect, the transducer signal was recorded with and without the cooling pump activated and no discernible difference in the output was found.
Kistler recommends to have a cooling water feed temperature less than 50°C at a flow rate of 0.3–0.5 L/min \[84\]. To reduce costs, an existing pump with a specified output of 1.1 L/min at 6 bar gauge pressure was used for this application. Due to the large overcapacity of the pump, a bypass line was incorporated into the system. This allowed the pressure and flow rate to be controlled through a metering valve in the bypass line. The system flow rate was determined by externally collecting and weighing the water flowing through the reservoir return line over a measured time period. The valve was adjusted until a flow rate of 0.43 L/min was achieved, resulting in a system pressure of 1.5 bar. The 11.4 L aluminum reservoir was found to provide enough thermal mass that air-cooling was sufficient to maintain the water temperature near ambient. A thermocouple located inside the reservoir was used to monitor the water temperature.

### 4.4.4 Temperature Measurement

All temperature measurements were taken using 1/16 inch diameter, K-type thermocouple (TC) probes; part numbers KMTXL and KMQL from Omega® Engineering Inc. The different part numbers correspond to the style of electrical connection. Both have a measuring uncertainty equal to the greater of 2.2‰ or 0.75 % of the reading. The TC signals were acquired by a National Instruments 24-bit, 16-channel, 9213 module installed in a USB cDAQ-9171 chassis. The module converts the millivolt TC signal into a digital temperature value and has built-in cold-junction compensation. Temperatures were sampled at a rate of 10 Hz since the approximate response time\(^4\) of a 1/16 inch sheathed TC probe is 250 ms in water and 4 s in air for a flow velocity of 0.3 m/s \[110\].

\(^4\) Response time is defined as the time required to reach 63.2% of an instantaneous temperature change.
Table 4.6 lists the location, type of junction, and part number prefix for each thermocouple measurement in the test apparatus. The junction type refers to the encasement (sheath) of the actual measuring junction. Three different types were used:

1. Grounded: the measurement junction is covered by, and touches, the sheath.

2. Ungrounded: the measurement junction is covered by the sheath without contact.

3. Exposed: the measuring junction is unsheathed and in direct contact with the fluid.

The sheathed TCs have a measuring junction that is not in direct contact with the fluid medium and are therefore less susceptible to corrosion or damage compared with exposed junction TCs. However, the sheath increases the TC response time. The ungrounded TC has the added benefit of being electrically isolated from the fluid medium, but also has the largest response time.

<table>
<thead>
<tr>
<th>No.</th>
<th>Measurement</th>
<th>Location</th>
<th>Junction</th>
<th>Part No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Crossover Passage</td>
<td>Midway Between Cylinders</td>
<td>EX</td>
<td>KMTXL</td>
</tr>
<tr>
<td>2</td>
<td>Cylinder Head Coolant</td>
<td>Below Intake Port</td>
<td>UG</td>
<td>KMTXL</td>
</tr>
<tr>
<td>3</td>
<td>Cylinder Head Coolant</td>
<td>Near Spark Plug</td>
<td>UG</td>
<td>KMTXL</td>
</tr>
<tr>
<td>4</td>
<td>Cylinder Head Coolant</td>
<td>Below Exhaust Port</td>
<td>UG</td>
<td>KMQXL</td>
</tr>
<tr>
<td>5</td>
<td>Cylinder Head Oil</td>
<td>Distribution Rail</td>
<td>GR</td>
<td>KMQXL</td>
</tr>
<tr>
<td>6</td>
<td>Transducer Coolant</td>
<td>Reservoir</td>
<td>GR</td>
<td>KMQXL</td>
</tr>
<tr>
<td>7</td>
<td>Engine Coolant</td>
<td>Heat Exchanger Outlet</td>
<td>GR</td>
<td>KMQXL</td>
</tr>
<tr>
<td>8</td>
<td>Intake Air</td>
<td>Below LFE</td>
<td>EX</td>
<td>KMQXL</td>
</tr>
<tr>
<td>9</td>
<td>Fuel</td>
<td>Surge Tank Outlet</td>
<td>GR</td>
<td>KMQXL</td>
</tr>
<tr>
<td>10</td>
<td>Engine Oil</td>
<td>Bottom of Engine Sump</td>
<td>UG</td>
<td>KMQXL</td>
</tr>
<tr>
<td>11</td>
<td>Engine Coolant</td>
<td>Thermostat Housing</td>
<td>UG</td>
<td>KMQXL</td>
</tr>
<tr>
<td>12</td>
<td>Exhaust Gas</td>
<td>Near Exhaust Port</td>
<td>EX</td>
<td>KMTXL</td>
</tr>
<tr>
<td>13</td>
<td>Exhaust Gas</td>
<td>Emissions Line, Pre-Chiller</td>
<td>GR</td>
<td>KMQXL</td>
</tr>
<tr>
<td>14</td>
<td>Exhaust Gas</td>
<td>Emissions Line, Post-Chiller</td>
<td>GR</td>
<td>KMQXL</td>
</tr>
</tbody>
</table>

GR = Grounded, UG = Ungrounded, EX = Exposed

4.4.5 Air/Fuel Ratio Measurement

The air-to-fuel ratio (AFR) was measured using a universal exhaust gas oxygen (UEGO) sensor, which compares the concentration of oxygen in the exhaust stream to that in the atmosphere. The sensor was mounted in the exhaust pipe wall, approximately 415 mm
downstream from the exhaust port. It was oriented to be on the top side of the pipe in order to prevent water and/or deposits from accumulating on the sensing element. A Horiba Mexa-700 meter was used to process the sensor output signal and is capable of displaying the result as either an oxygen percentage, AFR, or lambda ($\lambda$) ratio, the latter of which is defined in Equation (4.7). The Mexa-700 also outputs a configurable 0–5 VDC signal proportional to $\lambda$, which was recorded by the DAQ system. The air/fuel equivalence ratio ($\phi$) is the inverse of the lambda ratio and will primarily be used for describing the state of mixture stoichiometry in this work.

$$
\lambda = \frac{1}{\phi} = \frac{\text{AFR}_{\text{actual}}}{\text{AFR}_{\text{stoichiometric}}} = \frac{\left( \frac{m_{\text{air}}}{m_{\text{fuel}}} \right)_{\text{actual}}}{\left( \frac{m_{\text{air}}}{m_{\text{fuel}}} \right)_{\text{stoichiometric}}} 
$$

(4.7)

### 4.4.6 Spark Ignition System

The spark ignition system assembled for the split-cycle engine consists of a single spark plug connected to an inductive-type ignition coil through a high-tension lead. The spark plug is a model ER8EHIX made by NGK and has an M8 thread, one of the smallest available and selected specifically for ease of packaging within the 67 mm engine bore. The spark plug has a so-called heat rating of 8, which is the colder of the two plugs available in the M8 thread size and falls mid-range on NGK’s heat rating scale. It was assumed that the unoptimized cooling passages combined with 2-stroke operation of the combustion cylinder would result in a hotter than normal electrode temperature and thus a colder heat rating was desirable.

The ignition coil is a solid state, high-output unit from AEM Performance Electronics, model number 30-2853. According to the manufacturer, it can produce a maximum spark energy of 103 mJ for a duration of approximately 2.9 ms. The complete specifications of the spark coil can be found in Table 4.7.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Output (50pF load)</td>
<td>40 kV ± 10 %</td>
</tr>
<tr>
<td>Output Energy</td>
<td>103 mJ ± 7 %</td>
</tr>
<tr>
<td>Arc Duration</td>
<td>2.9 ms ± 10 %</td>
</tr>
<tr>
<td>Max Input Current</td>
<td>19 A</td>
</tr>
<tr>
<td>Base Dwell</td>
<td>3 ms</td>
</tr>
<tr>
<td>Max Continuous Dwell</td>
<td>9 ms$^*$</td>
</tr>
<tr>
<td>Max Cont. Duty Cycle</td>
<td>40 %</td>
</tr>
</tbody>
</table>

Table 4.7: Specifications of AEM inductive spark plug coil.
Timing of the spark is determined by a 5 VDC digital square wave generated in LabVIEW™ and supplied to the coil for a finite amount of time known as the *coil dwell period*. The spark is initiated at the end of the dwell period, as depicted in Figure 4.22. The manufacturer specifications for coil dwell were provided in milliseconds but the actual signal was implemented on a crank angle basis to ensure spark timing was repeatable in reference to TDC. The LabVIEW™ program given in Appendix E automatically adjusts the dwell start point based on the user-specified spark time and measured engine speed.

![Figure 4.22: Example of the spark coil dwell period in terms of crank angle degrees.](image)

### 4.4.7 Fuel Injection Control

The Siemens VDO injector used for the split-cycle engine is a peak-and-hold type, meaning a large voltage/current is supplied to the solenoid windings for a short duration, approximately 0.1–0.2 ms for rapid opening of the injector needle, followed by a period of low voltage/current excitation that maintains the injector in its open position. The electrical profile sent to the injector is controlled in LabVIEW™ through a National Instruments™ 9751 direct injector driver hardware module (originally developed by Drivven Inc.). The module is capable of operating both piezoelectric and solenoid type injectors, but the latter was used in this work due to the large packaging requirements of piezoelectric injectors. A “boost” power supply, internal to the 9751 module, is capable of outputting up to 190 V at 40 A and 15 A for peak and hold currents, respectively. The peak injector voltage is generated by the module and then switches to an external 12 V source for the hold period, which was supplied by a 12 V, 10 A power supply located in the instrumentation cabinet. Figure 4.23 shows a nominal peak-and-hold injector current profile, along with the specific
parameters that can be adjusted for the 9751 module through LabVIEW™. Assignment of these calibration parameters was done using the DI Calibrator (a pre-programmed VI from Drivven Inc.) through an iterative trial-and-error process. Interested readers can find the calibration procedure in the DI Driver user manual [48]. Table 4.8 lists the final calibration values that were used to operate the injector for the results presented in this work.

The quantity of fuel entering the engine was varied by changing the fuel injection duration, also known as the injection pulse width. For a nominal fuel line pressure of 68 bar, the injection durations were roughly between 3–5 ms for engine speeds and equivalence ratios ranging from 850–1200 RPM and $\phi = 0.8–1.0$, respectively.

![Nominal current profile for a solenoid type injector showing controllable parameters. Adapted from [48].](image)

**Figure 4.23:** Nominal current profile for a solenoid type injector showing controllable parameters. Adapted from [48].

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>High Voltage Target</td>
<td>60</td>
<td>V</td>
</tr>
<tr>
<td>Peak Current</td>
<td>8</td>
<td>A</td>
</tr>
<tr>
<td>Peak2 Current</td>
<td>5</td>
<td>A</td>
</tr>
<tr>
<td>Hold Current</td>
<td>3</td>
<td>A</td>
</tr>
<tr>
<td>HV Peak Time</td>
<td>0.27</td>
<td>ms</td>
</tr>
<tr>
<td>Peak Time</td>
<td>0.50</td>
<td>ms</td>
</tr>
<tr>
<td>Back Boost Time</td>
<td>1.00</td>
<td>ms</td>
</tr>
</tbody>
</table>
4.5 LabVIEW™ Programming

The programming work done in LabVIEW™ can be broken down into two parts: 1) data acquisition combined with low-level control of peripherals (Section 4.5.1), and 2) high-level control of the engine operation (Section 4.5.2). The primary difference between these two aspects is the hardware that they utilize. The DAQ and low-level control was performed on the USB-based devices (e.g. USB-6356), while the high-level engine control was done using the CompactRIO system. Both systems use virtual instrument (VI), block diagram based programming, which is comprised of a block diagram code and a graphical user interface (GUI) known as the front panel. Only a small portion of the block diagram code will be shown in this section, the remainder can be found in Appendix E.

4.5.1 DAQ and Low-Level Control

With the exception of exhaust emissions data, all acquisition, processing, display and logging of data occurs in a single VI that was programmed by the author. This VI also performs the low-level control (analog and digital outputs) for dynamometer speed, throttle position, and engine cooling. The basic VI structure contains six while-loops that can be segregated by their iteration speed: low sample-rate DAQ and low-level control occurs on a clock or time basis, whereas high-speed DAQ is triggered by, and thus based on, the crank angle position. A schematic of the VI structure is shown in Figure 4.24.

The crank angle based data is sampled every 0.1°CA, yielding data rates that are on the order of several hundred kHz. To ensure the data could be processed, displayed, and logged without impeding the real-time acquisition, a producer/consumer VI architecture was adopted. This architecture allows data to be transferred between while-loops in queued, first in/first out (FIFO) buffers that temporarily store the data in PC memory. By using separate loops with buffered communication, slower iterating loops, such as data display and logging, do not delay the acquisition of data from the DAQ device. The latter situation could potentially cause a device memory overflow condition, leading to data loss.

Digital Triggering for AI Synchronization

One of the main challenges for programming this data acquisition VI was enabling the measurements to be taken on a crank angle basis so that every analog sample was aligned with a tick of the encoder. To accomplish this task, the author has developed a small piece
Chapter 4: Experimental Setup

Data Acquisition and Low-Level Control VI

**Figure 4.24:** Structure of LabVIEW™ VI for DAQ and low-level control.

of code that monitors both encoder channels for state changes and then uses these events to trigger the analog acquisition. The encoder position, which is concurrently read by a 32-bit quadrature-enabled counter, simultaneously records alongside the analog data. The portion of block-diagram code that performs these tasks is shown in Figure 4.25.

The key elements of this code are as follows (labelled accordingly in the figure):

1. Digital change detection of channels A and B are used in lieu of a sample clock for acquiring the encoder position and analog inputs (AIs).

2. Digital de-bounce filters on each input line remove electrical noise/glitches, which were causing unintended counts to occur.

3. The time-based sample clock rate is used to set the maximum conceivable buffer size for each task.

4. An arm-start trigger begins data collection at 0°CA, so that each packet of buffered data corresponds to one complete engine revolution.

5. The ‘change detection’ and ‘AI input’ tasks are started before the ‘Cl encoder’ task (which contains the arm start trigger) to ensure they are ready to begin.
Figure 4.25: Segment of LabVIEW™ VI block diagram used to configure simultaneous acquisition of analog data and crank angle position.
4.5.2 High-Level Engine Control

The engine-control programming differs from the DAQ and low-level control because the VIs must execute at a much higher rate, and because multiple VIs must be used and intercommunicate across different hardware platforms. The entire control strategy must also be highly deterministic so that phase-critical aspects, such as ignition timing, occur precisely and reliably.

Figure 4.26 shows how the VIs are structured across the three platforms: the FPGA chip, the real-time processor, and the desktop PC. The FPGA VI is hard-coded onto a silicon chip and therefore tasks can execute with precise repeatability and true parallelism. The acquisition of crank angle position, and the subsequent control of fuel and spark, occurs at this level without user interface. Information is passed between the FPGA VI and the real-time (RT) VI using programmatic front-panel communication, which is a lossy type of data transfer. This allows the FPGA VI to execute its tasks with no adverse affects from RT processor operations. Despite being lossy, the communication between the RT and FPGA is fast and uses low overhead, making it ideal for low throughput data transfer, like fuel and spark timing set-point commands. The RT VI front panel is visible at the PC level via Ethernet communication with the cRIO system, and the Windows\textsuperscript{®}-based VI was only used for debugging purposes. The FPGA and RT VI block diagrams are shown in Appendix E, along with the RT VI front panel.

![Figure 4.26: Structure of LabVIEW\textsuperscript{™} project for engine control.]

4.6 Instrument Design Uncertainty

Inherent to any measurement system is uncertainty: error between the actual and measured values of a physical variable. As a first means of quantifying this error, the individual component uncertainty can be estimated. This is generally known as the design-stage uncertainty and is composed of two elements: zero-order uncertainty, \( u_0 \), which was taken
as ±\(\frac{1}{2}\) the measurement discretization resolution; and instrument uncertainty, \(u_c\), which is the systemic error of the DAQ system. The overall design uncertainty is calculated using the root-sum-squares (RSS) method on these individual components, as shown by Equation (4.8).

\[
u_d = \pm \sqrt{u_0^2 + u_c^2}
\]  

(4.8)

The systematic error is made up of many individual components, such as hysteresis, linearity, sensitivity, etc., and is calculated by taking the RSS of these component values:

\[
u_c = \sqrt{u_1^2 + u_2^2 + \ldots + u_i^2}
\]  

(4.9)

Table 4.9 lists the approximate design uncertainty, at full scale conditions, for every parameter measured in this work. Uncertainties were calculated from manufacturer provided specifications, with the exception of brake torque, which was estimated from the resolution of the scale used to weigh the calibration weights and the approximate placement error of the weights on the calibration arm. An example of the calculations used to produce the values in Table 4.9 is given in Appendix D.

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Range</th>
<th>Uncertainty</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air Flow Rate</td>
<td>0–46</td>
<td>±0.137 g/s</td>
<td></td>
</tr>
<tr>
<td>Fuel Flow Rate</td>
<td>0–0.418</td>
<td>±0.0043 g/s</td>
<td></td>
</tr>
<tr>
<td>Equivalence Ratio</td>
<td>0.034–1.1</td>
<td>±0.021</td>
<td>-</td>
</tr>
<tr>
<td>Fuel Pressure</td>
<td>0–68</td>
<td>±0.055 bar</td>
<td></td>
</tr>
<tr>
<td>Cylinder Pressure</td>
<td>0–250</td>
<td>±0.901 bar</td>
<td></td>
</tr>
<tr>
<td>Intake Pressure</td>
<td>0–5</td>
<td>±0.064 bar</td>
<td></td>
</tr>
<tr>
<td>Crossover Pressure</td>
<td>0–50</td>
<td>±0.520 bar</td>
<td></td>
</tr>
<tr>
<td>Brake Torque</td>
<td>0–565</td>
<td>±0.5 Nm</td>
<td></td>
</tr>
<tr>
<td>Temperature†</td>
<td>(−200)–1350</td>
<td>±2.2 °C</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>±0.75 %</td>
<td></td>
</tr>
</tbody>
</table>

† Uncertainty is the larger of ±2.2°C or ±0.75% of reading.
Chapter 5

Data Processing and Analysis

The digital test data, acquired by the DAQ system covered in Chapter 4, was post-processed using MATLAB®. This chapter provides a brief overview of how the data was processed and the evaluation metrics used for assessing the engine’s performance. Sections 5.1.1 and 5.1.2 describe the techniques used to properly scale and filter the cylinder pressure data, respectively, followed by the justification for the number of engines cycles used for data averaging in Section 5.1.3. The engine and combustion metrics used for performance analysis are covered in Section 5.2.

5.1 Data Post-Processing

All post-processing and off-line calculation was done using a MATLAB® script written by the author (split-cycle-post-processing.m), which can be found in Appendix C. The script verifies the integrity of the data by ensuring each cycle has the correct number of data points and confirms that the crank angle index counts sequentially from 0–360°. A programmatic check for misfiring cycles (see Section 5.2.6) records the cycle number and omits that cycle from statistical processing. This prevents misfiring cycles from skewing the combustion data. Corrections for non-absolute signals and filtering for noise is also performed programmatically. These aspects will now be discussed.

5.1.1 Zero-Level Pressure Correction

The piezoelectric cylinder pressure transducers are dynamic measuring devices that respond to a change in force on the measuring diaphragm. In other words, the output charge created
by the piezo crystal(s) diminishes with time and thus cannot be relied upon for an absolute measurement without zero-level correction. This is commonly referred to as pegging [41].

The compression cylinder was zero-level corrected using the intake port pressure as a reference datum. The cylinder pressure was adjusted according to Equation (5.1), where \( p(\theta) \) is the raw cylinder pressure, \( \rho_{in}(\theta) \) is the intake port pressure, \( p(\theta)' \) is the corrected cylinder pressure, and \( \theta_m \) and \( \theta_n \) define the start and end of the pegging range, respectively.

\[
p(\theta)' = p(\theta) + \frac{1}{n - m + 1} \left( \sum_{i=m}^{n} [p(\theta_i) - \rho_{in}(\theta_i)] \right)
\]

(5.1)

According to Davis and Patterson [41], inertial effects of a tuned intake system can impose systematic errors when using this method. However, for the low mean piston speeds used in this work (<3 m/s), the intake velocities are expected to be relatively small; therefore, air ramming effects were assumed to be negligible.

Ideally the pegging sensor would be located as close to, or even inside, the cylinder in order to minimize pressure differences caused by dynamic effects. The physical size of the Kistler 4043 sensor made this difficult, and it was therefore located in the wall of the intake runner, approximately 100 mm upstream from the intake valve. Nevertheless, overlaid plots of the cylinder and intake port pressures showed nearly identical trends with only a slight phase lag of approximately 5 °CA. A pegging interval from 160–180 °CA after TDC of the compression cylinder (ATDC-c) was used. The intake port pressure generally varied less than ±0.02 bar over this interval, which is less than the measurement uncertainty (see Table 4.9).

Pegging of the expansion cylinder pressure was complicated by the fact that the cylinder is not connected to an atmospheric intake port and the engine was not equipped with an exhaust port transducer. Therefore, the crossover pressure was used in place of the intake pressure in Equation (5.1). While the crossover transducer does measure absolute pressure, the oscillations present during the time of fluid transfer (i.e. the only time the two volumes are connected) make it less than ideal. Averaged values taken over a 15 °CA window, starting at TDC, were used to minimize the error associated with the pressure fluctuations. Figure 5.1 shows an example of raw and pegged data for the expansion cylinder under motoring conditions. Inlaid in the figure is an enhanced view of the pressure oscillations present during the time of fluid transfer. It can be seen that the crossover passage pressure follows the cylinder pressure oscillations quite closely, albeit with much less amplitude.
5.1.2 Data Filtering for Noise Rejection

Electrical noise is an unavoidable nuisance when low voltage signal cables are in close proximity to high voltage equipment, such as electric motors and devices powered by 120 V, 60 Hz service. Maintaining short cable lengths, segregating high and low voltage cabling in separate raceways, as well as using proper shielding methods (i.e. without ground loops), were the main noise minimization techniques employed in the development of the engine test bed presented in this work. Regardless, small amounts of high frequency noise were expectedly still present in the measurement chain.

For in-cylinder pressure measurements, signal noise was most noticeable during periods of low pressure, when the signal-to-noise ratio is the lowest. Signal noise has little effect on the results of cumulative-type calculations, since the noise amplitude tends to be small and self-cancelling over the summation period [127]. However, differential analysis of pressure data, such as the pressure rise rate (PRR) and the mass fraction burned (MFB), tend to exaggerate any noise present and thus data filtering/smoothing is commonplace [44, 64, 96, 125, 127].

Based on the various filtering/smoothing methods found in relevant literature [44, 64, 96, 125, 127], three different techniques were evaluated and compared for noise rejection:

![Figure 5.1: Example of zero-level correction on the expansion cylinder pressure trace using the crossover passage pressure as a datum. Cold motoring conditions at 850 rpm. Raw = uncorrected floating measurement from sensor, Pegged = raw data adjusted for offset error using absolute crossover pressure.](image-url)
1. A simple moving-average smoothing algorithm with a user-defined span. Both a weighted and unweighted version of this technique was applied, but only the unweighted results will be shown since they better depict the deficiency of this method.

2. The Savitzky-Golay filter, which uses unweighted linear least squares regression and a polynomial model of \( n^{th} \) degree to determine the filter coefficients [111]. In this comparison, a second-order polynomial fit and a 9-point span were used.

3. A Butterworth low-pass filter with a 5 kHz cut-off frequency. The built-in MATLAB\textsuperscript{®} function \texttt{filtfilt} was used to produce zero-phase distortion by sequentially processing the input data in the forward and reverse directions.

Figure 5.2 shows the results of the different filtering/smoothing methods on a single cycle of pressure data taken in the expansion cylinder of the split-cycle engine. Analysis of the magnified portions of the pressure trace show the high frequency smoothing effects of the aforementioned techniques.

![Figure 5.2: Comparison of pressure filtering methods used to reduce signal noise.](image)

The moving average was tested using a span ranging from 5 to 50 data points (0.5 to 5°CA). The curve shown in Figure 5.2 represents a 20 point average and shows excellent smoothing of the high frequency noise present during the low pressure part of the cycle. However, the amplitude of large pressure spikes is diminished with increasing span length and has difficulty following the trend of the physical data. Thus, a trade-off exists between high
frequency attenuation and the accuracy of the smoothed data. For this reason, the simple moving average was rejected as a means of noise filtering.

The Savitzky-Golay and Butterworth filters show very similar smoothing characteristics. Both methods closely follow the physical data trends and are within 0.05% of the peak pressure amplitude. For high frequency smoothing, the Butterworth filtered data is more fluid and shows less jitter when compared to the Savitzky-Golay data. Therefore, the Butterworth filter was selected as the best noise-rejection method for post-processing of the pressure data acquired in this work.

5.1.3 Selecting the Number of Engine Cycles to Analyse

To choose the number of consecutive engine cycles required for analytical accuracy, consideration of data storage capacity, processing time and parameter variability need to be addressed. Since data from multiple channels is acquired every 0.1°CA, the file size can quickly become excessively large. Numerical integration and manipulation of several million data points also has the potential to be cumbersome and time consuming.

Conversely, when cyclic variability is high, a large number of cycles may be required to assure confidence in the data. Cheung and Heywood [34] indicated the requirement of 100-plus engine cycles while statistically validating a one-zone burn-rate analysis. A more direct study by Lancaster et al. [89] indicates as few as 40 cycles may be necessary in achieving a sample mean within 3% of the population mean at a confidence level of 99.9%. However, the same study also reveals that under conditions of high cyclic variability, 300 cycles would be required to maintain the same level of confidence. An increased number of cycles is further supported by Brunt and Entage [25], who analysed the cycle-number effect on IMEP error. They concluded that over 100 cycles are generally required to produce an error of less than 1% in IMEP, but recommend using 300 cycles when practical.

Since most of the data processing was done offline using MATLAB®, 300 consecutive data cycles were selected for averaging single-operating-point parameters reported in the present work. With eight analog channels acquiring a data sample every 0.1°CA, combined with the crank angle encoder position measurement itself, a single data set contains 9.72 million data points. The data was saved from the LabVIEW™ environment to the hard drive of a PC in a .TDMS (Technical Data Management Solution) file format.
5.2 Engine and Combustion Analysis Metrics

The following is a brief overview of the metrics used to analyse the engine data. All metrics were calculated from individual engine cycles and then averaged to produce mean values. In other words, ensemble pressure traces were not used to calculate cycle-averaged parameters. The MATLAB® code written to perform these calculations can be found in Appendix C.

5.2.1 Mean Effective Pressure (MEP)

The torque output of an engine is a good means of assessing its practical work capability. However, in order to compare torque values between engines of differing sizes, one must also account for the size or displaced volume of the engine. Work output per unit volume is known as the mean effective pressure (MEP) and is calculated in one of two ways:

(i) Indicated Mean Effective Pressure (IMEP)

IMEP represents the work done by the combustion gases on a single piston, normalized by the displaced volume of that piston \( V_d \), as shown in Equation (5.2). IMEP is independent of the number of engine cylinders and rotational speed of the crankshaft. Positive values of IMEP represent work being done on the piston (e.g. during combustion), whereas a negative IMEP represents work being done on the gases.

\[
\text{IMEP} = \frac{\int p(V)dV}{V_d} \quad (5.2)
\]

The cyclic integral of \( p(V)dV \) represents the area enclosed by the pressure-volume trace of the indicated cylinder. In this work, numerical integration was performed using Simpson’s 1/3 method [32]. The integral approximation is shown in Equation (5.3), where \( N \) is the number of data points for a cycle.

\[
W_{\text{cycle}} = \int p(V)dV \approx \frac{1}{6} \sum_{n=1}^{N} (V_{n+2} - V_n) [p(V_n) + 4p(V_{n+1}) + p(V_{n+2})] \quad (5.3)
\]

In conventional four-stroke engines, the terms net and gross are used to distinguish IMEP values with and without the intake/exhaust pumping work, respectively [131]. Since the split-cycle engine does not perform these processes within a single cylinder, the net IMEP will be used to designate the combined IMEP of both cylinders, as depicted in Equation (5.4), where IMEP_{Cyl,1} is expected to be a negative value.

\[
\text{IMEP}_{\text{net}} = \text{IMEP}_{\text{Cyl,1}} + \text{IMEP}_{\text{Cyl,2}} \quad (5.4)
\]
(ii) **Brake Mean Effective Pressure (BMEP)**

BMEP is similar in concept to IMEP except the work is measured at the engine’s crankshaft, instead of pressures within the cylinder. It is more indicative of the useful work a given engine can produce, while still remaining independent of speed and displacement. The numeric difference between IMEP and BMEP, Equation (5.5), is known as the *friction mean effective pressure* (FMEP) and represents the mechanical losses of the engine and its driven accessories. In this specific application, the dynamometer belt drive, along with the associated bearings and pulleys, are located between the engine and torque transducer, and are therefore also included into the FMEP value. For this reason the majority of results will be given in terms of indicated power (IMEP).

$$FMEP = IMEP - BMEP \quad (5.5)$$

BMEP was calculated from the arithmetically averaged torque reading over three-hundred consecutive, steady-state cycles.

### 5.2.2 Coefficient of Variation (COV)

The coefficient of variation is the ratio of the standard deviation, \( \sigma \), to the mean, \( \bar{x} \):

$$COV = \frac{\sigma}{\bar{x}} \times 100 \% \quad (5.6)$$

The COV is useful for assessing cycle-to-cycle variation of a given parameter. The COV of IMEP is often used as an indicator of the inter-cycle repeatability with respect to the combustion process [70, 131]. Stone [131] indicates that \( COV_{\text{IMEP}} \) values greater than 5–10 % lead to a noticeable degradation in driveability for vehicular applications. The expanded version of Equation (5.6) is shown for IMEP in Equation (5.7), where \( \text{IMEP} \) is the mean value from \( N \) cycles. The coefficient of variation for the location of peak pressure (LPP) can be calculated in the same manner and will be used in the present work to assess cyclic variability of combustion phasing.

$$COV_{\text{IMEP}} = \sqrt{\frac{1}{N} \sum_{i=1}^{N} \left( \text{IMEP}_i - \text{IMEP} \right)^2} \times \frac{\text{IMEP}}{\text{IMEP}} \times 100 \% \quad (5.7)$$
5.2.3 Polytropic Indices

The compression and expansion processes can be approximated by the polytropic relation:

\[ pV^n = \text{constant} \quad (5.8) \]

where \( n \) is the polytropic index and is relatively constant over a large region of the compression and expansion strokes, before and after combustion, respectively. By expanding Equation (5.8) to include two points along each process path and taking the logarithmic value of both sides, the polytropic index can be calculated from Equation (5.9).

\[ n = \frac{\log\left(\frac{p_2}{p_1}\right)}{\log\left(\frac{V_1}{V_2}\right)} \quad (5.9) \]

It can be seen then, that the polytropic index is the slope of the process curve on a log-\( p \) versus log-\( V \) diagram. Within the post-processing script, a linear least-squares line was fit to the compression and expansion curves on the log \( p-V \) diagram and its slope was taken as the polytropic index for each respective process. Caution was employed in selecting the start and end points of the line fit. For the compression stroke, the starting point was delayed to avoid analog-to-digital discretization errors that are more prevalent at BDC due to the small change in pressure with volume; the ending point was selected to precede the crossover valve opening. Similarly, on the expansion stroke, the starting point was delayed to avoid overlapping with combustion; the ending point was constrained by a diminishing signal-to-noise ratio and/or opening of the exhaust valve. For the present work, the start and ending points of the least-squares fit line are given in Table 5.1. These values were determined through close scrutiny of the log \( p-V \) diagrams. Due to the large variability in combustion phasing, the starting point for the expansion index was selected dynamically by adding 40°CA to the LPP for each cycle. This prevented the expansion index from being calculated during the combustion period for slow burning cycles.

<table>
<thead>
<tr>
<th>Cylinder</th>
<th>Index</th>
<th>Start</th>
<th>End</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compression</td>
<td>( n_c )</td>
<td>60° ABDC</td>
<td>4° BXVO</td>
</tr>
<tr>
<td>Expansion</td>
<td>( n_e )</td>
<td>40° ALPP</td>
<td>10° BEVO</td>
</tr>
</tbody>
</table>

ABDC = After Bottom Dead Center  
BXVO = Before Crossover (Inlet) Valve Opens  
BEVO = Before Exhaust Valve Opens  
ALPP = After Location of Peak Pressure
5.2.4 Mass Fraction Burned (MFB)

One of the key aspects of this work was determining the rate at which combustion occurs within the split-cycle engine. This is generally known as the mass fraction burned (MFB), and is computed in terms of °CA. Two methods for calculating MFB were evaluated: the so-called Rassweiler and Withrow method (RW) and the normalized pressure ratio ($PR_N$) method. These methods will now be discussed and compared.

Rassweiler and Withrow Method (RW)

The original method for determining MFB from a cylinder pressure trace was developed by Rassweiler and Withrow in 1938 [118]. They were able to correlate flame images with the corresponding pressure traces to show that the unburned gas in the cylinder is compressed polytropically by the advancing flame front during combustion. In this way, the mass fraction burned, $x_b$, is related to the cylinder pressure, $p$, and cylinder volume, $V$, through:

$$x_b = \frac{p^\frac{1}{n}V - p_0^\frac{1}{n}V_0}{p_f^\frac{1}{n}V_f - p_0^\frac{1}{n}V_0}$$  \hspace{1cm} (5.10)

where:  
$p_0$ is the pressure at the time of spark  
$V_0$ is the volume at the time of spark  
$p_f$ is the pressure at the end of combustion  
$V_f$ is the volume at the end of combustion

The polytropic index $n$ is a constant, averaged from the compression and expansion indices. This is one of the inherent deficiencies of the original method, since $n$ is not actually constant during combustion. Also, $x_b$ automatically goes to unity at the user-defined end of combustion (EOC), when $V = V_f$. This presents a problem, as it is difficult to correctly define the EOC before having calculated the mass fraction burned.

The fundamental principle behind the work of Rassweiler and Withrow is that the pressure change within the cylinder, over a finite crank interval, is the combined result of combustion ($p_c$) and volume change ($p_V$):

$$\Delta p = \Delta p_c + \Delta p_V$$  \hspace{1cm} (5.11)
By taking the change in pressure associated with volume as a polytropic process, $\Delta p_V$ can be written in terms of the absolute cylinder pressure at a given crank angle, $p_\theta$:

$$\Delta p_V = p_\theta \left[ \left( \frac{V_\theta}{V_\theta + \Delta \theta} \right)^n - 1 \right]$$  \hspace{1cm} (5.12)

By noting that:

$$\Delta p = p_{\theta + \Delta \theta} - p_\theta$$  \hspace{1cm} (5.13)

Substitution of Equations (5.12) and (5.13) into Equation (5.11) yields the pressure rise due to combustion:

$$\Delta p_c = p_{\theta + \Delta \theta} - p_\theta \left( \frac{V_\theta}{V_\theta + \Delta \theta} \right)^n$$  \hspace{1cm} (5.14)

Since combustion does not take place at constant volume, a reference volume must used to normalize the data. The normalized combustion pressure is denoted by $p'_c$ in Equation (5.15). A common datum for the reference volume $V_{ref}$ is the clearance volume at piston TDC [13, 127].

$$\Delta p'_c = \Delta p_c \frac{V_\theta}{V_{ref}}$$  \hspace{1cm} (5.15)

Thus each rise in combustion pressure over a finite crank interval is assumed to be the result of a fraction of the fuel being burned. The EOC is indicated when $p_c$ becomes zero and changes in pressure are solely a function of volume. The MFB is therefore characterized as the cumulative combustion pressure rise at crank angle $\theta$ over the total cumulative combustion pressure rise:

$$x_b = \frac{\sum_{\theta_{spark}}^\theta \Delta p'_c}{\sum_{\theta_{spark}}^\theta EOC \Delta p'_c}$$  \hspace{1cm} (5.16)

Using this method Shayler et al. [127] outline three methods for defining EOC:

i) **First Negative** method: assumes combustion has ended when a single negative combustion pressure $\Delta p_c$ is calculated.

ii) **Sum Negative** method: similar to “first negative” but requires three successive points to reduce errors associated with signal noise.

iii) **Standard Error** method: assumes EOC is reached when combustion pressure $\Delta p_c$ has settled within one standard error of zero.

While the standard error method appears to be the most appropriate of the three, the authors noted its susceptibility in over predicting burn durations.
Chapter 5: Data Processing and Analysis

The method of Rassweiler and Withrow in both forms, Equations (5.10) and (5.16), are particularly vulnerable to the selection of the polytropic exponent, \( n \). However, it has been shown that with the proper selection of \( n \), the Rassweiler and Withrow method is in excellent agreement with more complex thermodynamic models [130].

The use of a variable polytropic exponent has also been investigated. Shayler [127] proposed switching polytropic coefficients, from a compression to expansion based value, at a user-defined point during the calculation of mass fraction burned. The study indicates that accurate prediction of the polytropic index is most important during the early and late stages of combustion, and recommends making the switch after “the first few percent of the charge has burned”.

In an effort to reduce the arbitrary nature of the method proposed by Shayler [127], Ball et al. [14] followed the same derivation used by Rassweiler and Withrow for Equation (5.10), but accounted for the difference in polytropic exponents between the burned and unburned gas. The consequent expression for the MFB is given in Equation (5.17), where \( n_c \) and \( n_e \) are the polytropic indices for compression and expansion strokes, respectively. This equation for MFB calculation will be hereafter referred to as the modified RW (MRW) method.

The benefit of using this method is that the EOC does not need to be known for any reason other than determining the interval over which the polytropic expansion index \( n_e \) can be calculated. By taking \( V_f \) as the volume at exhaust valve opening (EVO) instead of EOC, the MFB will rise to a value of unity and remain there, provided that the value of \( n_e \) is correct. An under-prediction of \( n_e \) will result in an overshoot of the MFB beyond one, returning to unity at EVO. Conversely, an over-prediction will result in the MFB only reaching unity at EVO. Comparisons made by Ball [14] between their method and the burn rate of a validated computer model are in excellent agreement, often superior to the original method of Rassweiler and Withrow.

**Normalized Pressure Ratio**

A second metric for analysing the pressure rise due to combustion is known as the *pressure ratio*, \( PR(\theta) \) [51]. It can be calculated according to Equation (5.18), where \( p_f(\theta) \) and \( p_m(\theta) \) represent the firing and motoring pressures for a given crank angle, respectively. The
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Pressure ratio therefore represents the departure of cylinder pressure from the motoring curve.

\[ PR(\theta) = \frac{p_f(\theta)}{p_m(\theta)} - 1 \]  \hspace{1cm} (5.18)

The normalized pressure ratio, \( PR_N(\theta) \), can then be calculated by dividing \( PR(\theta) \) by the maximum value in each cycle:

\[ PR_N(\theta) = \frac{PR(\theta)}{\max[PR(\theta)]} \]  \hspace{1cm} (5.19)

This method of estimating the MFB is advantageous due to its independence from the polytropic indices. An evaluation of this method performed by Eriksson [51] showed that the 50% MFB value varied less than 1% between the \( PR_N \) and standard RW methods.

Comparison of MFB Methods

The use of Equation (5.17) to calculate the MFB inherently assumes the pressure inside the cylinder changes according to the polytropic relationship \( pV^n = \text{constant} \), where the reference volume and pressure are taken at the start of combustion (SOC). It was quickly discovered by the author that for spark timings in advance of the crossover outlet valve closure (XOVC) time, Equation (5.17) yields incorrect results for the early stages of combustion. This is because the effective volume at these advanced spark timings encompasses the crossover passage and compression cylinder volume until the crossover outlet valve closes. Therefore, flow into the cylinder during the flame development period—when the pressure rise due to combustion is negligible, but the cylinder volume is expanding—manifests itself as a small rise in the MFB curve. This was determined by observing the difference between cycles with a SOC before and after XOVC. An example of this scenario is given in Figure 5.3, which shows that at 25°CA spark timing, both MFB calculation methods exhibit very similar curves, only deviating slightly towards the end of combustion (likely caused by an over-prediction of \( n \) for the MRW method). However, for the earlier spark timing of 19°CA, the MRW method clearly shows an error in the MFB for the early stages of combustion. This is because flow from the crossover passage is still entering the cylinder at this time and is misrepresented by Equation (5.17) as a rise in cylinder pressure due to combustion. Because the \( PR_N \) method is referenced from the motoring curve, this additional flow into the cylinder is inherently accounted for.

Because the \( PR_N \) method is more robust in the context of this engine, all MFB-related results presented in this work will be based on this method unless otherwise stated. All motoring curves used in the calculation of \( PR_N \) were taken with the engine at operating
temperature, immediately after firing operation. This was done to minimize the differences, such as wall heat transfer and piston ring sealing, between motored and fired engine cycles.

5.2.5 Burn Duration and Phasing

The rate at which combustion occurs and its timing relative to cylinder TDC are of primary interest in this research. The reader should become familiar with the following parameters as they will be used frequently throughout the remainder of this document:

- **CA\textsubscript{0−10}**: Crank angle interval from the start of combustion (SOC) to 10 % MFB. This will be frequently referred to as the (early) flame development period.
- **CA\textsubscript{50}**: Absolute crank angle position corresponding to 50 % MFB.
- **CA\textsubscript{10−90}**: Crank angle interval over which the bulk of combustion occurs; 10 % to 90 % MFB. This was the primary parameter used in quantifying the duration of combustion.
- **LPP**: Stands for location of peak pressure. This is the absolute crank angle at which maximum cylinder pressure ($p\textsubscript{max}$) occurs. Misfiring cycles were not included when calculating the mean LPP. The COV\textsubscript{LPP} was calculated in accordance with Equation (5.6).
5.2.6 Pressure Rise Rate (PRR)

By differentiating the cylinder pressure trace with respect to crank angle, the pressure rise rate (PRR), presented in units of bar/deg, was obtained. Very rapid PRRs can lead to mechanical failure and excessive combustion noise; therefore, PRR is important to monitor but minimal discussion will be provided herein on this metric.

The PRR was also used for misfire detection in the MATLAB® post-processing script by checking to see if a positive pressure change occurred between the SOC and EVO. This was more reliable than simply using the peak pressure value, since very late combustion cycles tend to have peak pressures lower than the pressure at the SOC—an attribute that is relatively unique to the split-cycle engine.

5.2.7 Fuel Conversion Efficiency

The fuel conversion efficiency, $\eta_f$, given by Equation (5.20), is the ratio of indicated cylinder work produced per cycle to the fuel energy supplied to the cylinder per cycle. The latter is calculated from the product of the measured fuel flow rate, $\dot{m}_f$, and the fuel’s heating value, $Q_{HV}$. In this work, the lower heating value\(^1\) (LHV) of methane at constant pressure was used, which corresponds to $Q_{LHV} = 50$ MJ/kg [117]. The cycle work was calculated according to Equation (5.3) and refers to the net work of both cylinders.

$$\eta_f = \frac{\dot{W}_{cycle}}{\dot{m}_f Q_{HV}} \times 100\% \quad (5.20)$$

5.2.8 Volumetric Efficiency

In engine terms, volumetric efficiency, $\eta_v$, is the amount of air ingested during the intake stroke normalized by the swept volume of the cylinder. In a practical sense, it is a measure of how effectively an engine can displace air, both in and out of the cylinder. In this work, the measured air entering the engine is specified in terms of a mass flow rate, and therefore the volumetric efficiency calculation is mass-based, as shown in Equation (5.21):

$$\eta_v = n \frac{\dot{m}_a}{\rho_0 V_d N} \times 100\% \quad (5.21)$$

\(^1\) The lower heating value refers to the fuel’s calorific value taken when all the combustion products are in gaseous states (i.e. the water has not been condensed).
where: \( n \) is number of revolutions per cycle (\( n = 1 \) for split-cycle)
\( \dot{m}_a \) is the actual mass air flow rate entering the cylinder
\( \rho_{a,0} \) is the density of air at NTP conditions
\( V_d \) is the displaced or swept volume of the cylinder
\( N \) is the rotational speed of the engine
Chapter 6

Engine Trials: Experimental Results and Discussion

6.1 General

The split-cycle engine presented in Chapter 3 was successfully fired for the first time in the fall of 2014. To date, the engine has logged over 60 hrs of run time in a variety of steady-state tests at engine speeds ranging from 850 RPM to 1200 RPM. The performance and emission characteristics of the engine, at various ignition timings and equivalence ratios, were evaluated using the metrics covered in Chapter 5. This chapter presents the results and discoveries from these tests, interleaved with discussion of their interpretation.

The chapter begins with an overview of the test conditions in Section 6.2. Section 6.3 explores the basic operating characteristics of the engine, in order to familiarize the reader with its function. Several important issues, limitations, and necessary mechanical adjustments are then discussed in Section 6.4. Section 6.5 provides a macroscopic analysis of the combustion characteristics, subdivided by duration, phasing, and stability. The exhaust gas emissions are presented in Section 6.6 and subsequently used to perform an energy analysis of the engine in Section 6.7.

All fixed operating points presented in this chapter have been repeated a minimum of three times, each time on a separate date. Parameters that were averaged from 300-cycle measurements were again averaged across the repeated data sets. From these three trials, the uncertainty or standard error (SE) of the mean is given as an average value for each plotted curve. Point-by-point error bars have not been used to enhance clarity of the figures.
6.2 Test Conditions

General

Table 6.1 lists the general specifications used to produce the results in this chapter. Three different engine speeds were investigated: 850 RPM (idle), 1000 RPM, and 1200 RPM (maximum for valvetrain with safety margin). The equivalence ratio was varied for each engine speed. Only full load, wide open throttle (WOT) conditions were investigated and without the use of any exhaust gas recirculation (EGR). All instruments were given a 2 hr warm-up time in advance of testing, and data was recorded for steady-state operation only.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value(s)¹</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Speed</td>
<td>RPM</td>
<td>850/1000/1200</td>
</tr>
<tr>
<td>Equivalence Ratio</td>
<td>φ</td>
<td>0.83–1.00</td>
</tr>
<tr>
<td>Fuel Injection Timing</td>
<td>°CA</td>
<td>90 (ATDC-e)</td>
</tr>
<tr>
<td>Throttle Position</td>
<td>%</td>
<td>100 (WOT)</td>
</tr>
<tr>
<td>EGR</td>
<td>%</td>
<td>0</td>
</tr>
<tr>
<td>Coolant Temperature</td>
<td>°C</td>
<td>75 ± 5</td>
</tr>
<tr>
<td>Intake Air Temperature</td>
<td>°C</td>
<td>23 ± 2</td>
</tr>
<tr>
<td>Fuel Temperature</td>
<td>°C</td>
<td>22 ± 2</td>
</tr>
<tr>
<td>Ambient Pressure</td>
<td>bar</td>
<td>1.033 ± 0.004</td>
</tr>
</tbody>
</table>

¹ Temperatures and pressures listed are average values measured from all tests, and ± indicates two standard deviations (2σ) of the measurement.

Engine Load

Figure 6.1 shows the highest load obtainable by the split-cycle engine over the range of air/fuel equivalence ratios tested. For all engine speeds, the highest net IMEP is approximately 6.1 bar and occurs near stoichiometric fuelling conditions. The average FMEP was found to be 2.30 ± 0.1 bar, and was negligibly affected by engine speed and load over the ranges that were tested. It is apparent from Figure 6.1 that the FMEP represents a significant fraction of the engine IMEP (mechanical efficiency, $\eta_m = 50$–67%), and the reader should remember that the brake torque measurement for BMEP includes losses through the dynamometer belt drive and associated support bearings. Due to the arbitrary nature of these losses, IMEP will be used exclusively to indicate load throughout this chapter, allowing for more accurate comparisons with other engines.
Chapter 6: Engine Trials: Experimental Results and Discussion

6.3 Engine Operating Characteristics

6.3.1 Working Pressures

Figure 6.2 is an example of the typical pressure traces measured simultaneously in each cylinder and the crossover passage of the split-cycle engine. For the data set shown, the spark timing or start of combustion (SOC) was 19°CA ATDC-e, and the fuel start of injection (SOI) was 90°CA ATDC-e (or ~290°CA before the next combustion event). The fuel injection timing was fixed for all results presented in this work. Valve timings shown in Figure 6.2 are approximate.

On a pressure-volume basis, a similar data set is shown in Figure 6.3 for both the compression and expansion cylinder, and more clearly demonstrates the cyclic process of each cylinder. In the left-hand figure, the compression cylinder process is divided into three main stages: 1) initial compression of the intake air, 2) transfer of compressed air into the crossover passage, and 3) the intake stroke. Ideally, the transfer into the crossover passage would be isobaric, but imperfect valve timing and dynamic effects of the flow cause a slight increase in pressure. Between stages 2 and 3, the crossover inlet valve closes and cylinder pressure falls rapidly as residual trapped mass re-expands. At the same time the intake valve opens and the pressure remains around atmospheric as fresh air fills the cylinder.

The right-hand figure is the corresponding pressure of the expansion cylinder. Starting at BDC, the piston rises on the exhaust stroke and, with no appreciable restriction, the
Figure 6.2: Example of measured cylinder and crossover passage pressure traces for the split-cycle engine. Valve timings shown are approximate.

Figure 6.3: Example of pressure-volume diagrams for the compression cylinder (left) and expansion cylinder (right). Data for 850 RPM, WOT, spark timing: 22°ATDC-e, $\phi = 1$. 
pressure remains marginally above atmospheric. Pressurization of the cylinder by the incoming air/fuel mixture begins to occur approximately 12° BTDC-e. The pressure reaches a maximum around 10° ATDC-e, which is the effective TDC\(^1\) of the engine. The cylinder pressure then begins to decrease as the piston moves away from TDC, followed by ignition of the air/fuel mixture that creates a large pressure rise. The reduction in pressure prior to combustion is undesirable, caused by a limitation of the engine design that is discussed in Section 6.4.2. Once combustion is complete, the products are expanded to BDC where the exhaust valve opens and blow-down occurs. Repetition of the cycle begins with the ensuing exhaust stroke.

Crossover Passage Pressure

Ideally, the pressure inside the crossover passage of a split-cycle engine would be isobaric. In reality, however, heat transfer and mass leakage cause a gradual decline in the crossover pressure throughout the engine’s cycle. During the period when both crossover valves are closed (between XOVC and XIVO), the average decrease in pressure was found to be approximately 1.0 bar or a difference of 6\%. This was consistent regardless of the engine speed or the crossover pressure magnitude. Table 6.2 lists the average pressure characteristics measured inside the crossover passage for all test conditions. The relatively small standard deviation of these numbers is a testament to the consistency in pressure with which the crossover passage operates. Peak pressures within the cycle were generally found to be around 3 bar above the average value, and occurred during the transfer period: approximately between \(-10\) to \(20\)° CA ATDC-e, as shown back in Figure 6.2. The injection of fuel into the crossover passage resulted in a pressure increase of \(0.1 \pm 0.05\) bar or approximately 0.5\% of the mean pressure; therefore, it can be considered negligible.

<table>
<thead>
<tr>
<th>RPM</th>
<th>Crossover Pressure (bar)</th>
<th>Compression Ratio(^2)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Cycle Mean</td>
<td>Cycle Max.</td>
</tr>
<tr>
<td>850</td>
<td>17.2 ± 0.2</td>
<td>20.3 ± 0.5</td>
</tr>
<tr>
<td>1000</td>
<td>17.4 ± 0.2</td>
<td>20.6 ± 0.4</td>
</tr>
<tr>
<td>1200</td>
<td>18.8 ± 0.4</td>
<td>22.2 ± 0.7</td>
</tr>
</tbody>
</table>

\(^1\) Pressure drop between XOVC and XIVO
\(^2\) Based on a polytropic indices given in Figure 6.4

\(^1\) At 10° ATDC-e, in reference to the expansion cylinder, the minimum compressed volume is achieved, since both cylinders are open to the crossover passage at this point in time.
From Table 6.2 it can be seen that the average increase in crossover pressure from 850 RPM to 1000 RPM is significantly less than that from 1000 RPM to 1200 RPM. In fact, the percent difference is 1.2% versus 7.7% for the former and latter, respectively. Since the pressure loss from inside the crossover passage over the duration of a cycle is practically identical for all three engine speeds, it can be inferred that the rate of heat transfer and mass leakage are presumably the same and do not account for this difference. Therefore, the increase in average pressure must originate from the interaction with one or both of the cylinders.

Figure 6.4 shows the compression cylinder pressure plotted versus cylinder volume on dual logarithmic axes, for three arbitrarily selected cycles (one for each engine speed). Recall from Chapter 5 that the slope of the logarithmic $p$-$V$ diagram represents the polytropic exponent of the process under examination. Figure 6.4 shows that despite the small increase in $n_c$ with RPM, the larger crossover pressure magnitude is actually a result of a delay in the XIVO time. This is shown with enhanced detail in the right-hand side of Figure 6.4, where the phase lag between 850 RPM and 1200 RPM is approximately 1–2°CA, causing the valve to open in the latter case when the cylinder pressure has risen an additional 1.5–2 bar. The magnitude of this increase correlates well with the average values given in Table 6.2.

The compression ratios listed in Table 6.2 were calculated by solving for the volume ratio of a polytrophic compression ($pV^n = $ constant) between the cylinder pressure at intake BDC and the mean crossover passage pressure. The average polytropic indices calculated for the compression cylinder ($n_c$), shown in Figure 6.4, were used for $n$. Despite having a geometric compression ratio of 108:1, the current configuration of the split-cycle engine only effectively
compresses the gas between 8.4:1 to 8.7:1 by volume. These values are lower than most current production engines, which have compression ratios around 10:1 [112]; although this number is based on geometry, not actual gas pressures. Temperature measurements taken inside the crossover passage, ranging from 165–185°C, indicate that an increase in compression ratio is certainly possible without exceeding the autoignition temperature of methane (∼400°C at 20 bar [106]). Since the compression cylinder clearance volume has already been minimized, this additional compression would need to come from a reduction in crossover passage volume or a change in crossover valve timing. The effects of making such changes are investigated numerically in Chapter 7.

**Pressure Oscillations**

The reader may have noted oscillations present in both the cylinder and crossover pressure traces of Figures 6.2 and 6.3. These should not be misinterpreted as measurement noise, which was already removed using a low-pass filter (see Section 5.1.2). The oscillations are caused by expansion waves created when either of the crossover valves are opened, and quickly dampen out once of the crossover valves have closed. The frequency of oscillation is on the order of 1–2 kHz, which corresponds very closely to the first harmonic of a standing wave in a closed-ended pipe that has a length equivalent to the centreline distance between the two cylinder pressure transducers. Furthermore, the oscillation amplitudes for the two cylinders are almost perfectly out of phase from one another, as would be the case for the ends of a closed pipe. The oscillation in the crossover passage follows the phase of the expansion cylinder, since the transducer is offset towards the outlet side of the crossover passage. However, its oscillatory amplitude is reduced since it is closer to the mid-point pressure node of the standing wave. The magnitude of these waves depends on the volume of the crossover passage, which will be discussed further in Section 7.4.2.

**Cylinder Filling Time**

When the expansion cylinder piston is nearing TDC on its exhaust stroke, the exhaust valve closes and the crossover outlet valve begins to open. Based on the valve timing used in this work, the trapped residual mass constitutes around 3–4% of the total product mass. As the cylinder is pressurized with the air/fuel mixture, combustion must be quickly initiated before the piston recedes too far from TDC. Thus, the crank angle duration required to fill the cylinder is important, since the bulk of combustion cannot occur until the crossover valve has closed, and the valve should not close until the cylinder has completely filled.
Figure 6.5 shows the average cylinder and crossover pressure traces for engine speeds of 850 RPM and 1200 RPM. The crank angle duration required to fill the cylinder was approximated as the interval between the nominal crossover outlet valve opening (XOVO), 345°CA, and the ensuing crank angle of peak pressure. For engine speeds of 850 RPM and 1200 RPM, the filling duration is therefore 9.3°CA and 10.2°CA, respectively. By making the assumption that the crank angle speed is constant throughout the cycle, the filling time was calculated to be approximately 1.8 ms at 850 RPM and 1.4 ms at 1200 RPM. In other words, despite the slightly longer filling duration (on a °CA basis) at higher speed, the cylinder is actually filling faster on a time basis. This is likely due to the increased crossover pressure at the higher RPM. At the time of XOVO, the cylinder-to-crossover pressure ratio for the 850 RPM case divided by the same ratio for the 1200 RPM case is approximately equal to the ratio of their respective fill times. Therefore, it can be speculated that the fill time is marginally affected by engine speed, since the elevated crossover passage pressures associated with higher engine speeds compensate for the reduced fill time available at higher RPM. Due to the limited range of engine speeds tested, it is not known if this trend would continue with engine speeds faster than those investigated.

![Figure 6.5](image-url)  
**Figure 6.5:** Example of expansion cylinder filling duration for engine speeds of 850 and 1200 RPM.

It is now apparent that the filling duration is relatively short in comparison to the crossover valve opening duration, the latter of which is nominally 54°CA including ramps, 36°CA without. To clarify, the XOVO location shown in Figure 6.5 corresponds to a static measured valve lift of 0.025 mm. Based on the lobe centreline, the crossover outlet valve is actually supposed to open at 339°CA; however, it can be seen that the cylinder pressure does not
begin to rapidly rise until the end of the ramp period, at approximately 350°CA. Thus, for the same ramp configuration, the ramp-to-ramp duration of the valve only needs be in the range of 5–10°CA for engine speeds up to 1200 RPM. This is assuming a similar flow coefficient, and thus lift height, could be achieved in that time frame—a difficult scenario for a conventional valvetrain. Provided the valve could be operated with such a short duration, the benefit of an early crossover outlet valve closure (XOVC) would be combustion phasing closer to TDC and potentially a higher crossover passage pressure, since the compression cylinder would still be finishing its compression stroke at the time of XOVC.

Through linear extrapolation, the filling duration at an engine speed of 5000 RPM is in the vicinity of 25°CA. Thus, it can be anticipated that fixed valve timing may pose a problem over a more extensive RPM range. The use of variable valve timing is likely to be required for successful operation over a wide range of engine speeds.

### 6.3.2 Exhaust Gas Temperature

The exhaust gas temperature (EGT) was measured approximately 76 mm downstream from the exhaust port and was sampled on a clock basis at 10 Hz, with no correlation to cycle position. Further temperature measurement details can be found in Section 4.4.4. Since neither the flow rate nor the gas temperature are steady with time, the EGT values provided here are under-predicted approximations. Based on the average standard error of repeated experiments, combined with the instrumentation uncertainty, the EGT measurement uncertainty can be estimated as ±4°C, not including the time-averaging bias.

Figure 6.6 shows the band of exhaust temperatures over the range of air/fuel ratios tested. Initially, the temperature is decreasing with leaner mixtures, since the fuelling rate to the engine is decreasing. This trend diminishes as the equivalence ratio is further leaned out and an upturn in temperature occurs around $\phi = 0.86$. The cause of this sudden rise in temperature comes from an increasing number of cycles that exhibit very slow and/or delayed reaction rates, leading to under-expanded conditions when the exhaust valve opens.

The change in EGT with ignition timing is plotted in Figure 6.7 for stoichiometric operation. For a given engine RPM, the figure shows an almost linear decrease in exhaust temperature as spark timing is advanced towards TDC. In fact, the change in temperature with spark timing is on the order of 12–16°C/°CA and demonstrates the strong dependence of EGT on combustion phasing. The reduction in exhaust gas energy (or enthalpy, $h(T)$) at earlier spark timings is recovered, at least partially, by an increase in piston work (IMEP).
Figure 6.6: Average exhaust gas temperature (EGT) as a function of equivalence ratio.

Figure 6.7: Average exhaust gas temperature (EGT) as a function of ignition timing.
### 6.3.3 Mixture Homogeneity

During the development of the split-cycle engine, one of the primary questions asked was whether or not adequate air-fuel mixing could be achieved prior to initiating combustion. In part, the answer to this question came by chance from an issue with the injection control module that resulted in fuel only being injected once every three engine cycles. This was discovered by analysing the individual-cycle pressure traces for the crossover passage, which show a distinct rise in pressure from the injection event. An example is shown in Figure 6.8, where the crossover pressure traces have been ensemble averaged in cycle multiples of three. Since fuel is only being injected once every third cycle, a relatively long injection duration of 12.7 ms was required to maintain stoichiometric operating conditions. The injection event is easily seen in the ensemble averaged pressure trace of cycles 3,6,9, etc.

![Figure 6.8: Ensemble averaged crossover passage pressure traces showing injection event every third cycle. SOI = Start Of Injection.](image)

Several data sets for engine speeds ranging from 850–1150 RPM at WOT and 25°CA spark timing were acquired before the injection problem was discovered. The simple fact that the engine was operational under these fuel dosing conditions is an indication of adequate mixture uniformity within the combustion chamber. Furthermore, the coefficient of variation for net IMEP, \( \text{COV}_{\text{IMEP}} \), which is a measure of cyclic variation, was between 3.4–6.6 %—a surprisingly low range given the circumstances.

Average IMEP values for the first-, second-, and third-cycle-multiples of the missed-pulse data are shown in Figures 6.9 and 6.10 for stoichiometric and lean operation (\( \phi \approx 0.95 \)),...
respectively. For stoichiometric conditions, the highest IMEP occurred in the cycle immediately following the injection pulse, and subsequently was reduced over the next two cycles. By contrast, for the slightly lean conditions, IMEP is more consistent across all three cycle multiples, especially between the first two cycles following injection. It is speculated that the larger plume of fuel introduced under stoichiometric operating conditions is more difficult to break apart than the smaller plume of the lean mixture. This leads to the richest mixture entering the cylinder first, followed by subsequently leaner mixtures, which causes the gradual decrease in IMEP. Under lean conditions, mixing appears to improve based upon the stabilization of IMEP across all three cycle multiples.

Figure 6.9: Average IMEP for first, second, and third cycle multiples, with fuel injection during cycle three only. Stoichiometric air-fuel ratio ($\phi = 1$).

Figure 6.10: Average IMEP for first, second, and third cycle multiples, with fuel injection during cycle three only. Lean air-fuel ratio ($\phi \approx 0.95$).
Consistency amongst the various engine speeds may indicate that the residence time of the fuel within the crossover passage has no effect on mixing, at least over the limited RPM range that was tested. Based on these observations, it can be inferred that fuel/air mixing is occurring both inside the crossover passage and during the highly-turbulent transfer period from the crossover passage into the cylinder. However, the latter is limited by large stratifications within the crossover passage dependent upon the relative proportions of air and fuel admitted into the combustion chamber. A more complete understanding of the mixture homogeneity over the duration of the cycle could be obtained by means of in-cylinder and/or crossover passage gas sampling.

It should be explicitly stated that under normal operating conditions (i.e. all other results presented in this work), the engine uses a single fuel injection pulse per cycle (revolution). Consequently, the fuel injection duration was also considerably smaller than what was listed in Figure 6.8, with typical pulse widths ranging from 3.5–5 ms.

### 6.3.4 Volumetric Efficiency

The volumetric efficiency, $\eta_v$, of the split-cycle engine was of primary interest, as the reader may recall from Chapter 1, due to the relatively poor volumetric efficiency observed in port-injected NG engines. By locating the fuel injector in the crossover passage of the split-cycle engine, the gaseous fuel no longer displaces the intake air charge, which is the root cause of the volumetric inefficiency in a typical NG engine. However, the volumetric efficiency of the split-cycle engine was still determined to be low, ranging from 71–75 $\%$ at WOT for all equivalence ratios and spark timings tested. As engine speed was increased, the average volumetric efficiency did not change by any appreciable amount, however, the spread of data was reduced, with the standard deviation in $\eta_v$ decreasing from $\pm 1.2$ $\%$ at 850 RPM to $\pm 0.5$ $\%$ at 1200 RPM.

For reference, the volumetric efficiency of the Kubota Z482 engine was measured on a separate test stand with an identical mass air flow meter, and compared with the split-cycle values. Since both engines have the same intake valve diameter, bore size, and stroke length, the potential for cylinder filling should be similar. The result of the comparison is shown in Figure 6.11, and it can be seen that the volumetric efficiency of the split-cycle engine is between 8–15 $\%$ lower than the Kubota engine.

The question that remains is: what causes the reduction in volumetric efficiency? Unlike a port injected NG engine, fuelling of the split-cycle engine has no effect on aspiration of the intake air—a fact verified by examining the volumetric efficiency under motoring conditions.
What the split-cycle does have, however, is a much higher cylinder pressure preceding the opening of the intake valve. This is because the clearance volume in the compression cylinder must always be a finite value to provide mechanical clearance for thermally expanding components. Combine this with the inevitable ring-land crevice volumes, and a fraction of the intake air mass will always remain in the compression cylinder, at high pressure, at the end of the compression stroke. On the subsequent intake stroke, this air must first re-expand before the induction of fresh air can take place. The effect of this residual mass can be seen in Figure 6.12, where the initial 30°CA of the intake stroke is devoted to trapped mass re-expansion. In fact, only the first 6°CA or so is a true expansion, at which point the intake valve opens (IVO) and the cylinder de-pressurizes back through the intake port, labelled as ‘back-flow’ on the diagram. Since the cylinder cannot begin to fill with fresh air until the pressure has reduced to atmospheric conditions, this initial 30°CA, or roughly 10% of the intake stroke by volume, does not contribute to the air induction process. If the displaced volume \((V_d)\) used for reference in the volumetric efficiency calculation, Equation (5.21), is reduced by 10% then the volumetric efficiency is increased proportionally resulting in values around 80%. Thus, it is apparent that the deficiency between the two engines is largely accounted for by the attenuated intake stroke.

Unfortunately, this is simply an operating characteristic of the split-cycle engine, which would only get worse with increasing crossover pressure. A reduction in the clearance and crevice volumes of the compression cylinder, and/or increasing the intake pressure
(i.e. boosting) are likely candidates to help improve the volumetric efficiency of the split-cycle engine. The residual volumetric efficiency loss can be attributed to those mechanisms typically found in all engines: charge heating, flow friction, and pulsation of the flow [70].

![Figure 6.12: Example of back flow through intake port of the split-cycle engine at IVO.](image)

If IVO timing was delayed to allow the cylinder pressure to fully expand back to atmospheric conditions, the compression cylinder IMEP could be reduced slightly. This is perhaps better understood by looking back to Figure 6.4 and visualizing the expansion process (left-hand side of pressure-volume loop) to be a straight line. In this case, the area enclosed inside the diagram, representing the piston pumping work, would be smaller, and no back-flow through the intake would occur. Furthermore, the cylinder de-pressurization for early IVO (while $p_{cyl} > p_{port}$) will generate a strong compression wave back through the intake runner. For a single-pipe system, as is the case here, the compression wave will be reflected back from the intake-opening as an expansion wave. If this occurs while the intake valve remains open, then a decrease in cylinder pressure will ensue, potentially decreasing the amount of trapped mass. Thus, it can be expected that delaying IVO until the cylinder pressure is equal to, or less than, the intake port pressure will reduce the compression work and possibly increase the trapping efficiency of the compression cylinder. The latter depends on both the intake valve timing and the intake manifold configuration.
6.4 Issues and Limitations

Throughout the testing regime, several problems were encountered with the engine in terms of its proper function. The most important of these issues, many of which limit or hinder the engine’s performance, will be addressed in this section.

6.4.1 Valve Timing Adjustment

The valve timing was set under static engine conditions, by physically measuring the lift of each valve relative to the crank angle location. Initially, the crossover inlet valve was adjusted to close approximately 4°CA BTDC-c, in incorrect anticipation of a closing delay expected from timing belt stretch. In reality, the spring pressure on the closing flank of the reverse poppet cam lobe caused the camshaft to accelerate during that time, effectively closing the crossover inlet valve 10°CA before the compression cylinder reached TDC. Consequently, a small mass of air was being trapped in the cylinder, compressed, and subsequently re-expanded, as shown by the pressure-crank angle diagram in Figure 6.13.

![Figure 6.13](image-url)

**Figure 6.13:** Example of valve timing effect on compression cylinder pumping work. R1 and R2 stand for the valve timing revisions shown in Table 6.3.

By retarding the closure of the crossover inlet valve, this undesirable pumping work was effectively changed into useful compression work, increasing the crossover passage pressure by approximately 1.9 bar or 11% at 850 RPM. Figure 6.13 shows an example of the crossover and cylinder pressures before and after the valve timing change, labelled as R1 and R2,
respectively. Despite eliminating the pressure spike at TDC, an increase of 0.15 bar or 3.8% was observed in the compression cylinder IMEP, due to the higher overall working pressure. The end result was beneficial, however, since the volumetric efficiency improved by approximately 4% (to the levels listed in Section 6.3.4), and the increased crossover pressure allowed for higher peak combustion pressures to be achieved.

To minimize the effects of stretch, a thicker timing belt was installed during the changeover from R1 to R2 valve timing. This required the belt pitch to increase from 5 mm to 8 mm, which resulted in minor changes to the crossover outlet valve timing. The intake and exhaust valves were also purposely phased earlier in the cycle, based on the difference in timing between nominal and dynamic operation. The exact changes made to the static valve timing are given in Table 6.3. All figures and data provided in this document, including all previous information in this chapter, is for R2 valve timing.

Table 6.3: Valve timing adjustments made after initial operation.

<table>
<thead>
<tr>
<th></th>
<th>Revision</th>
<th>Open</th>
<th>Close</th>
<th>Centreline‡</th>
</tr>
</thead>
<tbody>
<tr>
<td>Intake</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1</td>
<td>36° ATDC</td>
<td>38° ABDC</td>
<td>132° ATDC</td>
<td></td>
</tr>
<tr>
<td>R2</td>
<td>26° ATDC</td>
<td>31° ABDC</td>
<td>119° ATDC</td>
<td></td>
</tr>
<tr>
<td>Crossover In</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1</td>
<td>27° BTDC</td>
<td>16° ATDC</td>
<td>3.5° BTDC</td>
<td></td>
</tr>
<tr>
<td>R2</td>
<td>22° BTDC</td>
<td>23° ATDC</td>
<td>0.5° BTDC</td>
<td></td>
</tr>
<tr>
<td>Crossover Out</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1</td>
<td>11° BTDC</td>
<td>29° ATDC</td>
<td>8° ATDC</td>
<td></td>
</tr>
<tr>
<td>R2</td>
<td>15° BTDC</td>
<td>27° ATDC</td>
<td>6° ATDC</td>
<td></td>
</tr>
<tr>
<td>Exhaust</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>R1</td>
<td>12° BBDC</td>
<td>2° BTDC</td>
<td>95° BTDC</td>
<td></td>
</tr>
<tr>
<td>R2</td>
<td>21° BBDC</td>
<td>12° BTDC</td>
<td>108° BTDC</td>
<td></td>
</tr>
</tbody>
</table>

* Valve timing relative to TDC of expansion cylinder. Subtract 20° CA for compression cylinder.
** Open and closing values are referenced to 0.025 mm of static measurable valve lift.
‡ Measured centreline may not coincide exactly with calculated open/close midpoint.

6.4.2 Peak Pressure Limitation

The drop in expansion cylinder pressure immediately prior to combustion, as shown on the right side of Figure 6.3 and again in Figure 6.14, is detrimental to engine performance. Ideally, ignition timing should occur so that the cylinder pressure begins to rise as soon the transfer from the crossover passage is complete. In reference to the aforementioned figures, this would require an earlier ignition timing. However, advanced ignition timing leads to a higher peak combustion pressure and it was discovered that the crossover outlet valve has a limited ability to remain closed during this period of peak pressure.
For example, at 850 RPM, WOT, and 20° ATDC-e spark timing, cycles with a peak cylinder pressure in excess of approximately 32 bar were found to push open the crossover outlet valve at the point of peak cylinder pressure. This is evident by examining the crossover passage pressure traces in Figure 6.14, which shows two arbitrarily selected cycles: one with a peak pressure above 32 bar (cycle A) and one with a peak pressure below 32 bar (cycle B). Cycle A clearly shows a sudden rise in crossover pressure at the location of peak cylinder pressure, caused by the gas force from combustion overcoming the crossover valve closure force. This phenomena was further substantiated by the observed reduction in IMEP for all cycles with a suspected valve opening event. In the case of Figure 6.14, cycle A has a higher peak pressure than cycle B, but, counter-intuitively, has a lower IMEP by a magnitude of 0.22 bar. This is a direct consequence of the unwanted crossover valve opening, which allows the gas to expand without contributing to the piston work.

The closure force relies on the differential pressure across the head of the valve, in addition to the pre-load applied by the valve spring. Recall from Section 3.5.8 that the spring pre-load is adjustable, and was set to the maximum value obtainable by the current spring design: approximately 600 N. An updated version of Figure 3.31 in Chapter 3 has been re-drawn in Figure 6.15, showing the operating regime of the engine. There is a clear discrepancy between the allowable peak combustion pressure based on design (25 bar greater than the crossover pressure), versus the actual value, which is approximately 7–10 bar less. This discrepancy can be attributed to a faulty assumption made during the design, in which the backside area of the RPV was taken at its outer diameter. This implies the valve sealing
surface is roughly 4 mm wide, when in reality it is only around 1.5 mm wide. An additional range of closure forces has been added to Figure 6.15, using a more realistic front-to-back area ratio, \( A_f/A_b \), based on this thinner sealing surface. It shows the current operating regime is at the limit of the closure force, which explains why the valve is being pushed open by combustion pressure.

![Figure 6.15: Maximum allowable pressure difference between cylinder and crossover passage as a function of the absolute pressure in the crossover passage. Operating regime is shown to be at upper limit of valve closure force under realistic area-ratio conditions.](image)

A stiffer valve spring could be employed to obtain a higher pre-load, but would require reworking of the valve spring bridge and/or spring retainers. The strength of the valvetrain components would also need to be re-evaluated, since the design was based on the maximum forces generated by the current spring. Therefore, the original RPV spring was retained and the ignition timings used in the present work were limited by the peak combustion pressure. The most advanced spark timings achievable were 20, 18, and 16 °CA ATDC-e, corresponding to engine speeds of 850, 1000, and 1200 RPM, respectively. For the air/fuel ratio investigations, these values will be used almost exclusively. Because maximum brake torque (MBT) timing was not achieved, the potential for increased work exists.

### 6.4.3 Crankcase Blow-By

As part of the data validation process, redundant measurements were compared as a means of verifying instrument accuracy. One such measurement was the air/fuel ratio, which is interpreted directly by the exhaust oxygen content, and was compared with the actual air
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and fuel mass flow rates measured upstream of the engine. It was discovered that the values obtained by these two methods were not in agreement, with the mass flow meters showing a leaner mixture than the O\textsubscript{2} sensor in the exhaust. The difference in equivalence ratio was 0.05 ± 0.01, and was found to be consistent over the range of conditions tested.

The difference in air flow between $\phi = 1$ and $\phi = 0.95$ was determined by assuming the measured fuel flow rate was correct and solving for the mass air flow rate required for stoichiometry. The result was then compared to the actual measured air flow rate, yielding a difference of approximately 0.16 g/s (8 SLPM) at 850 RPM.

It was hypothesized that crankcase blow-by was to blame for the change in air/fuel ratio between the upstream and downstream measurements. To account for any mass not leaving the engine through the exhaust port, the crankcase vents were piped through a flow meter (Alicat Scientific, model: M-10SLPM-D) before venting to the atmosphere. A 60 L surge tank between the engine and flow meter was used to minimize the flow pulsations caused by the reciprocating piston motion. The blow-by gas was assumed to be pure air.

Measured flow rates were on the order of 7 SLPM, which accounts for 88% of the air/fuel ratio discrepancy. It was therefore confirmed that blow-by losses are significant. The remaining unaccounted air flow may still be through the crankcase, but might not have passed through the flow meter. The reason for this is that the meter imposes a pressure drop in the flow, causing the crankcase to pressurize by approximately 0.03 bar above atmospheric pressure. The oil seal used to prevent oil from leaking out around the spark plug wire guide, shown in Figure 6.16, was not intended to withstand pressure and air leakage was noted during these measurements. Consequently, not all crankcase blow-by was accounted for by the flow meter.

Figure 6.16: Photograph indicating leaking oil seal during blow-by tests.
Since the mass discrepancy affects the air/fuel ratio, it is assumed that the leakage must be occurring in the compression cylinder, where no fuel is present. The most likely candidate in this case would be the piston rings, which may not be sealing properly due to an overall temperature reduction in the absence of combustion. This theory correlates well with a lower-than-expected peak cylinder pressure when compared to the numerical engine model, discussed further in Chapter 7. The use of a smaller piston ring end-gap might be necessary, should this be confirmed as the problem. The crossover valve stem seals were also considered as a possible source of leakage, since they were originally designed by the manufacturer to be used at considerably lower pressures (<3 bar) than those present in the split-cycle engine (∼17 bar). It would be expected, however, that both fuel and air would be leaking from this location and the state of mixing would ultimately determine any effect on the air/fuel ratio. Measuring the lower crankcase blow-by separately from the cylinder head would be one way of narrowing the leakage source, but this is not easily accomplished due to connecting passages that are necessary for pushrod operation and oil drain-back. In summary, roughly 5–7% of measured intake air does not reach the combustion chamber and, based on blow-by measurements, is thought to be leaking into the crankcase.

Without an accurate account of blow-by losses and their composition, the air/fuel ratio measured by the lambda meter could not be confirmed using air and fuel mass flow rates. However, using Equation (6.1), the air/fuel ratio was back-calculated from the measured exhaust gas composition (see Appendix D for details). Figure 6.17 shows a comparison of the equivalence ratio measured by the lambda meter with that calculated using Equation (6.1), and with that based on air and fuel mass flow rates. The exhaust gas sampling confirms the operation of the lambda meter, with a maximum percent difference in \( \phi \) of 2%, occurring at the leanest air/fuel ratio. As such, all equivalence ratios provided in the current work are based on exhaust O\(_2\) measurements taken with the UEGO sensor.

\[
\phi = \frac{2n_{O_2}}{n_P \cdot \tilde{x}_{H_2O} + n_P (1 - \tilde{x}_{H_2O})(\tilde{x}_{CO}^* + 2\tilde{x}_{CO_2}^* + 2\tilde{x}_{O_2}^* + \tilde{x}_{NO}^* + 2\tilde{x}_{NO_2}^*)}
\]  

(6.1)

where:

- \( n_{O_2} \) is the stoichiometric number of oxygen molecules
- \( n_P \) is the total number of moles in the exhaust products
- \( \tilde{x}_{H_2O} \) is the wet mole fraction of water in the exhaust products
- \( \tilde{x}_{i}^* \) is the dry mole fraction of species \( i \) in the exhaust products
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6.4.4 Lean-Limit Backfire

Since the crossover passage contains a heated and pressurized air/fuel mixture, it is not hard to envision combustion occurring in this volume under the right circumstances. Throughout the testing regime, such circumstances could be realized in two different ways: first, overly advanced spark timing can lead to the flame arriving at the crossover outlet valve before it has closed (or not closed sufficiently to quench the flame); second, near the lean limit of operation, very late burning cycles were found to cause an auto-ignition event on the following cycle while the crossover valve was fully open. The first case is easily detected by the audible noise that occurs when the burnt gases in the crossover passage, now at much higher pressure than usual, back-flow into the compression cylinder when the crossover inlet valve opens. In such cases, the engine speed will reduce significantly or even stall. The second case is much more perplexing and warrants a detailed discussion.

During a routine air/fuel ratio sweep at 1200 RPM, combustion occurred inside the crossover passage—hereafter referred to as a backfire—on approach to the lean operating limit, $\sim \phi = 0.84$. What was unusual about this scenario was that the spark timing had been fixed at 16 °CA ATDC-e during stoichiometric operation, where no backfire issues occurred. Since the reaction rate decreases under leaning conditions, the later combustion phasing was not expected to cause a typical backfire (first aforementioned scenario). Closer inspection of the data revealed combustion had actually been initiated before the spark timing.

Figure 6.17: Comparison of equivalence ratio based on measurements from: exhaust lambda meter, upstream air and fuel flow rates, and exhaust gas composition.
Pressure traces of the expansion cylinder and crossover passage are shown for the cycles leading up to, and including, the backfire in Figure 6.18. The cycle prior to the backfire, labelled “A”, shows very late combustion starting just prior to the exhaust valve opening. The following cycle, labelled “B”, exhibits a very large spike in pressure at TDC, followed by autoignition of the air/fuel mixture in both the cylinder and crossover passage. This is shown more clearly by the inlaid plot of Figure 6.18, which reveals the pressure traces for cycle “B” immediately start to rise following the spike at TDC, well before the point where the spark plug is fired. Once the crossover outlet valve had closed, combustion occurring inside the crossover passage caused the pressure to rise rapidly on account of its fixed volume. The pressure quickly exceeded the 50 bar range of the sensor, as indicated in the figure.

![Figure 6.18: Example of combustion and crossover pressure traces leading up to and during backfire event.](image)

The exact source of ignition causing the backfire is not immediately clear. Only two cases of such an event were recorded, approximately 100 cycles apart from one another. Both exhibit the same very late onset of combustion in the preceding cycle, and showed similar pressure spikes at TDC-e. The spikes are believed to be artificial (i.e. noise), based on their large magnitude (>60 bar) and short duration (0.3°CA). Given both spikes occur at precisely TDC-e, it is suspected that it may be caused by mechanical contact with the piston. The recurrence of a very late burning cycle preceding the backfire might suggest the piston is contacting an over-heating exhaust valve that is not seating properly. This is supported by the fact that this type of backfire only happened at 1200 RPM and near the lean operating limit; conditions that correspond to longer combustion durations and
an increased number of late burning cycles (see Section 6.5), resulting in higher EGTs and a hotter exhaust valve. Furthermore, for late burning cycles like the one in Figure 6.18, the open exhaust valve is exposed to combustion temperatures (>1800 K) without the use of its seat for cooling. Based on these observations, it can be speculated that the cause of ignition was from the brief mechanical contact between the piston and an over-heated exhaust valve.

A single backfire occurrence is unlikely to cause engine damage, since the resulting increase in temperature and pressure are short in duration. However, as previously mentioned, this type of backfire occurred twice in short succession during a single test. The second time, the engine did not return to normal operation and had to be shut down. Figure 6.19 shows the temperature profile of the crossover passage and exhaust gas as a function of time, for the entire test duration. The first backfire occurs at cycle 36, indicated by the sudden temperature rise within the crossover passage. The engine then misfires until the exhaust products have been purged from the crossover passage. At cycle 144 the second backfire occurs, but this time the crossover temperature did not return to normal and the engine was shut down shortly thereafter. The reason the crossover temperature kept rising was due to combustion being sustained inside the passage. Hot exhaust products from the first backfire, mixed with the next-cycle’s incoming air from the compression cylinder, proved to be sufficient for autoignition of the fuel injection event. This was verified by looking at the crossover pressure traces for the cycles following the backfire, an example of which is given in Figure 6.20. It can be seen that approximately 60°CA after the end of fuel injection a distinct rise in crossover pressure occurs. The fact that both crossover valves are closed at this point in the cycle is a clear indication that combustion is taking place. The magnitude of the pressure change is small, because the charge is highly diluted by exhaust products, and the fuelling rate is reduced by the higher-than-normal crossover pressure.

The discovery of this backfire mode had no direct repercussions on the testing regime. Measurements were still successfully taken for equivalence ratios down to $\phi = 0.83$ for all engine speeds listed in Table 6.1. It is, however, an interesting discovery that shows the vulnerability of premixing the air and fuel inside the crossover passage. Further investigation is required to verify the suggested cause and determine the extent of conditions under which this mode of backfire is possible.
Figure 6.19: Temperature profiles of the crossover passage and exhaust gases during successive backfire events.

Figure 6.20: Crossover pressure trace showing sustained autoignition within the passage.
6.5 Combustion Characteristics

This section provides an in-depth analysis of the combustion process in terms of rate (Section 6.5.1), phasing (Section 6.5.2), and variability (Section 6.5.3). Where applicable, comparisons to other SI engines are made to provide the reader with a relative frame of reference.

6.5.1 Rate of Combustion

The rate or duration of combustion was one of the key elements that this research was targeted to address. It was hypothesized that the levels of turbulence generated by the fluid transfer from the crossover passage into the combustion chamber would significantly decrease the amount of time required to complete combustion. This section aims to answer this hypothesis by looking at the trends in burn duration over spark timing and equivalence ratio changes, and then comparing these results to data available in the literature.

Figure 6.21 shows the average crank angle duration required to burn 90% of the air/fuel mixture, corresponding to the spark timings listed in the figure and stoichiometric fuelling conditions. Each bar in the plot has been split into two parts: the interval for early flame development (10% MFB or CA\textsubscript{0-10}), and the interval for main combustion (10–90% MFB or CA\textsubscript{10-90}). The bars are stacked, so the overall height is CA\textsubscript{0-90}. Both the modified Rassweiler and Withrow (MRW) method and the normalized pressure ratio (\(PR_N\)) method are included in Figure 6.21. A comparison of these MFB calculation methods was given in Section 5.2.4, and the MRW method is only shown again here to reinstate the fact that both methods yield approximately the same overall duration for CA\textsubscript{0-90}; however, the proportion of early flame development, CA\textsubscript{0-10}, for the MRW method is incorrectly reducing with engine speed due to the increasing gap between ignition and XOVC timing. For this reason, all MFB data presented in this section is based on the \(PR_N\) method.

An additional purpose of Figure 6.21 is to show that the burn durations do increase marginally with engine speed, and that CA\textsubscript{0-10} has approximately the same magnitude as CA\textsubscript{10-90}. In other words, it takes the same amount of time (or °CA) for the first 10% of combustion as it does for the next 80%. Based on the work of Olsson et al. [109], this appears to be a trait of fast burning combustion chambers; slower burning chambers exhibited CA\textsubscript{10-90} values greater than CA\textsubscript{0-10}. The relatively small change in duration with engine RPM is expected, now that turbulence generation has been decoupled from the mean piston speed. To account for the different spark timings used at each engine speed, the effects of ignition timing on burn duration will now be discussed.
Figure 6.21: Stacked bar plot showing breakdown of CA$_{0-90}$ into early flame development period (CA$_{0-10}$) and main burn duration (CA$_{10-90}$). Modified Rassweiler and Withrow (MRW) calculation method has been included to show error in CA$_{0-10}$.

Figures 6.22 and 6.23 show the early flame development period, CA$_{0-10}$, and the main combustion duration, CA$_{10-90}$, respectively, as a function of spark timing. Both intervals of combustion decrease with an advance in spark timing, which can be attributed to a higher gas temperature at the time of ignition. This is because combustion is being initiated after TDC, and the pressure and temperature in the cylinder are dropping as the volume increases. The change in spark timing has a more pronounced effect on CA$_{0-10}$ than CA$_{10-90}$, which is apparent from the steeper slope(s) in Figure 6.22. The unburned gas temperature is again the likely cause, as the flame kernel and early flame are more susceptible to temperature changes and quenching compared with a fully developed flame.

For the fixed spark timing of 20°CA ATDC-e (where all three engine speeds overlap) it can be seen that the 1200 RPM case has a noticeably longer flame development period and main burn duration period compared to the lower engine speeds. The exact cause of this increase is unknown, but there are two things to note for the 1200 RPM case: the volumetric efficiency is lower, but scales proportionately with the other engine speeds and is therefore unlikely to be the driving factor; and the unburned gas pressure and temperature are higher, but the estimated difference in flame speed is negligible. By process of elimination, this leads the author to believe there may be a difference in turbulence intensity, and/or bulk flow in the vicinity of the spark plug at the time of ignition. This physically makes sense, given that the time between cylinder filling and ignition decreases with higher engine speed. It is therefore conceivable that the 1200 RPM case has a higher turbulence intensity (less...
Figure 6.22: Effect of spark timing on the average early flame development period ($CA_{0-10}$). $\phi = 1$, WOT.

Figure 6.23: Effect of spark timing on the average main combustion duration period ($CA_{10-90}$). $\phi = 1$, WOT.
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time for decay) and/or greater bulk flow velocities. In either case, this could negatively affect the flame kernel development if the kernel were to experience partial quenching.

On a time scale basis, the 1000 RPM case actually has a shorter flame development period, by ∼0.2 ms, compared to the 850 RPM case. In contrast, the flame development period of the 1200 RPM case increased by 0.2 ms compared to the 850 RPM case. This might be an indication that the conditions for accelerated flame growth are reaching a point of maximum return with respect to increasing turbulence intensity. The levelling-off in CA_{0–10} with advanced spark timing, for both the 850 RPM and the 1000 RPM cases would seemingly support this theory.

Figures 6.24 and 6.25 are plots of the average CA_{0–10} and CA_{10–90} combustion durations as a function of equivalence ratio, respectively. Both of these figures show very little change in duration under lean conditions down to approximately φ = 0.9. However, as the air/fuel ratio is made leaner than φ = 0.9, a simultaneous, non-linear increase in both CA_{0–10} and CA_{10–90} occurs until the lean operating limit is reached at φ = 0.83. By this point, both durations have increased by roughly 4–6°CA or around 50%. The ability for the combustion duration to be unaffected by increasingly dilute conditions is a characteristic of fast burning engines [70]. While this is only exhibited by the split-cycle engine down to φ = 0.9, it does represent a significant portion of its lean operating range. Mixtures leaner than φ = 0.9 begin to show combustion instabilities, which will be discussed further in Section 6.5.3.

In Figures 6.24 and 6.25, the trend between the average early flame development period and the average main combustion period are visually similar. By examining the CA_{0–10} and CA_{10–90} durations on a cycle-by-cycle basis, it was discovered that the same correlation exists on an individual-cycle level. In other words, a flame that is slow to develop during the early stages of combustion, was found to have on average a longer bulk combustion duration. Figure 6.26 shows the main burn duration (CA_{10–90}) as a function of the early flame development period (CA_{0–10}) for an arbitrarily selected dataset of 300 consecutive cycles. It can be seen that a medium-to-high correlation exists between the early flame development duration and the main burn duration. In fact, for all datasets the correlation coefficients ranged from 0.5 to 0.8, and showed no clear indication of a dependence on spark timing or equivalence ratio. Ultimately this makes complete sense, since a longer flame development period forces bulk combustion to occur later in the cycle, when the cylinder volume is larger and thus the temperature is lower, effectively reducing the combustion reaction rates. The importance of rapid flame development for the split-cycle engine is now obvious, since a longer flame development period is compounded by an increase in CA_{10–90}.
Figure 6.24: Effect of air-fuel equivalence ratio on the average early flame development period (CA$_{0-10}$).

Figure 6.25: Effect of air-fuel equivalence ratio on the average main burn duration (CA$_{10-90}$).
Figures 6.27, 6.28, and 6.29 each show three expansion cylinder pressure traces and the corresponding MFB curves at engine speeds of 850, 1000, and 1200 RPM. The three plots in each figure correspond to the slowest, fastest, and ensemble-averaged cycles at stoichiometric conditions and for the spark timing indicated in the figure. In identical graphical format, Figures 6.30, 6.31, and 6.32 show the same curves except for lean operating conditions ($\phi = 0.85$). The ignition timing for these cases has been advanced slightly to maximize load. A summary of the data for Figures 6.27 to 6.32 is provided in Table 6.4. Selection of the fastest and slowest cycles was based on $CA_{0-10}$ values.

Table 6.4: Summary of combustion durations, decomposed into $CA_{0-10}$ and $CA_{10-90}$, for plots shown in Figures 6.27 to 6.32.

<table>
<thead>
<tr>
<th>$\phi$</th>
<th>RPM</th>
<th>$\theta_{ign}$</th>
<th>Fastest $CA_{0-10}$</th>
<th>Fastest $CA_{10-90}$</th>
<th>Average $CA_{0-10}$</th>
<th>Average $CA_{10-90}$</th>
<th>Slowest $CA_{0-10}$</th>
<th>Slowest $CA_{10-90}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.00</td>
<td>850</td>
<td>20</td>
<td>8.6</td>
<td>8.9</td>
<td>10.4</td>
<td>12.5</td>
<td>18.4</td>
<td>20.0</td>
</tr>
<tr>
<td></td>
<td>1000</td>
<td>18</td>
<td>9.0</td>
<td>9.7</td>
<td>11.5</td>
<td>12.0</td>
<td>23.2</td>
<td>19.6</td>
</tr>
<tr>
<td></td>
<td>1200</td>
<td>16</td>
<td>8.7</td>
<td>9.6</td>
<td>11.5</td>
<td>11.8</td>
<td>33.4</td>
<td>23.0</td>
</tr>
<tr>
<td>0.85</td>
<td>850</td>
<td>19</td>
<td>11.6</td>
<td>10.2</td>
<td>14.9</td>
<td>14.1</td>
<td>33.0</td>
<td>37.1</td>
</tr>
<tr>
<td></td>
<td>1000</td>
<td>17</td>
<td>11.2</td>
<td>10.1</td>
<td>15.7</td>
<td>15.0</td>
<td>65.3</td>
<td>32.2</td>
</tr>
<tr>
<td></td>
<td>1200</td>
<td>15.5</td>
<td>10.9</td>
<td>10.2</td>
<td>14.9</td>
<td>14.1</td>
<td>41.6</td>
<td>37.1</td>
</tr>
</tbody>
</table>

* All combustion durations have units of °CA. $\theta_{ign}$ has units of °CA ATDC-e.
Figure 6.27: Comparison of expansion cylinder pressure and mass fraction burned (MFB) profiles, for the fastest and slowest burning cycles in a 300 consecutive-cycle data set, with the ensemble averaged data. Engine speed = 850 RPM.

Figure 6.28: Comparison of expansion cylinder pressure and mass fraction burned (MFB) profiles, for the fastest and slowest burning cycles in a 300 consecutive-cycle data set, with the ensemble averaged data. Engine speed = 1000 RPM.
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Figure 6.29: Comparison of expansion cylinder pressure and mass fraction burned (MFB) profiles, for the fastest and slowest burning cycles in a 300 consecutive-cycle data set, with the ensemble averaged data. Engine speed = 1200 RPM.

Figure 6.30: Comparison of expansion cylinder pressure and mass fraction burned (MFB) profiles, for the fastest and slowest burning cycles in a 300 consecutive-cycle data set, with the ensemble averaged data. Engine speed = 850 RPM.
**Figure 6.31:** Comparison of expansion cylinder pressure and mass fraction burned (MFB) profiles, for the fastest and slowest burning cycles in a 300 consecutive-cycle data set, with the ensemble averaged data. Engine speed = **1000 RPM**.

**Figure 6.32:** Comparison of expansion cylinder pressure and mass fraction burned (MFB) profiles, for the fastest and slowest burning cycles in a 300 consecutive-cycle data set, with the ensemble averaged data. Engine speed = **1200 RPM**.
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The net IMEP has been indicated on each curve in Figures 6.27 to 6.32, and the reader may notice that in Figures 6.27 and 6.29 the fastest burning cycle has a lower IMEP than the ensemble-averaged curve. This is due to the crossover outlet valve being pushed open, as discussed in Section 6.4. It is also the cause of the oscillatory behaviour in the MFB curve as it is approaching unity. These two pressure curves also have a much sharper peak than the others, caused by the sudden opening of the valve, which abruptly halts the rising cylinder pressure. For these cycles, some error in the MFB curve is to be expected. However, the most advanced ignition timing used in this work was selected such that only a few cycles out of 300 would exhibit this behaviour. Furthermore, combustion is generally 80% complete at the LPP, and since the main burn duration omits the last 10% of combustion, the error is assumed to be negligible for these measures.

Figures 6.27 through 6.32 are meant to show the large variation in the rate of combustion for a given operating condition. This cannot be portrayed through average metrics, which are represented by the ensemble-averaged curves in Figures 6.27 through 6.32. Although, the reader should be reminded that ensemble-average curves were not used to produce average combustion rates and other metrics. What is apparent from these figures is how the progression of combustion can be significantly different for two separate cycles under the same nominal operating conditions. For the stoichiometric cases, the shift in CA$_{50}$ between the fastest and slowest burning cases ranges from 17–34°CA, representing a percent difference of 41–72%. The difference in CA$_{0-10}$ was more consistent, ranging from 9.9–13.4°CA, but still represents a percent difference of 68–82%. The most noticeable difference between the fastest and slowest cycles, however, is the early flame development period, CA$_{0-10}$, within which the first few percent seem to dictate the remainder of combustion. The difference between slow and fast CA$_{0-10}$ durations ranged from 9.8–24.7°CA, which is a percent difference of 73–117%. The fastest cycles show detectable pressure rises within 5°CA from the time of ignition, whereas the slowest cycles have a prolonged period with almost no pressure rise. Since ignition occurs well after TDC, the expansion of the cylinder volume means the unburned gas pressure and temperature are reduced with any delay in the bulk heat release. As a result, the slowest cycles may not produce peak combustion pressures that surpass the average crossover passage pressure. With increased engine speed, the rate of cylinder volume change with time ($dV/dt$) also increases, and the effect of a slow burning cycle on peak combustion pressure is even more pronounced.

Despite the increase in average burn duration under lean air/fuel ratios (see Figures 6.24 and 6.25), the fastest lean-burning cycles are generally only 2–3°CA slower than the fastest stoichiometric cycles for the same operating conditions. They are still much quicker than the slowest cycles at stoichiometric air/fuel ratios (see Table 6.4). The opposite is true for
the slowest cycles operating at lean equivalence ratios, which are significantly slower than those at a stoichiometric equivalence ratio. For the 1000 RPM case shown in Figure 6.31, the early flame development period of the slowest cycle is so long that the work produced by combustion is barely sufficient to overcome the compression work. For all practical purposes, this particular cycle could be considered a misfire, despite measurable combustion occurring.

It is interesting to hypothetically interpret what is happening during this extended flame development period. By the time 10% of the cylinder mass has burned, about 65 °CA has surpassed since spark ignition occurred. With no other sources of ignition available, a flame must have developed and subsequently been subjected to unfavourable conditions for rapid growth. Presumably, at the molecular level, the production rate of reaction-evoking radicals must be closely balanced by terminating reactions (e.g. quenching). Thus, a stagnation in combustion transpires until later in the cycle, when perhaps the turbulence has decayed sufficiently for stable flame propagation.

The burn durations shown in Figures 6.27 through 6.32 and listed in Table 6.4 suggest that the ensemble-averaged values are skewed towards the fastest cycles, rather than the slowest. To further examine this possibility, the frequency distributions of CA$_{0-90}$ for Figures 6.27 to 6.32 have been plotted in Figure 6.33, along with the average value of each data set, indicated by the dashed line. Note that the slowest cycle from Figure 6.31 is not shown in Figure 6.33 to allow for a shorter x-axis domain and an enhanced view of the plot.

Figure 6.33 shows that the data is positively skewed for each one of the cases shown. In fact, all analysed data sets contained CA$_{0-90}$ frequency distributions that had skewness values greater than one. In a practical sense, this means the average burn duration value reported has been reduced by the slow, outlying cycles, of a non-uniform data distribution. Furthermore, the skewness is generally increasing under lean conditions, as the variability in burn duration increases. Thus, it can be stated that the mean burn duration value under-predicts that which is typical for the majority of engine cycles. This is true for all operating conditions, especially at lean air/fuel ratios.

**Comparison with Literature**

The CA$_{0-10}$ and CA$_{10-90}$ burn durations can be put into perspective by comparison to other real-world engine values. To do so, a comprehensive list of peer-reviewed papers regarding empirical combustion rates in CNG-fuelled engines has been assembled by the author and summarized in Table 6.5. The engine geometry and test conditions are also
Figure 6.33: Frequency distributions of CA$_{0-90}$ burn durations for Figures 6.27 to 6.32.
included (if available), as these parameters generally influence combustion. Durations were only assessed for equivalence ratios ranging from $\phi = 0.8$ to $\phi = 1.0$.

The published combustion durations shown in Table 6.5 vary substantially, from 14°CA to 42°CA for $CA_{0-10}$ and 7°CA to 75°CA for $CA_{10-90}$, although only half of the papers reviewed in the table indicated the former. At first glance the split-cycle engine appears to have very quick combustion in comparison with its peers, recalling the average values of 11.5–14.9°CA and 11.8–14.1°CA for $CA_{0-10}$ and $CA_{10-90}$, respectively, at 1200 RPM. However, all of the engines listed in Table 6.5 have larger displacements than the 0.24 L split-cycle engine, presumably meaning a greater amount of mass is being burned over the indicated crank angle range. The relationship between engine size and MFB duration is not clear, and may be irrelevant since the shortest combustion durations appear to arise with the largest engine displacements (references [109], [138], and [122]). It is worth noting, however, that all three of these references are utilizing the same engine at the University of Lund, and provide substantially quicker burn durations compared to the remaining cases in the table. The combustion chambers used in references [138] and [122] were the fastest burning designs taken from Olsson and Johansson [109], who investigated seven different chambers—all of which had high squish-to-bore ratios, with the exception of one flat (pancake) chamber. The latter corresponds to the longest combustion duration values given for the range in [109], and is perhaps the better comparison since the split-cycle also uses a pancake chamber.

Overall, the $CA_{0-90}$ combustion duration of the split-cycle engine is on par with the fastest values reported in Table 6.5, and it is approximately four times faster than the slowest durations. It should also be realized that all comparative cases, with the exception of [53], have higher compression ratios, and the fastest cases correspond to engines that use forced induction. Both of these characteristics lead to higher cylinder temperatures, which favourably affects the chemical reaction rates. Furthermore, the engine loads used in the comparison cases are generally much higher than those achieved by the split-cycle engine, and Baratta et al. [16] show that higher loads can significantly decrease the burn duration. Finally, the cases reported in Table 6.5 are operating at MBT conditions, which puts the LPP around 15°CA ATDC. In the split-cycle engine, combustion is phased much later, such that the LPP is $>30$°CA ATDC, where the cylinder volume is more rapidly expanding. In a conventional SI engine, these conditions are not conducive to a fast burn, and are a further testament to the significantly quick burn rates that have been realized in this research.
Table 6.5: Summary of literary findings on MFB durations and associated test conditions, for CNG-fuelled engines.

<table>
<thead>
<tr>
<th></th>
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<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
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<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference</td>
<td>[53]</td>
<td>[95]</td>
<td>[10]</td>
<td>[143]</td>
<td>[109]</td>
<td>[138]</td>
<td>[122]</td>
<td>[16]</td>
<td></td>
</tr>
<tr>
<td>Bore (mm)</td>
<td>80.3</td>
<td>105.0</td>
<td>92.3</td>
<td>100.0</td>
<td>120.7</td>
<td>120.7</td>
<td>120.7</td>
<td>72.0</td>
<td>67.0</td>
</tr>
<tr>
<td>Stroke (mm)</td>
<td>88.9</td>
<td>120.0</td>
<td>99.3</td>
<td>115.0</td>
<td>140.0</td>
<td>140.0</td>
<td>140.0</td>
<td>84</td>
<td>68.0</td>
</tr>
<tr>
<td>Displacement (L)</td>
<td>0.45</td>
<td>1.04</td>
<td>0.66</td>
<td>0.9</td>
<td>1.6</td>
<td>1.6</td>
<td>1.6</td>
<td>0.34</td>
<td>0.24</td>
</tr>
<tr>
<td>Comp. Ratio</td>
<td>8.3</td>
<td>10.5</td>
<td>10.3</td>
<td>12.0</td>
<td>12.0</td>
<td>12.0</td>
<td>11.8</td>
<td>9.8</td>
<td>8.5</td>
</tr>
<tr>
<td>CH₄ in Fuel (%V)</td>
<td>96.4</td>
<td></td>
<td>91.5</td>
<td>96.0</td>
<td>91.1 [75]</td>
<td>88</td>
<td>-</td>
<td>98</td>
<td>100</td>
</tr>
<tr>
<td>Engine RPM</td>
<td>1000</td>
<td>1200</td>
<td>1500</td>
<td>1200</td>
<td>1200</td>
<td>1200</td>
<td>1200</td>
<td>2000</td>
<td>850-1200</td>
</tr>
<tr>
<td>BMEP (bar)</td>
<td>5.0</td>
<td>-</td>
<td>7.9</td>
<td>1.4-6.3</td>
<td>-</td>
<td>5.0-8.0</td>
<td>7.0</td>
<td>2.0-7.9</td>
<td>4.0</td>
</tr>
<tr>
<td>Comb. Chamber</td>
<td>Bathtub</td>
<td>-</td>
<td>Pent-roof BIP</td>
<td>Various BIP</td>
<td>BIP</td>
<td>BIP</td>
<td>BIP</td>
<td>Pent-roof Pancake</td>
<td></td>
</tr>
<tr>
<td>Air Induction</td>
<td>NA³</td>
<td>NA</td>
<td>NA</td>
<td>NA</td>
<td>TC⁴</td>
<td>TC</td>
<td>TC</td>
<td>TC</td>
<td>NA</td>
</tr>
<tr>
<td>CA₀-₁₀ (deg)</td>
<td>30-42</td>
<td>42-45⁵</td>
<td>38</td>
<td>-</td>
<td>14-18</td>
<td>-</td>
<td>-</td>
<td>19-27</td>
<td>10-13</td>
</tr>
<tr>
<td>CA₁₀-₉₀ (deg)</td>
<td>48-75</td>
<td></td>
<td>50</td>
<td>48-58</td>
<td>7-22</td>
<td>10-18</td>
<td>16-18</td>
<td>26-29</td>
<td>11-12</td>
</tr>
</tbody>
</table>

¹ Approximated compression ratio.
² BIP = Bowl-in-Piston.
³ NA = Naturally Aspirated.
⁴ TC = Turbocharged.
⁵ CA₀-₉₈.
6.5.2 Combustion Phasing

Combustion phasing is characterised by the crank angle location of peak pressure (LPP) or value of 50% MFB (CA\textsubscript{50}) for a given engine cycle or dataset. In the case of the split-cycle engine, the LPP refers to the maximum \textit{combustion} pressure, and not \textit{cylinder} pressure, since late combustion may fail to produce a pressure rise that exceeds the motoring pressure at TDC. A conventional SI engine operating at MBT conditions will generally have a spark timing in advance of TDC, such that the LPP falls between 14–16°CA ATDC. This is true for all SI engines, regardless of engine speed or load [70, 116, 153]. By comparison, combustion phasing in the split-cycle engine is inherently late, as shown in Figure 6.34.

![Figure 6.34: Typical SI engine combustion phasing compared with split-cycle phasing.](image)

The late combustion phasing in the split-cycle engine is required to provide adequate time for crossover outlet valve closure (XOVC), preventing combustion from propagating back into the crossover passage. Currently, the nominal XOVC time is 27°CA ATDC-e, and the flame development period was shown to be on the order of 10–13°CA; therefore, it can be expected that the earliest viable spark timing would be in the vicinity of 15°CA ATDC-e. In reality, ignition timing can probably occur a few degrees earlier still, since the crossover valve is a finite distance from the spark plug, and given the likelihood of a flame being quenched when the valve is at low lift heights. Initiating combustion too early also has implications for the trapped cylinder mass, which will be discussed in Section 7.4.3.

Even though MBT timing was never achieved in this work, Figure 6.34 implies that an additional spark advance of 5°CA will still phase the LPP approximately 10°CA later than...
a conventional SI engine. Therefore, the late combustion phasing of a generic split-cycle engine should not be seen as defect, but more as a necessary operational characteristic. In this work, the phasing can be considered overly late, as evidenced by the drop in cylinder pressure prior to the main heat release, which is a limitation imposed by the crossover outlet valve (see Section 6.4).

For all conditions investigated, the average LPP ranged from approximately 37°CA to 60°CA, and was found to be affected by engine speed, equivalence ratio, and spark timing—the latter of which has already been demonstrated in Figure 6.34. The extent to which these variables affect combustion phasing will now be discussed.

Figure 6.35 shows the effect of spark timing on the LPP under stoichiometric fuelling conditions. At the later ignition timings for each engine speed case, the LPP is shown to advance by roughly 3–3.5°CA for every 1°CA of spark timing advance. The 850 RPM and 1000 RPM cases indicate a decay in this trend with further spark timing advances; however, the 1200 RPM case maintains a linear relationship. With the limited spark timing range available, it is unclear if this difference at 1200 RPM is significant. Regardless, the additional phase shift in LPP (beyond the physical change in spark timing) correlates well with the decrease in overall burn duration as the ignition timing is advanced (shown in Figures 6.22 and 6.23). As previously stated in the analysis of these figures, the cause is likely the higher in-cylinder temperatures present at earlier spark timings, and the strong dependence that reaction rates have on temperature.

For a given spark timing, the LPP is retarded as engine speed is increased. This is also thought to be caused by changes in burn duration, both CA\(_{0−10}\) and CA\(_{10−90}\), which were observed to be slightly longer at faster engine speeds (see Figure 6.21). The larger difference in LPP between the 1000 RPM and 1200 RPM cases versus the 850 RPM and 1000 RPM cases, was also found in the burn durations (see Figures 6.22 and 6.23). It is expected that further advances in phasing would be possible with earlier spark timing, once the crossover valve issue has been solved.

For lean air/fuel ratios, combustion phasing is relatively constant between \(\phi = 1\) and \(\phi = 0.9\), as shown in Figure 6.36. Conditions leaner than \(\phi = 0.9\) begin to retard the LPP, which is consistent with the changes in burn duration, shown previously in Figures 6.24 and 6.25. From \(\phi = 1\) to \(\phi = 0.83\), the average LPP was retarded by approximately 9°CA for all engine speeds. The differences in LPP between engine speeds, for a given equivalence ratio, is accounted for by the different spark timings used, as labelled in Figure 6.36.
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Figure 6.35: Phasing of average peak combustion pressure as a function of spark timing.

Figure 6.36: Phasing of average peak combustion pressure as a function of air/fuel equivalence ratio.
The MFB, expansion cylinder pressure, and pressure rise rate (PRR) curves are overlain in Figure 6.37 for an arbitrarily selected engine cycle. It can be seen that the maximum PRR is achieved several degrees in advance of CA$_{50}$, and the LPP occurs when approximately 80% of the fuel has been burned. The combustion reaction is complete shortly after the peak in cylinder pressure, as indicated by the MFB curve reaching unity. The relative phrasing between these characteristics is typical for a conventional SI engine [70], and implies no major anomalies exist in the split-cycle combustion process.

![Figure 6.37: Example of phasing between the expansion cylinder pressure, the pressure rise rate (PRR), and the mass fraction burned (MFB).](image)

The individual-cycle relationship between the maximum combustion pressure, $p_{max}$, and its corresponding crank angle location, or LPP, is shown in Figure 6.38. All three engine speeds show an expected increase in $p_{max}$ as combustion phasing moves closer to TDC. Each engine speed exhibits roughly the same upper and lower data limits, but with increasing RPM there are a greater number of cycles with a higher $p_{max}$ and earlier LPP. In part, this is due to the advanced spark timings used at higher RPM, but, for the same LPP, $p_{max}$ is also increasing with engine speed. The cause of this increase can be attributed to a higher crossover pressure, which results in a greater pre-combustion cylinder pressure.

Spark timing also affects the LPP, as shown previously in Figure 6.35, and while the earliest spark timings do indeed populate the upper end of the peak pressure trend in Figure 6.38, and vice-versa, a large amount of overlap exists between cycles with different spark timings. This cycle-to-cycle variation has already been witnessed in the discussion on burn duration (Section 6.5.1) and its implications on the repeatability of combustion will now be discussed.
Figure 6.38: Individual-cycle maximum combustion pressure versus the crank angle at which it occurs, known as the location of peak pressure (LPP). Three different spark timings are shown for each engine speed. All data is for stoichiometric operation.
6.5.3 Cyclic Variability and Lean Operating Limit

The COV\textsubscript{IMEP} and COV\textsubscript{LPP} are shown together in Figure 6.39 as a function of spark timing. A parallel trend between the variation in IMEP and LPP can be seen, although the magnitude of the COV\textsubscript{LPP} is 2–3 times that of COV\textsubscript{IMEP}. Generally, the COVs are reducing as ignition timing is advanced, indicating that combustion stability is improving with earlier phasing. This was expected since the charge temperature will be higher, and the rate of expansion \( \frac{dV}{d\theta} \) lower, as ignition timing is advanced towards TDC. The COV levels at these early spark timings are within the normal range of a typical SI engine [131].

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure6.39.png}
\caption{COV\textsubscript{IMEP} and COV\textsubscript{LPP} as a function of spark timing.}
\end{figure}

At 20 °CA spark timing, where all engine speeds have overlapping data points in Figure 6.39, it is evident that the instability of the 1200 RPM case is disproportionately high. A similar jump in the data between the 1000 RPM and the 1200 RPM case for the early flame development duration was seen in Figure 6.22 of Section 6.5.1, and the overall delay in combustion phasing is the root cause of these higher variations in LPP and IMEP.

For lean air/fuel ratios, the COV\textsubscript{IMEP} and COV\textsubscript{LPP} both increase in a non-linear fashion, as shown in Figures 6.40 and 6.41, respectively. Similar to the trends seen in combustion duration (Figures 6.24 and 6.25) and combustion phasing (Figure 6.36) with changes in equivalence ratio, the combustion stability is negligibly affected by lean air/fuel ratios down to approximately \( \phi = 0.9 \). Further leaning out the charge leads to rapidly destabilizing conditions, until the lean operational limit is reached at \( \phi = 0.83 \).
When comparing the lean operating limit with the lean flammability limit of methane in standard air ($\phi = 0.53$) [117], or the values typically realised in conventional NG engine research ($\sim \phi = 0.55–0.65$) [9, 35, 40, 75, 80, 138], it is evident that the current value is relatively high. This may be in part due to the definition used for the lean operating limit, which was subjectively defined as the point when a stable air/fuel ratio measurement could no longer be realized. Had the blow-by issue not been present (see Section 6.4), the air and fuel mass flow rates could have been used to acquire data for leaner equivalence ratios. However, the misfire rate below $\phi = 0.82$ was exceeding approximately 3%, and the increasing level of misfires is proof that the conditions required to sustain a chemical reaction were not being met. Based on the physical evidence that the mixture is well homogenised (see Section 6.3.3), the most important remaining factors for combustion stability are temperature and turbulence. Thus, it can be postulated that the relatively high lean operating limit is a consequence of low in-cylinder temperature, caused by post-compression heat losses in the crossover passage and during transfer into the expansion cylinder, combined with high flame strain rates that are a consequence of the turbulent flow conditions.

Thus far, the measures of stability have been focused on cycle-averaged parameters (e.g. IMEP, LPP), which are good at evaluating cyclic variations, but do not provide any indication of a possible relationship between cycles. The two-stroke operating nature of the split-cycle engine means both the exhaust and filling processes of the expansion cylinder must occur, at least partially, in the same stroke. A large overlap of the cylinder in-flow and out-flow processes is undesirable from an emissions stand-point, since raw fuel is likely to end up in the exhaust stream. The current engine configuration requires the exhaust valve to close at approximately 12°CA BTDC-e (see Table 6.3 in Section 6.4 for all timing specifications). As a consequence, an estimated 3–4% of combustion products are undoubtedly trapped in the cylinder at the end of each exhaust stroke. Depending on the combustion characteristics of the previous cycle, the composition and state properties of these residuals may vary significantly. To examine how cyclic variations affect stability of combustion on an individual-cycle basis, the IMEP of a given cycle, $i$, has been plotted against the following cycle, $i + 1$, in Figure 6.42. The plots on the left-hand side are for stoichiometric air/fuel ratios, and those on the right-hand side are for cycles operating at $\phi = 0.85$.

The close grouping of points for the 850 RPM, stoichiometric case (top left) indicates no inter-cycle correlation exists, and that fluctuation in IMEP (combustion stability) is not related to the previous cycle. With increasing engine speed, a few rogue cycles begin to indicate that average-valued cycles are followed by exceptionally low IMEP cycles, and vice-versa. In fact, what the data is showing is that low IMEP cycles are generally surrounded by regular cycles. In other words, back-to-back low IMEP cycles are very uncommon. This
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Figure 6.40: Increase in COV\textsubscript{IMEP} with decreasing air/fuel equivalence ratio.

Figure 6.41: Increase in COV\textsubscript{LPP} with decreasing air/fuel equivalence ratio.
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Figure 6.42: Comparison of adjacent-cycle IMEP correlations between stoichiometric ($\phi = 1$) and lean ($\phi = 0.85$) burn conditions.
bi-directional scattering forms a near triangular shape, which is much more distributed and prevalent under lean conditions. It is postulated that the low IMEP cycles are a result of incomplete combustion, resulting in larger fractions of unburned hydrocarbons in the residual gas, creating a more readily ignitable mixture for the next cycle. Low IMEP cycles are also synonymous with longer combustion durations (Figures 6.27 to 6.32), resulting in higher EGTs for these particular cycles. This further aids the following cycle in achieving complete combustion. It can be stated that the combustion stability in this split-cycle engine is, to some extent, self-correcting; the lack of positive valve overlap, combined with a large amount of residuals, provides a direct link between the current and previous cycle.

6.6 Exhaust Gas Emissions

6.6.1 General

The volume concentrations of CO, CO$_2$, O$_2$, NO$_x$, and THC in the exhaust stream were measured using the equipment outlined in Section 4.3. All species except THC were measured on a dry basis, by method of condensation using a chiller. NO and NO$_2$ were taken collectively as NO$_x$, although the molar mass for NO has been used for all calculations.

Hydrocarbon levels were measured and recorded as ppm-C$_1$, and include both CH$_4$ and non-methane hydrocarbon (NMHC) emissions. However, throughout the testing regime NMHC levels were periodically checked by operating the analyser in dual mode, which measures THC and CH$_4$ intermittently for short periods of time to segregate the NMHCs. Data was not recorded in this mode, but it was observed over the experimentation course that the composition of HCs in the split-cycle exhaust stream is primarily CH$_4$. Since no positive valve overlap exists between the crossover outlet valve and the exhaust valve, short-circuiting of the fuel directly into the exhaust stream is unlikely, implying THC emissions are primarily from fuel not consumed during combustion (e.g. crevice volumes, flame quenching). For all operating conditions tested, NMHC emissions were consistently between 150 to 250 ppm; the unwavering nature of the NMHC emissions with the test variables leads the author to believe that they are the products of lubricating oil consumption.

For all tests utilizing stoichiometric fuelling conditions, the dry-fraction CO$_2$ content was found to be 10.0 ± 0.1%. The consistency of this measurement was expected, given the fixed air/fuel ratio and the lack of substantial changes in CO emissions. However, the absolute magnitude of CO$_2$ is low. Based on the global reaction equation for methane-air, CO$_2$ should constitute approximately 11.5% of the dry exhaust products. Rigorous
checks of the sampling line and equipment did not reveal any faults, and to date nothing conclusive has been discovered to explain this discrepancy. It is possible that the span gas concentration (8% CO$_2$ by volume) is too low for the 0–15% range being used. Any non-linearity in the measuring device will result in a calibration error. The author concedes that, as a best practice, the concentration of the span gas should be within 10% of the analyser range, which is the case for all the other measured gases. At this time, the use of a higher span gas concentration remains to be attempted.

### 6.6.2 Ignition Timing Effects

To observe the effects of spark timing on exhaust gas emissions, a spark sweep was performed with the air-fuel ratio adjusted to maintain $\phi = 1$. The sub-plots in Figure 6.43 show the resulting CO, NO$_x$, THC, and O$_2$ emissions. The CO$_2$ emissions remained stable at 10.0 ± 0.1% throughout the sweep, and are therefore not presented graphically.

It can be seen that CO varies between 1500–2000 ppm, and has no discernible increasing or decreasing trend with spark timing. CO output was found to be very sensitive to small changes in equivalence ratio throughout the spark sweep. Despite best efforts to maintain a stoichiometric mixture, cyclic variation can cause the average equivalence ratio, according to the exhaust O$_2$ sensor, to vary by as much as ±0.005 for a 300-cycle data set. This is thought to be the main reason for the scatter in CO values. The magnitude of CO emissions are very low compared with those from a gasoline-fuelled SI engine operating at stoichiometric conditions, which typically range from 5000–10000 ppm [70]. This is a good testament to the mixture uniformity, since the oxidation of CO to CO$_2$ relies on an even distribution of oxygen throughout the reaction zone.

The NO$_x$ emissions varied little with spark timing, as shown in Figure 6.43. With the exception of the later spark timings at 1200 RPM, the average NO$_x$ levels were 730±20 ppm or approximately 3–4 g/kWh. These levels are similar in magnitude to some NG engines [10, 53] and considerably lower than others [16, 40, 80, 87, 95, 109, 122]. This can be attributed to a low peak cylinder pressure/temperature, which is the result of initiating combustion after TDC. Further advances in spark timing, beyond what was achievable in the current research, is expected produce higher levels of NO$_x$ as the engine load increases. However, based on the very small change in NO$_x$ per degree of timing advance shown in Figure 6.43, it is anticipated that these increases will not be significant.

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2 Using indicated power and a mechanical efficiency, $\eta_m = 87\%$. 

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For $\phi = 1$, CO typically between 1500–2000 ppm.

- $SE_{850} = \pm 50$ ppm
- $SE_{1000} = \pm 110$ ppm
- $SE_{1200} = \pm 44$ ppm

Longer burn durations *excluded from average

- $SE_{850} = \pm 15$ ppm
- $SE_{1000} = \pm 13$ ppm
- $SE_{1200} = \pm 15$ ppm

$NO_x = 730$ ppm*

- $SE_{850} = \pm 42$ ppm
- $SE_{1000} = \pm 78$ ppm
- $SE_{1200} = \pm 69$ ppm

$THC - C_1$ vs. Spark Timing (deg. ATDC-e)

- $R^2 = 0.84$
- $SE_{850} = \pm 0.04 \%$
- $SE_{1000} = \pm 0.04 \%$
- $SE_{1200} = \pm 0.02 \%$

$O_2$ vs. Spark Timing (deg. ATDC-e)

- $R^2 = 0.88$
- $SE_{850} = \pm 0.04 \%$
- $SE_{1000} = \pm 0.04 \%$
- $SE_{1200} = \pm 0.02 \%$

Figure 6.43: Effect of spark timing on exhaust gas emissions.
For spark timing values of 18°CA, 19°CA, and 20°CA at 1200 RPM, slightly lower levels of NO$_x$ were observed. This is thought to be the combination of longer burn durations and higher rates of volumetric expansion $(dV/dt)$ at these operating conditions. Both attributes are effective in reducing peak cylinder temperatures, which is the primary facilitator for NO$_x$ production. The durations of CA$_{0-10}$ exhibit a similar jump in magnitude between 17°CA and 18°CA spark timing (see Figure 6.22), but as was previously discussed in Section 6.5.1, it is not entirely clear what causes this abrupt change.

The THCs are shown to increase with advanced spark timing by roughly 130 ppm/°CA. In part, this trend can be attributed to a reduction in post-combustion oxidation of hydrocarbon species, which occurs during the expansion and exhaust processes as crevice volume and oil-absorbed HCs are mixed back into the hot exhaust gases. For all engine speeds, the measured EGT was found to decrease by approximately 12–16°C per degree of spark timing advance (see Figure 6.7), caused by an earlier phased LPP and consequently a lower gas pressure/temperature at the time of EVO. The rate at which HC oxidation occurs in the exhaust gases is significantly dependent on temperature [70], and therefore expected to be one of the reasons for this increasing trend.

The EGTs vary substantially for different engine speeds, yet, at 20°CA spark timing, all three RPM cases have nearly identical THC levels. Therefore, either more combustion-evaded hydrocarbons are produced at faster engine speeds, but offset by greater amounts of post-oxidation from higher EGTs, or the THC formation is directly linked to the ignition timing. If the latter is true, the connection between THC levels and the crank angle location of ignition may be the combustion chamber clearance height. It is well known that flame quenching in the cold gas layer adjacent to any surface is a common source of HC emissions [72]. The extent of quenching that occurs on the piston and cylinder head surfaces, especially during the early part of combustion, may well be related to the distance between them. From 16°CA ATDC-e to 25°CA ATDC-e the minimum clearance height in the expansion cylinder changes from approximately 2.6 mm to 5 mm. While insufficient information is available to confirm causation, it is possible that the clearance height at time of ignition affects the exhaust HC levels.

In addition to the aforementioned sources, an increase in the average maximum cylinder pressure, which occurs with advanced ignition timing, will also increase the partial pressure of the HCs in the wall boundary layers. In accordance with Henry’s Law$^3$, raw fuel absorbed into the oil layer on the cylinder liner is thought to be proportional to the partial pressure of

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$^3$ The solubility fraction of a gas at a constant temperature, $x_g$, is equal to the partial pressure of the gas, $p_g$, multiplied by Henry’s law constant, $H$, or: $x_g = H \cdot p_g$. 
the fuel [33]. Since these HCs will desorb during expansion, this presents another pathway for increased HC emissions in the exhaust with advanced spark timing.

For the 1200 RPM data, a noticeable increase in THC emissions between 18°CA and 17°CA spark timing was observed, similar to that for the NO\textsubscript{x} emissions. This can again be attributed to the decrease in burn duration between these two ignition timings, which resulted in a 3.6 bar increase in the average maximum cylinder pressure (compared to 2.5 bar or less for all other 1°CA spark timing changes). The proportional drop in exhaust temperature, combined with an increase in oil-layer HC absorption are the likely causes for the jump in THC emissions.

The O\textsubscript{2} levels are also plotted in Figure 6.43, and linearly increase with advanced spark timing, at about two times the rate of THC. This makes sense since the volumetric ratio of O\textsubscript{2} to CH\textsubscript{4} in the intake charge is 2:1 at stoichiometric conditions. As discussed in Section 6.4.3, the O\textsubscript{2} levels correspond well with those indicated by the wide-band lambda sensor.

### 6.6.3 Air/Fuel Ratio Effects

To assess the emission levels under lean air/fuel ratios, a sweep of equivalence ratios was made from $\phi = 1$ to $\phi = 0.83$. The ignition timing was adjusted for maximum allowable load at stoichiometric conditions—as dictated by the crossover outlet valve—and remained fixed throughout the sweep. For engine speeds of 850 RPM, 1000 RPM, and 1200 RPM, the ignition timing was set to 20°CA, 18°CA, and 16°CA (ATDC-e), respectively. Slightly earlier spark timing could have been employed under lean conditions, on the order of 1-2°CA, but with increasing cyclic variability at lean air/fuel ratios it became difficult to detect unintended opening events of the crossover outlet valve. Such an event would affect the next-cycle gas composition, and presumably the emissions; thus, the author elected to hold the spark timing constant. Figure 6.44 shows the sub-plots of CO, NO\textsubscript{x}, THC, and CO\textsubscript{2} emissions as a function of equivalence ratio.

The CO emissions were found to reduce asymptotically as the air/fuel ratio changed from stoichiometric to lean. By $\phi = 0.96$ the levels of CO had decreased from the 1500–2000 ppm range to approximately 500 ppm, where they remained nearly constant until the lean operating limit was reached at $\phi = 0.83$. This is a characteristic trend for lean SI engine operation, because the excess oxygen in the combustion chamber ensures the majority of CO can be oxidized to CO\textsubscript{2}. Relative to typical gasoline-fuelled SI engines, these levels continue to be roughly 50% less, and in line with other SI NG engines [70, 87, 139].
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\[ SE_{850} = \pm 27 \text{ ppm} \]
\[ SE_{1000} = \pm 23 \text{ ppm} \]
\[ SE_{1200} = \pm 20 \text{ ppm} \]

\[ R^2 = 0.98 \]

\[ SE_{850} = \pm 35 \text{ ppm} \]
\[ SE_{1000} = \pm 22 \text{ ppm} \]
\[ SE_{1200} = \pm 24 \text{ ppm} \]

\[ R^2 = 0.98 \]

\[ SE_{850} = \pm 60 \text{ ppm} \]
\[ SE_{1000} = \pm 132 \text{ ppm} \]
\[ SE_{1200} = \pm 106 \text{ ppm} \]

\[ R^2 = 0.74 \]
\[ R^2 = 0.69 \]
\[ R^2 = 0.68 \]

\[ SE_{850} = \pm 0.01 \% \]
\[ SE_{1000} = \pm 0.01 \% \]
\[ SE_{1200} = \pm 0.01 \% \]

\[ R^2 = 0.99 \]

\[ \phi \]

Figure 6.44: Effect of air/fuel ratio on exhaust gas emissions.
Nitrogen oxide emissions followed the typical trend of most SI engines: increasing slightly from stoichiometric levels, reaching a peak around $\phi = 0.97$—where the combustion temperatures remain high and excess oxygen is available for NO$_x$ formation—and then proceeding to decrease until the lean operating limit was reached. At this point, the NO$_x$ levels were below 200 ppm for all engine speeds tested. Despite the lowering of IMEP under leaning conditions, the brake specific levels actually improved slightly, ranging from 1–3 g/kWh.

The THCs were found to initially decrease as the equivalence ratio was reduced from $\phi = 1$ to $\phi = 0.9$, followed by an increase for subsequently leaner mixtures. This initial decline in HC emissions with leaning of the air/fuel ratio is expected, since the fraction of fuel in the cylinder, and thus crevice volumes and oil layers, is being reduced. The increase in THC emissions for conditions leaner than $\phi = 0.9$ can be attributed to slower or partial burning cycles that significantly offset the measurement average. This was discovered by observing the THC emissions over time; they tended to have intermittent spikes that increased with frequency on approach to the lean limit. This can be explained as the result of incomplete combustion for certain cycles: as the cylinder pressure and temperature reduce with expansion, a very slow burning cycle may not completely consume all of the end gas before a diminishing reaction rate reaches the point of extinction. The remaining unburned fuel causes a sudden rise in the HC measurement over a short amount of time, which increases the mean value. Slow burning cycles are much more prevalent under lean air/fuel ratios (see Figure 6.33), hence the increase in THCs for these conditions. A complete misfire, which is not uncommon below $\phi = 0.85$, is even worse since the entire fuel charge is passed into the exhaust stream. A single misfiring cycle has the ability to significantly affect the average of its dataset. The finite time over which the exhaust measurements are recorded means the sample average is susceptible to the random nature of these slow/misfiring cycles. It is for this reason that there is considerable variation in the data below $\phi = 0.9$. The least-squares regression lines for the THC plot in Figure 6.44 are not well-suited to predict the data under these lean conditions, as indicated by the mediocre correlation coefficients.

Overall, the THC emission levels still remain relatively low throughout the air/fuel ratio sweep, and do not exceed the values measured at stoichiometric conditions. For a given equivalence ratio, the difference shown between RPM cases can be attributed to the earlier ignition timing used for faster engine speeds. The effect of spark timing on HC emissions was discussed in Section 6.6.2.
6.7 Energy Balance

The engine’s thermal efficiency is a measure of the fuel fraction being used to generate useful work, but does not indicate where the remainder of the fuel’s energy is distributed. For better understanding of the distribution pathways, an energy balance can be performed by taking the entire engine as the control volume, as shown in Figure 6.45. The inputs to the control volume are fuel, air, and shaft work; heat losses, blow-by losses, and exhaust gas are the outputs. A first-law energy balance yields Equation (6.2).

\[ \dot{m}_a h_a + \dot{m}_f h_f = Q + \dot{W}_b + \dot{m}_b h_b + \dot{m}_e h_e \]  

(6.2)

where: \( \dot{m} \) are mass flow rates
\( h \) are total specific enthalpies
\( Q \) represents all heat losses
\( \dot{W}_b \) is the brake power of the engine

The mass flow rates of air and fuel are measured quantities, along with the brake power of the engine. In this case, the brake power was replaced by indicated power due to the abnormally low mechanical efficiency of the split-cycle engine (\( \eta_m \approx 60\% \)). Because the brake torque is measured after the dynamometer belt drive system, it is expected a significant portion of the frictional losses are external to the engine. Therefore, the measured brake torque in this situation is not representative of the true shaft work of the engine.

Several assumptions were made when solving Equation (6.2):

i. All gases are ideal.

ii. The intake air is composed of O\(_2\) and N\(_2\) only.
iii. The intake air and fuel enter the engine at 298 K and 1 atm.

iv. The blow-by gas is composed of air only, and exits the engine at 298 K and 1 atm.

v. The blow-by mass flow rate is 5% of the intake air flow rate.

Based on these assumptions, the $\dot{m}_bh_b$ term can be dropped from Equation (6.2) if $\dot{m}_a$ is multiplied by 0.95, since both represent air flow rates at the same state. The exhaust mass flow rate, $\dot{m}_e$, is then simply the summation of the incoming air and fuel flow rates. Also, the $\dot{m}_ah_a$ term becomes zero, since the total enthalpy of O$_2$ and N$_2$ is zero for a reference state of 298 K and 1 atm.

To solve for the heat loss term, $\dot{Q}$, the total exhaust enthalpy must be known. This can be determined according to Equation (6.3), where $\dot{h}_f^\circ$ is the enthalpy of formation, $\dot{h}^\circ$ is the sensible enthalpy relative to 298 K and 1 atm, and $\tilde{x}_i$ is the mole fraction for each species in the exhaust stream. The exhaust mass flow rate term, $\dot{m}_e$, can then be replaced by the exhaust molar flow rate, $\dot{n}_e$, accordingly.

$$\dot{n}_eh_e = \dot{n}_e \sum \tilde{x}_i \left( \dot{h}_f^\circ + \dot{h}^\circ \right)_i \quad (6.3)$$

By separating the formation and sensible enthalpy terms, and substitution of the lower heating values (LHVs), the energy balance for Figure 6.45 can be written according to Equation (6.4). A complete derivation of this equation is can be found in Appendix D.

$$\dot{m}_fLHV_f = \dot{Q} + \dot{W} + \dot{n}_e \left( \tilde{x}_{CH_4}LHV_{CH_4} + \tilde{x}_{CO}LHV_{CO} \right) + \dot{n}_e \sum \tilde{x}_i \dot{h}_i^\circ \quad (6.4)$$

The third term on the right-hand side of Equation (6.4) represents the energy associated with incomplete combustion, and the remaining term to its right is the sensible enthalpy of the exhaust, which is based on the measured exhaust gas temperatures and the enthalpy values taken from the JANAF thermochemical tables [104]. The wet mole fractions of the exhaust were determined from the same equations used to calculate the equivalence ratio in Section 6.3.3, which can be found in Appendix D.

By solving for the heat loss term, $\dot{Q}$, the fuel distribution percentage between heat losses, shaft power, incomplete combustion, and exhaust enthalpy was obtained and plotted in Figure 6.46, with the sub-plots corresponding to 850, 1000, and 1200 RPM. The energy distribution was found to be very consistent over the range of equivalence ratios. As output power is reduced under leaning conditions, by approximately 3% overall, the sensible exhaust enthalpy is increased by the same amount. This can be attributed to the slower burn rates under lean conditions, which causes a lower specific power output and a higher
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exhaust gas temperature. The magnitude of the heat losses does not change with equivalence ratio, which may be an indication that much of the heat loss is not associated with combustion (e.g. heat transfer in the crossover passage).

The combustion efficiency was found to be very good, with the energy wasted by incomplete combustion ranging between 2.5\% at 850 RPM to 4\% at 1200 RPM. Thus, despite the flame being subjected to potentially strenuous conditions, on average more than 96\% of the fuel was oxidized on each cycle.

Of the total fuel energy input to the engine, approximately 50\% is lost to heat transfer at 850 RPM. This amount was reduced by 3–4\% as engine speed increased to 1200 RPM, converted to a 1–2\% gain in indicated power, with the remainder leaving the engine as a rise in sensible exhaust enthalpy. When compared to conventional engines, these heat transfer losses are judged to be approximately 5–10\% greater than normal [10, 70]. This includes a portion of mechanical losses within the engine that are ultimately rejected as heat. However, it needs to be emphasized that this is a rough approximation only, and underestimated exhaust enthalpies may account for at least a significant portion, if not all, of the abnormal heat losses. This is because time-averaged EGT measurements do not adequately capture the higher temperatures present during the blow-down process. Calculations based on the cylinder pressure at the time of exhaust blow-down show a deficiency in the measured EGT values by approximately 200\textdegree\text{C}. This is in line with Heywood [70], who reports that time-averaged values are approximately 100\textdegree\text{C} lower than mass-averaged values. When the energy balance calculations were re-run using measured EGTs + 200\textdegree\text{C}, the portion of sensible exhaust enthalpy increased in all cases by 10\%. Thus, without significantly increasing the temporal resolution of the EGT measurement, it is difficult to make accurate assertions with regards to the fraction of energy that leaves with the exhaust flow. The EGT calculations can be found in Appendix D.

What is certain, is that the 22–27\% indicated fuel conversion efficiency, $\eta_f$, is approximately 10–15\% lower than current SI NG engines [9, 47, 95, 139]. Since the combustion efficiency is high, the fuel is being burned, but roughly 75\% of its energy is lost to heat transfer and/or the sensible exhaust enthalpy. Both avenues are likely higher than normal. Increased heat losses can be expected from the highly turbulent flow within the engine and the extended period of time that the compressed gas is in contact with the walls of the crossover passage. The spark timing limitation also results in overly late combustion phasing, which allows the compressed gas to re-expand slightly before combustion can begin. The lower compression ratio is detrimental to efficiency, and the late combustion phasing results in average cylinder pressures around 3.5 bar at the time of EVO, meaning additional expansion work is possible.
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Figure 6.46: Breakdown of fuel energy pathways as a function of equivalence ratio.
6.8 Summary

The experimental results of a split-cycle engine designed by the author have been presented in this chapter. Data was acquired for engine speeds of 850 RPM, 1000 RPM, and 1200 RPM, over a range of spark timings and stoichiometric-to-lean air/fuel ratios, at wide open throttle conditions. The maximum load achieved was approximately 6 bar IMEP (4 bar BMEP), and handicapped by the peak cylinder pressure forcing the crossover outlet valve to open at high-load/fast burn conditions. As such, peak cylinder pressures were limited to 35–40 bar. The limit is not exact because it depends on the crossover passage pressure, which had a cycle-mean value in the range of 17–19 bar, with the higher values corresponding to faster engine speeds. The crossover pressure was found to be nearly isobaric throughout the cycle, varying by less than 10% of its average.

Injecting fuel directly into the crossover passage was successful in producing a uniform air/fuel mixture for combustion. However, the volumetric efficiency improvements that were expected from doing so were not realized. For WOT conditions, the split-cycle engine achieved a maximum volumetric efficiency of 75%, which is comparable to, or slightly worse than, a port injected NG engine. High pressure residual air, trapped in the compression cylinder at the end of its compression stroke, effectively delayed the start of the intake stroke by approximately 30°CA, reducing the cylinder filling capability. This is an unavoidable attribute for this split-cycle engine, and indicates that forced induction could be beneficial.

The split-cycle combustion rates were shown to be relatively fast in comparison to conventional SI NG engines; up to four times faster than naturally aspirated engines operating at similar conditions (see Table 6.5). For stoichiometric operation of the split-cycle engine, the average early flame development period (CA\textsubscript{0−10}) and main burn duration (CA\textsubscript{10−90}) were both on the order of 10–13°CA for all engine speeds tested. The fastest burning cycles were typically 2–3°CA quicker than the average, and actually encompassed the majority of cycles since the distribution of data was heavily skewed (see Figure 6.33). Increasingly lean conditions, down to \(\phi = 0.9\), had a negligible effect on the combustion rate, but were shown to rapidly increase thereafter. At the lean limit of operation, \(\phi = 0.83\), all the combustion durations had consistently increased by approximately 4–6°CA. In general, the fastest burn rates were shown to occur at the earliest spark timings and at, or near, stoichiometric conditions, which corresponds to the highest average in-cylinder temperature. Based on the trends found in the current research, further increases in the combustion rate are projected if the ignition timing can be phased closer to TDC.
Phasing of the combustion process is inherently late due to the crossover outlet valve closure timing, which, it was determined, could be made earlier based on the short cylinder filling time (∼10°CA) in comparison to the valve opening duration. The phasing is also highly dependent on the early combustion development period (CA₀₋₁₀), which is quite sensitive to spark timing. A one degree shift in ignition can affect the LPP by 3–3.5°CA. It was also discovered that any delay in the early flame development period ultimately leads to a longer main burn duration, caused by the reduction in charge temperature with expanding cylinder volume. This is one drawback to initiating combustion after TDC.

The rapid combustion process was found to be very stable and robust for all engine speeds operating at air/fuel ratios between φ = 0.9–1.0, and favouring the most advanced spark timings investigated. The rapid increase in burn duration for mixtures leaner than φ = 0.9 was accompanied by a non-linear increase of instability metrics (COVIMEP and COVLPP), indicating highly variable conditions within combustion chamber between cycles. Mixtures leaner than φ = 0.83 resulted in a sufficient number of misfiring cycles (>3%), which prevented accurate air/fuel ratio measurements from being obtained by the O₂ sensor. This relatively high lean operating limit was speculated to be caused by low pre-combustion gas temperatures combined with high strain rates in the flame.

Exhaust gas emission measurements displayed trends common to the majority of SI engines, and, with the exception of a slightly low CO₂ reading, did not show any abnormalities. The CO emissions were found to be very low, roughly 70% less than a typical gasoline-fuelled SI engine operating at stoichiometric conditions. The NOₓ emissions were also low by comparison to conventional SI engines (both gasoline and NG), reaching a maximum of only 800 ppm at φ = 0.97 and a minimum of approximately 150 ppm at φ = 0.83. The THC emission levels ranged from 1500–3000 ppm and consisted of approximately 90% methane. Overall, the split-cycle engine developed in the current research can be considered “clean burning” under the conditions tested.

An energy balance of the engine showed that for the amount of energy supplied by the fuel, approximately 50% is lost to heat transfer, 23–25% is removed by the sensible exhaust enthalpy, 3–4% is by-passed through incomplete combustion, and the remainder provides useful piston work. The high combustion efficiencies were present for all equivalence ratios, confirming mixture uniformity and complete combustion. The high magnitude of losses (internal and exhaust) can be attributed to the highly turbulent flow within the engine, long exposure of compressed gas to cold walls, and late combustion phasing.
Chapter 7

1-D Numerical Engine Simulation

7.1 General

To evaluate the effects of certain parameters on engine performance, a one-dimensional (1-D) numerical fluid model of the split-cycle engine was created using AVL BOOST software. In addition to the internal engine flow, the model includes all the external engine ducting, from the intake air filter to the exhaust surge tank, as shown by the block diagram in Figure 7.1. The flow geometry was discretized into 5 mm lengths and NTP conditions were used for the intake and exhaust boundary conditions. Combustion, heat transfer, and flow losses were predicted using the various procedures outlined in Section 7.2 of this chapter.

Three investigations were performed using this modelling tool:

1. Initial valve timing study for lobe profile development (not discussed here in detail)
2. Effects of crossover passage volume on engine performance
3. Effects and optimization of valve timing using an empirically tuned model

The optimization software used for the valve timing studies (items 1 and 3) is covered in Section 7.3. The results of the valve timing study for lobe profile development are given in Section 3.6.3 of Chapter 3 and will not be presented again in this chapter. However, the same methodology was used for the valve timing optimization study, which is presented in Section 7.4.3. Accuracy of the model is discussed in Section 7.4.1, where a comparison between the empirical data and the numerical results is provided. The effect of crossover volume on engine performance is also given in Section 7.4.2.
7.2 Model Equations

This section outlines several important equations and applicable constants used in the state property calculations of the AVL BOOST model. The fundamental equations used by the software (e.g. conservation equations, solver methods) will not be presented here and can be found in Chapter 2 of the BOOST manual [12]. The information provided in this section is intended to give the reader insight into the specific sub-models used to calculate heat transfer, fluid mass flow rates, and the heat released due to combustion.

7.2.1 Heat Transfer

In-Cylinder Heat Transfer

In-cylinder heat transfer between the gas and walls (i.e. cylinder head, piston, and liner) was calculated using the generic convective heat transfer formula:

\[ Q_{w,i} = A_i h_w (T_c - T_{w,i}) \]  

(7.1)
Each wall element is defined by its own surface area, $A_i$, and temperature, $T_{w,i}$. The in-cylinder gas temperature is $T_{c,1}$, and assumed to be uniform throughout its volume. The heat transfer coefficient, $h_w$, in each cylinder was estimated by the Woschni model [70], Equation (7.2), with constants $C_1$ and $C_2$ pre-defined in BOOST.

$$h_w = 130 \cdot B^{-0.2} \cdot p_c^{0.8} \cdot T_e^{-0.53} \left[ C_1 \bar{S}_p + C_2 \frac{V_d T_{c,1}}{p_{c,1} V_{c,1}} \cdot (p_e - p_{c,o}) \right]^{0.8} \tag{7.2}$$

where:
- $C_1 = 2.28 + 0.308 \cdot c_u \cdot \bar{S}_p^{-1}$
- $C_2 = 0.00622$
- $B$ is the cylinder bore
- $\bar{S}_p$ is the mean piston speed
- $c_u$ is the local average gas velocity
- $V_d$ is the cylinder displacement volume
- $p_{c,o}$ is the motored cylinder pressure
- $T_{c,1}$ is the in-cylinder temperature at IVC
- $p_{c,1}$ is the in-cylinder pressure at IVC

Wall temperatures were all assumed to be constant with the exception of the cylinder liner temperature, $T_L$, which was calculated from Equations (7.3) and (7.4) to account for variation along the cylinder axis. The variable $x$ represents the actual piston position in relation to the full stroke.

$$T_L = T_{L,TDC} \cdot \frac{1 - e^{-cx}}{x \cdot c} \tag{7.3}$$

$$c = \ln \left( \frac{T_{L,TDC}}{T_{L,BDC}} \right) \tag{7.4}$$

Table 7.1 lists the wall temperatures used in the model. The expansion cylinder wall temperatures were estimated from values given by Heywood [70]. The compression cylinder wall temperature, which does not experience heat from combustion, was estimated based upon the measured gas temperature in the crossover passage and the engine coolant temperature.

**Pipe Flow Heat Transfer**

Pipe flow heat transfer was used for the intake runner, the crossover passage, and up to the EGT measurement location in the exhaust pipe (see Figure 7.1). Convective gas-to-wall heat transfer in the piping was calculated based on a Nusselt number approach using the Colburn model, given in Equation (7.5).

$$Nu = 0.0243 \cdot Pr^{0.4} \cdot Re^{0.8} \tag{7.5}$$
Table 7.1: Surface temperatures (°C) used for AVL BOOST model.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Compression Cylinder</th>
<th>Expansion Cylinder</th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston Surface</td>
<td>130</td>
<td>250</td>
</tr>
<tr>
<td>Cylinder Head</td>
<td>130</td>
<td>180</td>
</tr>
<tr>
<td>Cylinder Wall @ TDC</td>
<td>130</td>
<td>180</td>
</tr>
<tr>
<td>Cylinder Wall @ BDC</td>
<td>100</td>
<td>150</td>
</tr>
<tr>
<td>Intake/Exhaust Port</td>
<td>70</td>
<td>300</td>
</tr>
<tr>
<td>Crossover Passage</td>
<td>90</td>
<td>90</td>
</tr>
</tbody>
</table>

The convective heat transfer coefficient, \( h \), can then be solved from the defining equation of the Nusselt number: \( Nu = hL/k \), where \( L \) is a characteristic length and \( k \) is the thermal conductivity of the fluid. The standard convective heat transfer formula, of identical form to Equation (7.1), can be solved if the wall temperatures are known. The external wall temperatures of the intake and exhaust pipes were measured using a hand-held infrared thermometer and input into the BOOST model—the temperature difference across the pipe wall was assumed to be negligible. Values of 60°C and 400°C were used for the intake and exhaust pipes attached directly to the engine, respectively. The crossover wall temperature was assumed to be slightly above the coolant temperature at 90°C.

Port Heat Transfer

Heat transfer inside the intake and exhaust ports was estimated using the modified Zapf heat transfer model, given in Equation (7.6) [12].

\[
T_d = (T_u - T_w) \cdot e^{-A_w \frac{h_p}{m c_p}} + T_w
\]

(7.6)

where:  
- \( T_d \) is the temperature downstream from port  
- \( T_u \) is the temperature upstream from port  
- \( T_w \) is the wall temperature  
- \( A_w \) is the port wall area  
- \( h_p \) is the heat transfer coefficient  
- \( m \) is the fluid mass flow rate  
- \( c_p \) is the specific heat at constant pressure
For flow into the cylinder, the heat transfer coefficient is determined by:

\[ h_p = (C_1 + C_2 \cdot T_u - C_3 \cdot T_u^2) \cdot T_u^{0.44} \cdot \dot{m}^{0.5} \cdot d_{vi}^{-1.5} \cdot \left[ 1 - 0.797 \cdot \frac{h_v}{d_{vi}} \right] \] (7.7)

And for flow out of the cylinder, the heat transfer coefficient is determined by:

\[ h_p = \left( C_1 + C_2 \cdot T_u - C_3 \cdot T_u^2 \right) \cdot T_u^{0.33} \cdot \dot{m}^{0.68} \cdot d_{vi}^{-1.68} \cdot \left[ 1 - 0.765 \cdot \frac{h_v}{d_{vi}} \right] \] (7.8)

where \( h_v \) is the valve lift height, and \( d_{vi} \) is the inner valve seat diameter. The constants \( C_1, C_2, \) and \( C_3 \) vary depending on the flow direction, and are listed in Table 7.2. The modified Zapf model is based on empirical data from conventional poppet valves operating at typical pressure ratios. The increased heat transfer caused by the high fluid velocity through the reverse poppet valves is accounted for in the mass flow term (\( \dot{m} \)); however, it is questionable whether or not the coefficients are correct. More complex temperature measurements within the engine cylinder and ports would be required to further validate the accuracy of the port heat transfer model under split-cycle operating conditions.

<table>
<thead>
<tr>
<th>Table 7.2: Constants for port heat transfer coefficient - Equations (7.7) and (7.8).</th>
</tr>
</thead>
<tbody>
<tr>
<td>Constant</td>
</tr>
<tr>
<td>( C_1 )</td>
</tr>
<tr>
<td>( C_2 )</td>
</tr>
<tr>
<td>( C_3 )</td>
</tr>
</tbody>
</table>

### 7.2.2 Combustion

For simulations performed in advance of having the physical engine running, a simple Vibe (Wiebe) model was used to represent combustion. Once empirical combustion data was available, the Vibe model was replaced by a heat release rate derived from the physical data, which is described further in Section 7.4.1. In all cases, because the cylinder is zero-dimensional, the mixture is assumed to be perfectly homogeneous, and no distinction between burned and unburned gas properties is made.

#### Vibe (Wiebe) Model

The Vibe model, also commonly spelled Wiebe \([70]\), is a method of specifying the mass fraction burned, \( x_b \), as a function of crank angle, \( \theta \), directly in terms of the combustion
duration, $\Delta \theta$, a combustion completeness parameter, $a$, and a shape parameter, $m$, as shown in Equation (7.9). For simulations using the Vibe model, $a = 6.9$ was used for complete combustion and $m$ was set to 0.5 in order to best mimic the shape of a heat release curve published by Phillips et al. [115] of a split-cycle engine operating on gasoline. The start of combustion, $\theta_o$, was left as a variable in the valve timing study. The complete combustion duration was set to 35°CA, also estimated from the work of Phillips [115].

$$x_b = 1 - \exp \left[ -a \left( \frac{\theta - \theta_o}{\Delta \theta} \right)^{m+1} \right]$$  \hspace{1cm} (7.9)

### 7.2.3 Gas Exchange

The mass flow rate through the engine ports was calculated using the equations for isentropic flow through an orifice: Equation (7.10) for subsonic flow, and Equation (7.11) for sonic flow. The flow rates are determined by the pressure ratio across each valve.

$$\dot{m}_s = A_e \cdot p_{o,1} \cdot \left[ \frac{2\gamma}{RT_{o,1}(\gamma - 1)} \left( \left( \frac{p_2}{p_{o,1}} \right)^{\frac{2}{\gamma}} - \left( \frac{p_2}{p_{o,1}} \right)^{\frac{\gamma + 1}{\gamma}} \right) \right]^{\frac{1}{2}}$$ \hspace{1cm} (7.10)

$$\dot{m}_s = A_e \cdot p_{o,1} \cdot \sqrt{\frac{2\gamma}{RT_{o,1}(\gamma + 1)}} \left( \frac{2}{\gamma + 1} \right)^{\frac{1}{\gamma - 1}}$$  \hspace{1cm} (7.11)

The variables of Equations (7.10) and (7.11) are defined as follows:

- $A_e$ is the effective flow area (referenced to inner seat diameter in Boost [12])
- $p_{o,1}$ is the upstream stagnation pressure
- $T_{o,1}$ is the upstream stagnation temperature
- $R$ is the gas constant for air
- $p_2$ is the downstream static pressure
- $\gamma$ is the ratio of specific heats

To account for irreversibility losses, a discharge coefficient was applied to each port/valve combination. As shown in Equation (7.12), the discharge coefficient, $C_d$, accounts for all inefficiencies between the actual mass flow rate, $\dot{m}_a$, and the isentropic flow rate through a converging nozzle of comparable reference diameter, $\dot{m}_s$. The actual mass flow rates were either measured or estimated using computational fluid dynamic (CFD) simulations.

$$C_d = \frac{\dot{m}_a}{\dot{m}_s}$$ \hspace{1cm} (7.12)
For the intake and exhaust valves, the mass flow rate through each port was physically measured using a Saenz S600 flow bench at a pressure differential of 69.7 mbar and with the flow moving in the same direction as the actual engine. Due to the high pressure ratio of the crossover valves, their flow rates were estimated by CFD simulations in both flow directions at a downstream-to-upstream pressure ratio of 1:2. The author would like to acknowledge and thank Mr. Zakaria Movahedi for performing these simulations.

Figure 7.2 shows the resulting discharge coefficients for all four valves as a function of lift, with the corresponding pressure ratios given in the legend. As predicted by Figure 3.39 in Section 3.6.3, minimal gains in flow are seen above 6–7 mm of lift for the intake and exhaust valves due to the transition of minimum flow area (i.e. throat area) from the valve aperture opening to the port cross-sectional area.

![Figure 7.2: Valve port discharge coefficients as a function of lift.](image)

With the discharge coefficients known, BOOST uses Equation (7.12) to solve for the actual mass flow rate. It should be noted that only a single pressure ratio was used to define the discharge coefficients, meaning they are taken as a constant for all pressure ratios in the simulation. This is one aspect of the model that could use improvement.

### 7.3 Third-Party Optimization Software

Optimization of the engine valve timing was performed by integrating modeFRONTIER®, a multi-objective optimization software produced by Esteco, with the 1-D numerical model in BOOST. ModeFRONTIER® was used to programmatically adjust the valve timing
within the engine model based on user-defined parameter targets and an optimization routine. For each valve timing configuration, the output parameters were passed back to modeFRONTIER® as inputs to the optimization algorithm. For example, the maximization of the average air mass flow rate through the engine and net IMEP were used as feedback quantifiers in the design and optimization of the intake and exhaust cam lobe durations. Ranges and limits imposed on the optimizer were used to prevent unrealistic timing values from being selected, such as intake and exhaust opening values that would cause piston-to-valve interference in the actual engine. The circular exchange of information between the two software packages is depicted in Figure 7.3.

![Feedback flow diagram between modeFRONTIER® and AVL BOOST.](image)

**Figure 7.3:** Feedback flow diagram between modeFRONTIER® and AVL BOOST.

Prior to implementing the optimization routine, the design space was first sampled using a Uniform Latin Hypercube (ULH) Design of Experiments (DOE). The DOE consisted of 20 unique cases and provided a starting point for the optimizer. A HYBRID search algorithm was selected for the optimization, which is the combination of a steady-state genetic algorithm and a sequential quadratic programming optimizer [52]. The HYBRID method does not have a convergence control and the number of trials was initially set to 1000, but it was later discovered that 750 trials were adequate to generate a well-defined Pareto front. Repetition of cases was prevented in the software, and therefore each of the 750 trials had a unique combination of valve and spark timings.

---

1 Pareto front refers to the conditions of maximum efficiency for conflicting objectives. For example, if A and B are conflicting objectives, the line along which no A can be greater without sacrificing B, and vice-versa, is the Pareto front.
7.4 Modelling Results

7.4.1 Comparison with Experimental Data

To check the accuracy of the AVL BOOST engine model, the measured pressure traces from both cylinder and the crossover passage were compared with the numerically simulated traces. This was initially performed using the motoring curves in order to extricate the process from errors associated with the combustion model. First comparisons revealed higher post-compression pressures in the model by 2–3 bar, which originated from the near isentropic compression process of the model ($n_c = 1.4$), compared to the real engine that has a value of approximately $n_c = 1.33$. This discrepancy is believed to be related to the blow-by issue discussed in Section 6.4.3. To account for the difference, the “effective blow-by gap” for the compression cylinder in the model was increased until the pressure magnitudes were equal; an increase of 5.2$\mu$m was required. The BOOST manual does not explicitly state how this length is used to calculate the blow-by flow area.

Small adjustments were also made to the valve timing in the numerical model in order to properly phase the pressure traces with those from the actual engine. These initial discrepancies between the simulated (nominal) valve timing and the empirically observed values can be attributed to compliances in the physical valvetrain and timing drive (e.g. belt stretch). The aligned pressure curves are shown in Figure 7.4 for an engine speed of 850 RPM. The largest difference between the two curves can be seen in the compression cylinder at the start of the intake stroke, during the de-pressurization period ($30–50^\circ$CA). Since the flow is choked and going backwards through the intake valve at this time, it is reasonable to expect the discrepancy comes from an inaccurate discharge coefficient, which was measured under non-choked and forward flowing conditions.

The only other discrepancy between the simulated and measured data is the amplitude and frequency of oscillations present while the crossover valves are open; the BOOST model depicts larger amplitude and lower frequency wave data. It was discovered that decreasing the discretization length within the model improved these characteristics, but with a significant increase in computational time. The discretization length was therefore kept at 5 mm. With the exception of these two aspects, the AVL BOOST model closely represents the pressures measured inside the engine under motoring conditions.

With the valve timing corrected, combustion was reinstated into the simulation. The Vibe model originally used for combustion significantly over-predicted the rate of heat release, yielding a peak combustion pressure in the range of 5–7 bar above the physically tested
averages. Limited success in aligning the combustion traces was achieved by manipulation of the combustion model constants \((a\) and \(m\)), and positive results were only found when parameter \(a\) was reduced below 5, which effectively represents an incomplete burn. From the experimental emissions data, this is known to not be the real case, and therefore an alternative combustion model was pursued.

AVL BOOST has a built-in utility called BURN that allows the user to import an experimental combustion pressure trace into the software, for which it will then calculate the rate of heat release. The utility is somewhat of a ‘black box’, and does not indicate what method(s) is/are being used in the heat release calculation; however, based on the information required by the BURN utility (temperatures, mass flow rates, etc.) it can be assumed that a First Law analysis is being performed.

Applying the heat release profile calculated by the BURN utility to the split-cycle engine model produced results very similar to the actual engine, as shown in Figure 7.5. The slight variation in peak pressure is of little consequence, since cyclic variability in the engine causes fluctuations larger than this—keeping in mind the COV of peak pressure is on the order of 7–10 \%. Immediately following the SOC, during the flame development period, the AVL model over-predicts the heat release rate, and does not fully capture the drop in pressure before bulk combustion occurs. This is likely a misinterpretation by the BURN utility’s heat release calculation, stemming from the fact that the crossover outlet valve is open during the early stages of combustion—recall the MFB error presented in Section 5.2.4.

**Figure 7.4:** Comparison of measured and simulated pressure traces over a complete engine cycle under motoring conditions at 850 RPM, WOT.
A comparison of other engine parameters, for which measured data was available, is given in Table 7.3. The largest discrepancy between the model and the physical measurements is the crossover passage temperature, with the model being $90^\circ$C hotter. There is probable evidence that the error lies within the empirical value. Experiments were performed with the crossover passage thermocouple at various depths across the passage, and temperature measurements were found to reduce by $60^\circ$C in the vicinity of the passage wall. Given that the thermocouple is located approximately $4\text{ mm}$ above the crossover passage floor, it is likely that localised wall-cooling effects are causing the measured value to be lower than the actual bulk gas temperature.

The remainder of the parameters listed in Table 7.3 are within $10\%$ of the experimental values and this was considered adequate for the purposes of this study. It should be noted that the simulated exhaust temperature represents the cyclic average. However, the model predicts peak values up to $930^\circ$C at the outlet of the exhaust port. This corroborates with the EGT values calculated for the energy balance in Section 6.7.
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Table 7.3: Comparison of measured and simulated parameters for engine conditions of 850 RPM, WOT, $\phi = 1$, spark timing 20°CA ATDC-e.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Engine</th>
<th>AVL</th>
<th>Difference</th>
<th>% Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>IMEP-c (bar)</td>
<td>-3.92</td>
<td>-4.31</td>
<td>0.39</td>
<td>9.5</td>
</tr>
<tr>
<td>IMEP-e (bar)</td>
<td>10.12</td>
<td>10.21</td>
<td>0.09</td>
<td>0.9</td>
</tr>
<tr>
<td>IMEP net (bar)</td>
<td>6.20</td>
<td>5.85</td>
<td>0.35</td>
<td>5.8</td>
</tr>
<tr>
<td>Combustion $p_{\text{max}}$ (bar)</td>
<td>30.7</td>
<td>29.4</td>
<td>1.3</td>
<td>4.2</td>
</tr>
<tr>
<td>Air Flow (g/s)</td>
<td>3.05</td>
<td>3.25</td>
<td>0.20</td>
<td>6.4</td>
</tr>
<tr>
<td>Crossover Pressure (bar)</td>
<td>17.2</td>
<td>18.3</td>
<td>1.1</td>
<td>6.2</td>
</tr>
<tr>
<td>Crossover Temp. (°C)</td>
<td>170</td>
<td>260</td>
<td>90</td>
<td>41.9</td>
</tr>
<tr>
<td>Exhaust Temp. (°C)</td>
<td>524</td>
<td>527</td>
<td>3</td>
<td>0.6</td>
</tr>
</tbody>
</table>

* IMEP-c, IMEP-e are for compression and expansion cylinders, respectively.

7.4.2 Crossover Volume Effects

General

During the period when both crossover valves are open and mass is being transferred into and out of the crossover passage, compression continues until effective TDC is reached at 10°CA ATDC-e. At this time, the total gas volume being compressed includes that of the crossover passage. Therefore, the extent of actual gas compression is a function of the crossover volume.

To investigate the effects of the crossover passage volume on the performance of the engine, a case study was used to vary the crossover volume as a fraction of its current physical value; from 0.1 to 2, in increments of 0.1. The current passage is modelled as a simple round duct with a diameter of 26 mm and a length of 230 mm, which equates to the actual volume in the physical engine of 122 cm$^3$. The diameter in the model was selected to match the minimum cross-sectional area of the actual passage. The volume was changed by varying only the length of the crossover passage in order to maintain a similar flow velocity, and therefore heat flux, between cases. Spark timing was fixed at 20°CA ATDC-e, an engine speed of 850 RPM was used, and the same combustion profile was applied to all cases.

Results and Discussion

The effect of crossover volume on the net IMEP is shown in Figure 7.6. It can be seen that decreasing the volume below its current value exponentially increases the indicated
performance. In fact, reducing the volume fraction from 1 to 0.1 resulted in a 23\% increase in net IMEP. For crossover volume fractions greater than unity the trend appears to level off, with net IMEP values oscillating within ±4\% of the original value. This indicates that increasing the current crossover passage volume of the engine would have little effect on IMEP; however, a decrease would lead to higher indicated performance.

Since the combustion heat release rate is the same for every case, the growth in net IMEP must come from an increase in pre-combustion cylinder pressure, which is determined by the crossover pressure. Figure 7.7 shows the effect of crossover volume on the average crossover passage pressure. The general trend indicates that a smaller passage volume leads to a higher overall pressure, which is logical since the latter part of the compression process includes the gas inside the crossover passage; a smaller volume will compress to a higher final pressure given the same initial conditions and displaced volume.

The increase in net IMEP could also be achieved by reducing the compression cylinder work (IMEP-c). For the smallest crossover volume, the simulation showed a 2.2\% reduction in volumetric efficiency, which could be expected to reduce IMEP-c. However, despite the lower mass being ingested in this case, IMEP-c was found to be at its maximum on account of the high post-compression pressure. The volumetric efficiency was reduced by the corresponding density increase of the residual trapped air mass in the compression cylinder. Since this mass must re-expand before intake flow can begin, less fresh charge was able to be inducted during the intake stroke.
Figure 7.7: Effect of crossover volume on crossover pressure.

Also shown in Figure 7.7 are the standard deviations in crossover pressure over the simulated cycle at each operating point. It was found that a larger crossover volume had an improved ability to dampen pressure waves caused during the gas exchange process, and therefore had less variation in pressure over the cycle. This is not in reference to the harmonic pressure oscillations, which tend to actually have a larger amplitude with increasing crossover volume, but the variations caused by gas exchange between the crossover passage and cylinders. The two were easily distinguished by the higher oscillation frequency of the harmonics.

Curiously, the increase in crossover pressure (Figure 7.7) does not follow the same exponential trend as the net IMEP (Figure 7.6). However, examination of the expansion cylinder pressure at the SOC did show an exponential trend similar to that of the net IMEP. In fact, the same is true for the crossover pressure, but the trend is simply down-washed by the cyclic averaging process and the large pressure variations for smaller crossover passages.

At this point the combined geometric compression ratio \((CR_{gc})\) can be introduced:

\[
CR_{gc} = \frac{V_{cl} + V_x}{V_e}
\]  \hspace{1cm} (7.13)

where \(V_{cl}\) is the compression cylinder volume at BDC, \(V_x\) is the crossover passage volume, and \(V_e\) is the combined volumes of both cylinders and the crossover passage at effective TDC (i.e. 10°CA ATDC-e). Therefore, \(CR_{gc}\) is the ratio of the initial gas volume in both the compression cylinder and crossover passage, to the overall minimum volume when all three chambers are openly connected. This is not a typical compression ratio, in the sense that the initial conditions of the crossover passage and the cylinder are not equal.
Figure 7.8: Relationship between crossover passage volume and combined geometric compression ratio.

Figure 7.9: Effect of combined geometric compression ratio on net IMEP and average crossover passage pressure.
Figure 7.8 shows how $CR_{gc}$ increases non-linearly as the crossover passage volume is reduced, in a nearly identical fashion to the net IMEP in Figure 7.6. Therefore, by plotting the net IMEP as a function of the combined geometric compression ratio, as shown in Figure 7.9, a linear correlation between the two parameters emerges. The clustering of data points near the y-axis reveals the small effect a volume change has on the compression ratio when the crossover volume is large. The non-linear relationship between the crossover volume and the $CR_{gc}$ results in a large increase in the latter when the crossover volume becomes very small. Thus, it can be said that the exponential increase in IMEP with decreasing crossover passage volume is a direct result of the net increase in actual gas compression.

Figure 7.9 also shows the average crossover pressure as a function of the $CR_{gc}$. It can be seen that minimal gains in mean crossover pressure are realized beyond a value of $CR_{gc} \approx 4.5$. This may seem trivial since, despite the levelling off in crossover pressure, net IMEP continues to rise. However, the shortest crossover passage may not be desirable from an air/fuel mixing stand-point, which will be discussed in the next section. Limitations on the maximum gas temperature, in terms of autoignition, will also dictate the upper limit for $CR_{gc}$. The simulated crossover passage temperature follows the same trend as the pressure, and at $CR_{gc} = 4.5$ has a mean value of 375°C, which is within the vicinity of autoignition for methane-air mixtures at the corresponding pressure [106]. Based on Figure 7.8, this means the current crossover passage volume could be decreased by approximately 50% before experiencing autoignition, although this estimate should be considered imprecise.

**Conclusions and Deficiencies**

Based on a 1-D numerical analysis, the following conclusions can be drawn in regard to the effects of the crossover volume:

- Reducing the crossover passage volume increases its average pressure (for a fixed valve timing), resulting in a higher pre-combustion expansion cylinder pressure and consequently an increase in net engine IMEP.

- The increase in IMEP is linearly correlated with the combined geometric compression ratio. The non-linear relationship of the combined geometric compression ratio with the crossover volume causes an exponential increase in IMEP when the crossover volume is decreased.

- A correlation between crossover volume and the amplitude of the pressure fluctuations within the crossover passage exists. Increasing the crossover volume helps to absorb
pressure changes caused by incoming and outgoing flow, effectively reducing the peak-to-peak amplitude over the course of a cycle.

Minimization of the crossover passage volume is advisable for the purposes of reducing heat transfer losses and increasing the actual gas compression ratio. However, the numerical model does not incorporate the fuel/air mixing process that takes place in the crossover passage of the actual engine. By reducing the crossover passage volume, the length of time for air/fuel mixing within the crossover passage is reduced, and charge stratification could result. A more complex, 3-D fluid model and/or physical testing would be required to determine if adequate mixing could still be achieved for a smaller crossover passage.

The 1-D model is also deficient in predicting the changes in combustion effected by varying conditions in the combustion chamber. For example, a greater crossover passage pressure leads to a higher charge temperature, which has an enhancing effect on the reaction rate. Furthermore, in the case of the test engine, it also allows higher peak cylinder pressures to be achieved without pushing open the crossover outlet valve. This means more advanced timing can be implemented, which should increase both the IMEP and the thermal efficiency. Coupling the 1-D model with a more comprehensive engine cylinder model that includes species transport/mixing and combustion would be required to numerically investigate crossover volume effects with greater accuracy and more insight.

In addition to the autoignition temperature, the decrease in crossover passage volume is also practically limited by the packaging requirements of the valves and fuel injector. However, it is worthwhile to note that an opposed piston, split-cycle engine operating without the use of a crossover passage does exist [136]. To the best of the author’s knowledge, no peer-reviewed literature has been published on that specific application. Fuel/air mixing, adequate thermal management, valve packaging, and sensitivity to piston and/or valve timing are suspected to be the primary challenges with this type of configuration.

### 7.4.3 Valve Timing Effects

The crossover valve timing plays an important role in determining the operating characteristics of the split-cycle engine. During the compression stroke, the crossover inlet valve opening (XIVO) point determines to what extent the gas is pressurized before being transferred into the crossover passage. Similarly, the crossover outlet valve closing (XOVC) event controls how much gas expansion is possible before the exhaust valve opens. By delaying the XIVO time or advancing the XOVC time, the degree of compression or expansion in
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Each respective cylinder approaches the standard geometric compression or expansion ratio, both of which are irrespective of any fluid transfer. The finite period required for gas exchange to and from the crossover passage prevents these ratios from being practicable.

The question that still remains is: what combination of crossover valve timing will produce the largest net IMEP for a given operating condition? Furthermore, for a fixed crossover volume, does changing the valve timing to achieve a higher crossover passage pressure necessarily lead to a higher net IMEP? A higher crossover passage pressure is desirable to increase the allowable margin of peak combustion pressure whilst maintaining crossover valve closure (see Section 6.4.2), but what is not known at this point is how the crossover valve timing affects the crossover passage pressure. This section is intended to answer these questions and explore the effect each crossover valve has on the performance of the split-cycle engine. The same 1-D numerical model discussed previously in this chapter was used to perform the study, alongside the optimization software that was presented in Section 7.3. The BURN combustion model was also reused from the previous study (see Section 7.4.1).

Optimization Work Flow

The following are the input variables that were optimized in this study, along with their range of allowed variation in square brackets:

- Crossover inlet valve opening (XIVO), [27–42°CA BTDC-c]
- Crossover outlet valve opening (XOVO), [5–15°CA BTDC-e]
- Crossover outlet valve closing (XOVC), [10–30°CA ATDC-e]
- Start of combustion (SOC), [10–20°CA ATDC-e]

The crossover inlet valve closing (XIVC) time was not included in the optimization, since it was already known that XIVC should occur at TDC-c in order to maximize the mass transferred out of the compression cylinder before TDC-c, and to prevent subsequent backflow into the cylinder after TDC-c. The SOC was allowed to vary since it has a strong impact on engine performance. It is also dependent on the XOVC time, since the bulk of combustion should not occur while the crossover outlet valve is still open. However, no limitation was imposed on the SOC relative to XOVC in the optimization scheme.

The output parameters from the BOOST model used for optimization were the maximization of both net IMEP and crossover passage pressure. Net IMEP was used in order to maximize work output. It also correlates well with the actual gas compression ratio, and therefore should increase simultaneously with overall efficiency. Crossover pressure was
selected because a higher mean value would allow greater peak combustion pressures to be utilized in practice. To ensure the resulting valve timings could be implemented with current crossover valve spring design, the optimizer was constrained to a maximum peak cylinder-to-crossover pressure differential of 18 bar.

The modeFRONTIER® work flow diagram used to coordinate the optimization is shown in Figure 7.10. Details of the optimization algorithm are given in Section 7.3.

![Figure 7.10: Screenshot of modeFRONTIER® work flow model.](image)

**Optimization Results**

All results presented in this section are for an engine speed of 1000 RPM, at WOT, and for a stoichiometric air/fuel ratio. Figure 7.11 shows the resulting trade-off (Pareto front) between the net IMEP and the mean crossover passage pressure, $\bar{p}_x$, for 750 unique combinations of valve and spark timing selected by the optimizer. It is evident from the figure that the highest crossover pressures do not result in the largest net IMEP. In fact, along the Pareto front, increasing $\bar{p}_x$ from 27 bar to 44 bar reduces the net IMEP by nearly 25%.

Three distinct cases have been selected for a detailed presentation, marked A, B, and C on Figure 7.11, and are categorized as follows:

- **Case A**: highest net IMEP for all combinations investigated.
- **Case B**: mid-point, representing a good trade-off between both variables.
- **Case C**: highest crossover pressure before sharp decline in net IMEP.
Chapter 7: 1-D Numerical Engine Simulation

**Figure 7.11:** Trade-off between net IMEP and crossover pressure for 750 different combinations of crossover valve and spark timing.

The specifics of each case are given in Table 7.4, where $\bar{p}_x$ is the mean crossover passage pressure, and $\Delta p = p_{max} - \bar{p}_x$ (i.e. the difference between the peak combustion pressure and the mean crossover passage pressure). All valve timings are listed in terms of degrees ATDC-e and the values have been rounded to the nearest half-degree.

**Table 7.4:** Specifications of selected cases (A, B, & C) from Figure 7.11.

<table>
<thead>
<tr>
<th>Case</th>
<th>A</th>
<th>B</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>SOC (°CA)</td>
<td>14.1</td>
<td>10.6</td>
<td>13.8</td>
</tr>
<tr>
<td>XIVO (°CA)</td>
<td>348</td>
<td>349</td>
<td>348.5</td>
</tr>
<tr>
<td>XIVC (°CA)</td>
<td>22</td>
<td>22</td>
<td>22</td>
</tr>
<tr>
<td>XOVO (°CA)</td>
<td>353.5</td>
<td>349.5</td>
<td>349</td>
</tr>
<tr>
<td>XOVC (°CA)</td>
<td>25</td>
<td>25</td>
<td>10</td>
</tr>
<tr>
<td>IMEP-c (bar)</td>
<td>-5.20</td>
<td>-5.69</td>
<td>-6.08</td>
</tr>
<tr>
<td>IMEP-e (bar)</td>
<td>12.90</td>
<td>12.48</td>
<td>11.75</td>
</tr>
<tr>
<td>IMEP net (bar)</td>
<td>7.70</td>
<td>6.79</td>
<td>5.67</td>
</tr>
<tr>
<td>$\bar{p}_x$ (bar)</td>
<td>26.9</td>
<td>36.2</td>
<td>43.7</td>
</tr>
<tr>
<td>$\Delta p$ (bar)</td>
<td>16.3</td>
<td>9.4</td>
<td>-2.2</td>
</tr>
</tbody>
</table>

*All crank angle values are in reference to ATDC-e.

The compression, expansion, and crossover pressure curves for cases A, B, and C are shown in the sub-plots of Figure 7.12.
Chapter 7: 1-D Numerical Engine Simulation

Figure 7.12: Engine pressure curves for select cases A, B, and C.
A matrix showing the Pearson correlation coefficient between all input and output variables used in the model is given in Figure 7.13. The number inside each box represents the correlation coefficient, and each box is also colour contrast weighted according to its value. Red and blue colours indicate positive and negative correlations, respectively.

![Matrix showing Pearson correlation coefficients for changes in crossover valve timing.](image)

**Figure 7.13:** Pearson correlation coefficients for changes in crossover valve timing.

**Discussion**

The purpose of this study was to determine what effect the valve timing has on the crossover passage pressure and net IMEP of the engine. Figure 7.11 reveals that having a highly pressurized crossover passage and a large net IMEP are mutually antagonistic conditions for this specific engine configuration. Although this may seem contradictory to the results from Section 7.4.2, the reader should be cognisant of the fact that the crossover volume was fixed in this case, and thus the increase in $\bar{p}_x$ came from the valve timing, which evidently resulted in the sacrifice of net IMEP. The three select cases shown in Figure 7.12 provide insight into this trade-off between net IMEP and $\bar{p}_x$.

From case C, it is apparent that a large reduction in cylinder pressure prior to the bulk combustion period leads to a peak cylinder pressure below that of the crossover passage. Interestingly, it is this case that has the earliest XOVC timing: 10°CA ATDC-e compared to 25°CA ATDC-e for the other two cases, neither of which exhibit any pressure drop between TDC and the rise due to combustion. There are three reasons for this: 1) the earlier XOVC timing occurs when the piston is closer to TDC where a small change in volume is more pronounced, since it represents a larger fraction of the total volume, 2) the SOC
occurs after XOVC for case C, in contrast to cases A and B, which start burning before the
crossover valve closes, and 3) despite case C having a higher crossover pressure, the smaller
cylinder volume at the time of XOVC results in less mass (air/fuel charge) being captured
for combustion. In fact, there is approximately 11% less mass in the combustion chamber
of case C than case A. This also explains why the crossover passage pressure is substantially
higher for case C; a back-log of mass is being built up in the crossover passage, which is
also detrimental to the compression IMEP as evidenced in Table 7.4. It should be clarified
that this back-log is not indefinite and the results given here represent a steady-state value.

Advancing the spark timing of case C would appear to be a logical method to eliminate the
pre-combustion pressure drop and improve the net IMEP. The result of doing so actually
lies within case B, which has nearly identical inputs to case C with the exception of an
earlier spark timing and a slightly later XOVO. Case B also has a substantially higher
crossover passage pressure, yet exhibits a lower net IMEP. The reason for this is that the
rise in combustion pressure, while the crossover valve is still open, forces fresh charge back
into the crossover passage, again reducing the amount of trapped mass in the combustion
chamber at the time of XOVC. This was observed in the model by examining the mass flow
rate through the crossover outlet valve, which becomes negative for a reverse flow. Thus,
advancing the spark timing in case A does increase IMEP to a small extent. However,
the higher crossover passage pressures for those cases also increases the magnitude of the
compression IMEP and SOC is limited by the XOVC timing.

Case A shows that the highest net IMEP is achieved with a crossover passage pressure
of approximately 27 bar and combustion being initiated at 14°CA ATDC-e, 11°CA before
XOVC. This should be expected, recalling from Chapter 6 that the flame development
period was found to be on the order of 10°CA. The back-flow in this case is small, but
physical testing would be required to determine whether earlier XOVC or later SOC would
be required to prevent combustion from propagating back into the crossover passage. The
late XOVC timing for case A is indicative that trapped mass is the driving force for the
trade-off between IMEP and crossover passage pressure.

From an engine design perspective, there is a conflict between minimizing the expansion
cylinder clearance volume for full expulsion of exhaust gas residuals, and providing a com-
bustion chamber volume large enough to trap a sizeable air/fuel mass. Employing a larger
clearance height or combustion chamber volume could be used in conjunction with positive
valve overlap (PVO) to drive out exhaust products with the incoming charge. For the case
of the split-cycle engine, PVO refers to the crank angle duration over which the exhaust
valve and the crossover outlet valve are both open. However, for a natural gas fuelled, split-cycle engine, this is a risky strategy given the potential for short-circuiting during PVO and the potency of methane as a greenhouse gas.

Thus far, the discussion has been primarily focused on the effects of the XOVC time, with no mention of the opening-time effects. From Figure 7.13, it can be seen that the XOVO time is only marginally correlated with both the IMEP and the mean crossover pressure. This is because the exact timing of the cylinder filling does not have significant consequences on either parameter, provided the exhaust valve has already closed. If the XOVO time precedes EVC, short-circuiting to the exhaust lowers the net IMEP and $\bar{p}_x$ due to lost compression work. If the exhaust valve is closed, an early XOVO time (i.e. BTDC-e) simply results in part of the expansion cylinder mass being driven back into the crossover passage immediately after the filling period, as the expansion piston finishes its stroke to TDC. The greatest consequence of this will be larger heat transfer losses caused by unnecessary flow back and forth through the crossover valve. It is for these reasons why the optimizer converged to a XOVO time that filled the expansion cylinder exactly when the piston reached TDC. This led to nominal opening times that were around 6°CA BTDC-e, allotting a few degrees for the ramp period and a few degrees for the fill time. Since the filling time depends on the crossover passage pressure and the engine speed, it can be expected that the XOVO time will need to be adjusted in accordance with the operating speed and load.

The XIVO time was found to have very little correlation with any of the output variables (see Figure 7.13). Its timing only affects the initial pressure differential across the valve: opening too early causes the pressure in the crossover passage to be higher than the cylinder, resulting in a short back-flow period; opening too late causes a blow-down period to occur into the crossover passage, resulting in a higher-than-normal initial gas velocity through the valve. The effect of these unintended flow characteristics would be increased heat losses, which reduce both $\bar{p}_x$ and the net IMEP. The XIVO should therefore coincide with the equalization of pressure between the compression cylinder and crossover passage. The use of a one-way style valve would be beneficial in this respect, as it would automatically account for changes in crossover pressure with engine speed. Developing a reliable, fast-acting, one-way valve that can withstand pressures in excess of 20 bar would be a challenging task, however.

The reader may have noticed the very low correlation between the SOC and the IMEP in Figure 7.13, which contradicts the empirical results presented in Chapter 6. However, this is somewhat misleading. The majority of the optimization runs can be separated into two
categories: early XOVC timing, which leads to a higher value of $\bar{p}_x$, and late XOVC timing, which leads to higher values of IMEP. By segregating the data based upon XOVC timing, stronger correlations between SOC and IMEP were found, but the trends were still clearly dominated by the XOVC timing. It is also known that gains in IMEP realized with advanced spark timing come from shorter burn durations, which were neglected in the numerical simulation. The simplicity of the combustion model used is one of the primary deficiencies of this investigation. A more advanced scheme that has provisions for combustion rate changes with temperature and flow field conditions is required for a more comprehensive assessment of the split-cycle engine using numerical techniques.

Regardless of any model deficiencies, it is clear that keeping the crossover outlet valve open while the compression piston finishes its stroke to TDC is beneficial to improving the net IMEP of the engine. Thus, the flow from the crossover passage into the expansion cylinder can be thought of in two stages: the blow-down stage, where the pressure rapidly rises in the expansion cylinder and the flow velocity is very high; and the pumping stage, where the remaining $\sim 20^\circ$CA of the compression stroke forces more charge into the expansion cylinder while its piston recedes. Again, the benefits of this two-stage process on combustion may not be fully realized by the numerical model. The higher the intensity of the blow-down process, the more distributed the turbulent length scale will be, and the greater its effectiveness on wrinkling a small developing flame. The pumping stage then continues to supply the energy required to sustain a turbulent flow-field as the flame ball grows. In an ideal and hypothetical scenario, the pumping stage would supply a stream of reactants to an already developed flame front, generating a stationary distributed reaction zone.

**Summary and Recommendations**

Based on the results of the numerical study, the valve timing changes listed in Table 7.5 are recommended for the split-cycle engine developed in this work. These values are expected to maximize the net IMEP and also increase the average crossover passage pressure.

<table>
<thead>
<tr>
<th>Valve Event</th>
<th>Current</th>
<th>Recommended</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>XIVO (*CA)</td>
<td>42</td>
<td>31</td>
<td>BTDC-e</td>
</tr>
<tr>
<td>XOVO (*CA)</td>
<td>15</td>
<td>7</td>
<td>BTDC-e</td>
</tr>
<tr>
<td>XOVC (*CA)</td>
<td>27</td>
<td>25</td>
<td>ATDC-e</td>
</tr>
</tbody>
</table>

*All values in reference to 0.025 mm of measured lift.*
It can be seen that the crossover inlet valve is currently opening too early, although this is unlikely to be of any significant detriment. To reduce the inlet valve’s duration by 11 °CA, a nominal ramp-to-ramp lobe duration of 25 °CA is required. The crossover outlet valve is also currently opening too early, but closing at approximately the right time. By phasing the opening point closer to TDC the turbulence intensity at the time of ignition may increase, although it is not clear what affect this will have on combustion. To reduce the crossover outlet valve duration by 10 °CA, a nominal ramp-to-ramp lobe duration of 26 °CA is required.

A complete summary of the crossover valve timing effects on engine performance is given in Table 7.6 on the following page.
Table 7.6: Effects of early and late crossover valve timing.

<table>
<thead>
<tr>
<th>Valve Event</th>
<th>Ideal Timing</th>
<th>Too Early</th>
<th>Too Late</th>
</tr>
</thead>
<tbody>
<tr>
<td>XIVO</td>
<td>When the compression cylinder pressure equalizes with the crossover passage pressure.</td>
<td>$p_{\text{cyl}} &lt; p_x$, causing back flow from the crossover passage into the compression cylinder. Non-isentropic fluid transfer through the valve will lead to heat transfer losses.</td>
<td>$p_{\text{cyl}} &gt; p_x$, higher flow velocity across the valve may lead to increased heat transfer.</td>
</tr>
<tr>
<td>XIVC</td>
<td>At TDC of the compression cylinder, with provisions to account for minor flow variations caused by fluid inertia or harmonic wave effects.</td>
<td>Compression piston has not reached TDC and transfer of intake mass from cylinder into crossover passage is incomplete. Reduction in volumetric efficiency results.</td>
<td>Valve closes after TDC-c, resulting in a back flow from the crossover passage into the cylinder and a subsequent reduction in volumetric efficiency.</td>
</tr>
<tr>
<td>XOVO</td>
<td>Slightly in advance of the expansion cylinder TDC, such that equalization of pressure between the crossover passage and the expansion cylinder occurs at TDC.</td>
<td>Exhaust valve may be open causing short-circuiting of air/fuel mixture into exhaust stream. Also, since the expansion piston is still on its up-stroke, the gas entering the cylinder may be partially pushed back in the crossover passage. Heat transfer losses can be expected during this unnecessary flow reversal.</td>
<td>Ideally the crossover valve should open as close to TDC-e as possible and late opening has little effect other than to delay the SOC due to the finite valve duration.</td>
</tr>
<tr>
<td>XOVC</td>
<td>Approximately when the compression piston reaches TDC or several degrees after that time to allow for the flow delay through the crossover passage.</td>
<td>The mass of fuel and air admitted into the expansion cylinder will be small, resulting in a build-up of pressure in the crossover passage. Higher compression IMEP will result.</td>
<td>Expansion of the charge leads to low peak combustion pressure and reduced IMEP. Lower crossover passage pressures were also observed.</td>
</tr>
</tbody>
</table>
Chapter 8

In-Cylinder Turbulence and Combustion Regime Estimation

8.1 Purpose and Scope

To this point, it has been presumed that the flow field within the combustion chamber of the split-cycle engine is highly turbulent. This was implicitly substantiated by the combustion duration results presented in Chapter 6. However, the actual turbulent flow characteristics present in the engine remain to be evaluated. This chapter details several experiments that were formulated to qualitatively and quantitatively analyse the downstream flow emerging from a reverse poppet valve (RPV) under choked flow conditions. These experiments were intended to provide insight into the level of turbulence present within the split-cycle combustion chamber at the SOC, which were subsequently used to estimate the combustion regime. The results presented in this chapter should be considered approximations, since the experiments were performed off-line from the engine using the flow apparatus presented in Section 8.2. A visual analysis of the flow field is presented in Section 8.3, obtained using planar laser techniques and high-speed imaging. Section 8.4 focuses on quantitatively assessing the velocity characteristics of the flow by method of laser Doppler velocimetry (LDV). The velocity measurements were taken for two separate experimental configurations: an unconfined flow discharging through the RPV onto a flat plate (Section 8.4.6), and flow through the RPV into a confined volume that mimics the combustion chamber of the engine (Section 8.4.7). The results of these experiments were then used to estimate the turbulence characteristics and combustion regime within the split-cycle engine (Section 8.5).
8.2 RPV Flow Apparatus

Visual access to the flow inside the engine was impractical due to the large physical packaging requirements and high cost of fabricating an optically accessible combustion chamber. Therefore, the flow conditions in the cylinder during the inflow period from the crossover passage were physically modelled using a bench-test apparatus. The apparatus, shown in Figure 8.1, consists of a small aluminum chamber that houses the reverse poppet valve and represents the crossover passage. This will be hereafter referred to as the RPV chamber. The downstream side of the valve is open to the atmosphere, but the RPV chamber has provisions for attaching a secondary volume on the bottom (see Section 8.4.7). Once the RPV chamber is pressurized, the gas pressure holds the valve in the closed position until it is pulled open by an electrical solenoid actuated using a push-button switch. Both the solenoid and the chamber are mounted to a modular extruded aluminum frame, which allows for adjustment of the valve travel. The frame is securely fastened to a 76 cm x 122 cm x 110 cm Newport optical table for the purpose of aligning the laser devices covered in Sections 8.3.2 and 8.4.3. Note that the configuration shown in Figure 8.1 was used for the flow visualization experiments only (Section 8.3); the velocity measurements were taken using the same basic apparatus, but with a different laser beam source and chamber orientation. The details of the latter experiment will be covered in Section 8.4.

Figure 8.1: Photograph of RPV flow visualization apparatus.
A breakout view drawing of the RPV chamber is shown in Figure 8.2. The chamber is comprised of two parts: the bottom part, which has an internal cavity that matches the geometry of the valve seat and its surrounding area; and the upper part, which has a press-fit bronze valve guide and is dowelled to the lower section to ensure the valve remains concentric with its seat. The upper and lower sections of the pressure chamber are bolted together and sealed using copper gasket. A Viton® valve stem seal prevents any significant air leakage through the valve guide. The pressurized air enters into the chamber through a mixing tee where seeding particles are added to the flow (see Section 8.3.3). Note that the mixing tee was not used for the LDV experiments; see Section 8.4.4 for details.

The internal volume of the chamber is approximately 73 cm$^3$ and smoothly tapers from an internal bore diameter of 36 mm to a 19 mm opening. The 44° valve seat is identical to that found in the engine. Figure 8.3 shows a fully dimensioned schematic of the RPV chamber, including the lift height of the valve, which was set to 3.5 mm for all experiments.

The solenoid used to actuate the valve is a pull-type, 120 VAC unit manufactured by Guardian Electric Manufacturing Co. (model 18-I-120A) energized through a 24 VDC relay made by Omron Corporation (model G7L-1A-TUBJ-CB). The solenoid has a maximum pulling force of 93.4 N over a plunger distance of 3.175 mm. When the chamber is pressurized, the force acting of the backside of the poppet valve is significantly greater than the pulling capacity of the solenoid. The cost of a solenoid with adequate pulling force was prohibitive, so a coil spring aid was installed between the pressure chamber and the solenoid connecting block, as shown in Figure 8.2. The pre-load on the spring when the valve is closed helps offset the pressure force acting on the valve head. Shims placed under the spring allow for pre-load adjustment without altering the lift height of the valve.

Experimental test results from Chapter 6 showed that realistic pressure values within the crossover passage are on the order of 17–19 bar over the range of engine speeds tested. For the experiments in this chapter, compressed shop air regulated to 6.2 bar was used due to its availability within the laboratory. The drawback to using a lower pressure is that the duration of time for which the flow is choked is reduced. Regardless of this limitation, the flow is choked at the beginning of the discharge, and the initial exit velocity is assumed to be the same for both the flow experiment and the actual engine. The absence of fuel, approximately 9.5% by volume under stoichiometric conditions, was assumed to have a negligible effect on the flow characteristics as it only alters the density by approximately 4%. The higher gas temperatures in the engine compared with the experimental apparatus, however, are estimated to increase the kinematic viscosity by approximately 100%. The consequences of this change will be discussed in Section 8.4.8.
Figure 8.2: Breakout view of pressure chamber used for RPV flow measurements.
8.3 Flow Visualization

8.3.1 General

For the purpose of better understanding the structure of the flow field inside the combustion chamber of the engine, high-speed images of the flow exiting the RPV chamber apparatus were obtained. A stationary flat plate placed below the outlet of the valve was used to represent the piston in the engine. The effect of piston motion was assumed to be negligible since the change in clearance height is only 1.5 mm in the last 15°CA approaching TDC, which roughly corresponds to the 2 ms discharge period at 1200 RPM. The outlet side of the valve flow was unconfined at the edge of the plate and free to expand to atmospheric pressure.
8.3.2 Hardware and Configuration

The flow images shown in this work were taken using a Photron Mini UX100 high speed camera at a sampling rate of 8000 fps and a 1280 x 616 pixel resolution. Cross-sectional illumination of the flow was accomplished using a 5 mW Optikon model LM2P helium-neon laser expanded through a cylindrical lens. The lens created a laser sheet with a Gaussian light intensity distribution, which was focused between the valve outlet and the plate. The sheet bisected the flow area in a plane that was parallel to the valve stem and perpendicular to the camera.

The flow impingement plate measures 140 x 140 x 12 mm and was centred at a distance of 11 mm below of the bottom surface of the pressure chamber. Additional attempts were made to take images with the plate closer to the valve, but concluded unsuccessfully due to inadequate light exposure for the desired frame rate.

8.3.3 Flow Seeding

The mixing tee in Figure 8.2 was used to seed the air with solid particles that become illuminated while passing through the laser sheet. W-210 Zeeospheres $^{\text{TM}}$ micro-particles, manufactured by 3M Company, were used for this experiment. According to the specification data, these particles are white ceramic spheres with a 95$^{\text{th}}$ percentile size of 12 microns. To seed the flow, a small quantity of particles, roughly the size of a pea, were placed in the bottom of the mixing tee before pressurization of the RPV chamber. The particles were then entrained into the air during the pressurization process. To minimize the chance of the suspended particles from settling, the experiment was initiated immediately following pressurization of the chamber.

8.3.4 Visualization Results and Discussion

Figure 8.4 shows a sequential series of images taken from a pressurized flow discharging through a reverse poppet valve over a duration of 2 ms. It should be noted that the flow appears to be asymmetric; however, this is due to light dispersion of the laser sheet, which enters the image from the left-hand side.

Close observation of the frame at time $t = 0.125$ ms, shows the flow is emerging with a dispersive cross-section, as one might expect from any standard orifice flow. However, due to the angled nature of the valve seat, the flow is converging upon itself along the vertical
Figure 8.4: Time-lapse photographs of a pressurized flow discharging through a reverse poppet valve and impinging on a flat plate.
In a 3-dimensional space, the flow can be envisioned as an inverted cone. This is in direct contrast to a standard poppet valve, where the flow disperses radially away from the valve head. The converging flow from the reverse poppet valve appears to form a solid jet around $t = 0.375\,\text{ms}$, and proceeds to impact the plate and spread radially over the next several frames. At $t = 0.750\,\text{ms}$, a recirculation vortex begins to appear on the outer edge of the flow as it travels horizontally across the plate. The coherent nature of this vortex is apparent by its time scale, which outlasts the visualization period.

From the images of Figure 8.4 and the known plate distance, it can be approximated that the flow convergence mid-point is approximately 5–7 mm below the cylinder head (chamber) surface. This is of particular interest in relation to the engine, as the cylinder clearance height would vary with XVO timing. The current split-cycle engine configuration provides roughly 2–3 mm of clearance height at the time of XVO, and thus the flow will impinge on the piston crown before it fully converges upon itself. The coherent vortex present in Figure 8.4 may or may not develop in the actual engine, since the solid jet is unlikely to form and the chamber height is significantly shorter. It is reasonable to assume that the development of any coherent structure will slightly prolong or delay the turbulence, similar to how intake tumble and/or swirl break down at the end of the compression stroke. In any case, the self-converging nature of the flow would be likely to generate significant turbulence.

### 8.4 Velocity Measurements

#### 8.4.1 General

Due to the random nature of turbulent flow, no general analytic solution of its fluid motion exists. However, empirical data can be used to formulate a statistical approach by taking velocity measurements in the field of interest. More specifically, turbulence can be quantified as the fluctuating component of velocity in relation to the time-averaged mean. The large spread of such fluctuations causes rapid mixing and enhances the rates of momentum, mass, and heat transfer to several orders of magnitude above molecular diffusion alone.

This section focuses on the experimental techniques and results used to measure fluid velocity within a mock combustion chamber, in both unconstrained and constrained (pressurizing) conditions, for a flow emerging through a reverse poppet valve. The unconstrained case is similar to the visualization experiment, where the flow impinged upon a flat plate but was free to expand to the atmosphere. The constrained case used an optical chamber on the discharge side of the flow to allow for pressurization and account for temporal effects.
8.4.2 Laser-Doppler Velocimetry (LDV)

Laser Doppler velocimetry (LDV), synonymous with laser Doppler anemometry (LDA), is a non-intrusive method of measuring the velocity of a fluid at a point within a flow field. The technique has directional sensitivity, high spatial and temporal resolution, and tri-axial component capability. For these reasons it is the ideal resource for measuring highly turbulent flows, where the flow is three-dimensional, the frequencies may be large, and flow reversals are likely. It does, however, require flow seeding, to be discussed in Section 8.4.4.

Measuring Principles

LDV is based upon the scattering of high-intensity light by small particles moving within the flow. The technique relies upon the Doppler effect, which, for optics, is a measurable shift in light frequency when the light source is moving or light is scattered off of a moving object. In dual-beam LDV, which is the method used in this work, two coherent laser beams intersect at an angle of $2\alpha$ forming a measurement volume in their overlap, as illustrated in Figure 8.5. When a particle suspended within the flow passes through this measuring volume, light from both beams is scattered simultaneously. The velocity component that is normal to the bisector of the beams will scatter the light differently from each beam on account of the individual Doppler effects. In simpler terms, for whichever component of velocity that is being measured, the particle is moving towards one beam and away from the other, creating two different reflected-light frequencies.

The superposition of these two reflected light waves, at differing frequencies and magnitudes, leads to optical interference, as shown in Figure 8.6. The superimposed light wave is made up of both high-frequency and low-frequency components, the latter of which is known as the beat frequency, $f_b = f_1 - f_2$. In LDV terminology, the beat frequency is simply known as the Doppler frequency and can be easily and accurately measured since it is of many orders of magnitude smaller than the laser light frequency ($\sim 6 \times 10^{14}$ Hz difference).
It can be shown that the Doppler frequency, $f_D$, is directly proportional to the particle velocity component that is perpendicular to the bisector of the two laser beams, $u_{\perp p}$ [151]:

$$u_{\perp p} = \frac{\lambda_0}{2 \sin \alpha} f_D$$  \hspace{1cm} (8.1)

where $\lambda_0$ is the wavelength of the light source and $\alpha$ is the half-angle of the two laser beams. Thus, the term preceding the Doppler frequency in Equation (8.1) is a constant and it is for this reason why LDV is said to be without calibration [151]. By measuring the Doppler frequency of the scattered light beams, the particle velocity can be determined directly.

The reflective light waves of a single particle passing through the measurement volume are known as *Doppler bursts* and consist of the Doppler frequency superimposed on a Gaussian distribution, an example of which is shown in Figure 8.7. The Gaussian distribution of the burst comes from the characteristic Gaussian intensity of the laser beam and is often referred to as the *pedestal*. The pedestal is easily removed using a high-pass filter during processing of the Doppler signal.

The manner by which the Doppler frequency is produced means it will always yield a positive value, regardless of the particle direction. To circumvent this directional ambiguity, a technique known as *frequency shifting* must be employed. For reasons of clarity, the fringe model will be discussed first, followed by a description of frequency shifting.
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Fringe Model

The fringe model is an alternative explanation for operation of the dual-beam LDV system that does not make direct reference to the Doppler effect. According to Goldstein [62], the fringe model is not entirely correct and can predict unrealistic results depending on the relative particle-to-fringe size. Regardless, it is often referred to in LDV literature and is easier to visualize than the Doppler model.

The fringe model is based upon the observation that the intersection of two beams forms a measuring volume that consists of equally spaced interference fringes: regions of light and dark, as shown in Figure 8.8.

![Figure 8.8: Illustration of LDV fringe model.](image)

The spacing of the fringes is a constant and determined from Equation (8.2), calculated from the wavelength and angle of incidence of the laser beams:

\[ d_f = \frac{\lambda_0}{2 \sin \alpha} \quad (8.2) \]

When a small particle passes through the fringe pattern, the reflected light oscillates sinusoidally with the fringes. The frequency of this oscillation is a function of the particle velocity and fringe spacing, \( f = \frac{u_{\perp p}}{d_f} \). Substitution of Equation (8.2) for the fringe spacing yields:

\[ f = \frac{2 u_{\perp p} \sin \alpha}{\lambda_0} \quad (8.3) \]
Close inspection of Equation (8.3) reveals that it is identical, albeit rearranged, to the Doppler model, Equation (8.1). Thus, one can visualize the Doppler frequency as the alternating frequency of light intensity as a particle passes through the fringes of the measuring volume. This is generally correct, although interested readers can learn about the limitations of the fringe model in Goldstein [62].

**Frequency Shifting**

As previously mentioned, a particle passing perpendicular to the fringes will produce the same Doppler frequency regardless of its directional sign and when a particle approaches zero velocity, the output signal also converges to zero. To distinguish between a positive and negative velocity of the same magnitude, a slight shift in the frequency of one of the paired laser beams can be achieved using a Bragg cell.\(^1\) The purpose of altering the beam frequency is to create moving fringes inside the measuring volume in a predefined direction: from the beam with the higher frequency to that with the lower. The detected signal is then the superposition of the Doppler frequency and the shift frequency. For example, a particle travelling in the direction of the fringes would have a detected frequency of the Doppler frequency minus the shift frequency and this would be considered a negative or reverse flow. Figure 8.9 illustrates the concept of frequency shifting to eliminate directional ambiguity of the velocity measurement. It is evident from Figure 8.9 that a particle with zero velocity will have an apparent Doppler frequency corresponding to the magnitude of the shift frequency. Therefore, the shift frequency is later subtracted from the detected signal in order to obtain the actual Doppler frequency and its corresponding velocity.

**Effective Frequency Shifting**

For the LDV system used in this work, a frequency shift of 40 MHz exists by default in one beam of each component pairing, meaning the fringes are always moving at 40 MHz in the direction from the shifted to the unshifted beam. The number of fringes crossed by a particle will ultimately depend on the flow direction relative to the fringe motion. Therefore, to maximize the number of cycles, the dominant flow direction should oppose that of the fringe

\(^1\) A Bragg cell is an acousto-optic modulator (AOM) that uses a piezoelectric crystal attached to an optic material through which light can be diffracted to a desired frequency. Other frequency shifting devices are available; however, the Bragg cell is common in LDV applications.
motion. This can be configured in the LDV software, discussed in Section 8.4.3, without the need to re-orient the optical probe in the test environment.

The 40 MHz shift allows for the most efficient use of the signal processor and easy removal of the pedestal using a high-pass filter. However, a 40 MHz shift corresponds to a maximum negative velocity of approximately 136–144 m/s (dependant on the inherent beam frequency), which may not be necessary for the majority of flow conditions. In addition, the larger the frequency shift, the lower the measurement resolution. Therefore, an effective frequency shift is implemented using electronic mixing circuitry in the optic receiver, enabling the user to down-mix the frequency shift to a lower value. Equation (8.4) shows mathematically how a 40 MHz frequency is mixed with the effective shift frequency and then subtracted from the incoming signal, which consists of the Doppler frequency and the 40 MHz fringe motion.

\[
\begin{align*}
\text{Incoming Signal} & : |(f_{\text{doppler}}) \pm (40 \text{ MHz})| \\
\text{Mixing Process} & : (40 \text{ MHz}) \pm (f_{\text{effective shift}}) \\
\text{Output Signal to Processor} & : (f_{\text{doppler}}) + (f_{\text{effective shift}}^*)
\end{align*}
\]

\((8.4)\)
In Equation (8.4), the signed portion of the incoming signal depends on whether the seeding particle is moving with the fringes or against the fringes. The signed portion in the mixing process is user configurable based on the presumed dominant (or positive) flow direction. This allows for a virtual direction change in the processing software without the need to physically re-orient the projection of the laser beams in the test environment.

Two factors need to be assessed when selecting an effective frequency shift magnitude:

(i) The maximum negative velocity to be measured  
(ii) The signal processor’s minimum cycle requirement

According to the TSI Model 9230 instruction manual [137], the general rule of the thumb when measuring flow reversals is to select a frequency shift that is twice the Doppler frequency corresponding to the maximum negative velocity anticipated. This is based upon the following calculations.

When no frequency shift is present, the number of fringes, \( N_F \), in a measurement volume is given as:

\[
N_F = \frac{d_{mv}}{d_f}
\]  
(8.5)

and the Doppler frequency, \( f_D \), as:

\[
f_D = \frac{|u_{\perp p}|}{d_f}
\]  
(8.6)

where \( d_{mv} \) is the mean diameter of the measuring volume and \( d_f \) is the fringe spacing, which is known from the laser light properties.

Correspondingly, the number of cycles, \( N_1 \), or fringes that a particle crosses for a given trajectory angle, \( \theta \), is given by Equation (8.7). When \( \theta \) is zero degrees, \( N_1 = N_F \) since the particles are travelling perpendicular to the fringes, and when \( \theta = 90^\circ \), \( N_1 = 0 \) since the particles are moving parallel to the fringes.

\[
N_1 = N_F \cos \theta
\]  
(8.7)

The residence time, \( t \), of a particle inside the measuring volume can be estimated by dividing the measuring volume diameter by the absolute perpendicular particle velocity (e.g. \( t = d_{mv}/|u_{\perp p}| \)). With frequency shifting, one can deduce that for a stationary particle residing in the measurement volume for time \( t \), the number of moving fringes, \( N_s \) to pass the stationary particle is:

\[
N_s = t \cdot f_s = \frac{d_{mv}}{|u_{\perp p}|} \cdot f_s
\]  
(8.8)

where \( f_s \) is the magnitude of the shift frequency.
Thus the total number of cycles, $N$, is the summation of the shifted, $N_s$, and non-shifted, $N_1$, cycles:

$$N = N_1 + N_s$$  \hspace{1cm} (8.9)

Substitution of Equations (8.6), (8.7), and (8.8) into (8.9) yields:

$$N = N_F \left( \cos \theta + \frac{f_s}{f_D} \right)$$  \hspace{1cm} (8.10)

For the special case of a complete flow reversal ($\theta = 180^\circ$) and where $N_F = N$, Equation (8.10) reduces to:

$$\frac{f_s}{f_D} = 2 \rightarrow f_s = 2f_D$$  \hspace{1cm} (8.11)

In other words, when the number of cycles is equal to the static number of fringes in the measurement volume ($N_F = N$), no fringe loss occurs provided the shift frequency is twice the Doppler frequency corresponding to the maximum negative velocity.

The user-selectable frequency shift is applied through TSI’s “FIND” for Windows” software program.

**Measurement Volume Shift Through a Planar Window**

For the experiment involving pressurization of an enclosed chamber (mock engine cylinder), the laser beams optically accessed the inside of the chamber through a 12 mm thick window made of poly-methyl methacrylate (PMMA), more commonly known as acrylic. Since the refractive index of PMMA does not coincide with that of air, a shift in the measurement volume position occurs due to the refraction of light as it passes through each side of the window, as shown in Figure 8.10. Provided the LDV head is aligned perpendicular with the optical window, the refraction of both beams is symmetrical and the on-axis position of the measurement volume $x_{mv}$ can be approximated from:

$$x_{mv} = x_0 \frac{n_3}{n_1} + t \frac{n_3}{n_1} - t \frac{n_3}{n_2}$$  \hspace{1cm} (8.12)

where $x_0$ is the undisturbed focal length of the beams, $t$ is the window thickness, and $n_1$, $n_2$, $n_3$ are the refractory indices of the mediums before, including, and after the window, respectively. Both $x_0$ and $x_{mv}$ are distances referenced to either plane bisecting any two fluid mediums. Equation (8.12) was derived from geometry, utilizing Snell’s Law$^2$ of refraction, and with the assumption that $\tan \alpha \approx \sin \alpha$ for small angles. The index of refraction is a

---

$^2$ Snell’s Law: $n_1 \sin \alpha_1 = n_2 \sin \alpha_2$
function of the light source wavelength, which in this case is 514.5 nm and 488 nm for the green and blue beams, respectively. However, the difference in refraction is so small that it had a negligible effect on the location of measuring volume.

![Diagram of beam refraction through on-axis optical window.](image)

**Figure 8.10:** Illustration of beam refraction through on-axis optical window.

Using $n_1 = n_3 = 1$ for air and $n_2 = 1.49$ for PMMA, the difference in focal length, $x_{mv} - x_0$, was calculated to be 4 mm; the LDV probe was positioned further away from the experimental chamber, accordingly. The uncertainty in focal location with respect to the desired measuring plane is estimated to be ±1 mm.

### 8.4.3 LDV Hardware and Configuration

The velocity measurements acquired for this work were done using a two-component LDV system from TSI® Incorporated. A schematic of the test system is given in Figure 8.11. An Innova 70 argon-ion laser was used as the light source, with the power output set to 4 W. The collimated beam that it generates is sent to a TSI® 9201 COLORBURST® cell, where it is divided into four beams of equal intensity, two green beams at 514.5 nm and two blue beams at 488.0 nm. One beam from each pair is frequency-shifted by 40 MHz using a Bragg cell and all beams are then passed to the optical probe via fibre-optic cable.

From the optical probe the paired beams emerge in transverse planes and converge to a focal point, which forms the measuring volume of the system. The fibre-optic couplers were adjusted to balance the intensity of all four beams in order to prevent a directional bias.

This particular system operates in backscatter mode, meaning that the reflected light from the particles is collected by the same optical probe and coupled into a single receiving fibre.
The incoming light is sent to a TSI® 9230 Colorlink® where the beam colours are again separated and converted into digital signals. The Colorlink® is also used to drive the Bragg cell mentioned in Section 8.4.2.

Digital signals from the Colorlink® are routed to an IFA 755 digital burst correlator, where each particle burst is filtered and analysed. User-adjustable band-pass filters are used to reduce noise and eliminate the pedestal. The signals are amplified, followed by a signal-to-noise ratio validation, and finally the Doppler frequency is extracted and sent to the computer for display and logging.

### 8.4.4 Flow Seeding

Initially, flow seeding was achieved using Zeeospheres™ particles in the same manner that was employed for the flow visualization experiment (Section 8.3.3). However, after a short while, particle residue on the valve seat area accumulated to the point of preventing the valve from fully closing. While this did not present as an issue in the flow visualization
experiments, it did cause erroneous velocity readings (Doppler bursts) to occur before the valve was opened. The solution to this problem was to seed the flow with a vaporized fogging oil, which had no adverse effect on valve sealing.

Introduction of fog into the air stream was accomplished by flowing compressed air through a pressure vessel containing a small quantity of fogging oil in the bottom, which was wicked up into a resistance heating coil and subsequently vaporized. A breakout view of the apparatus is shown in Figure 8.12. The heating coil was made from 14 turns of 0.8 mm Nickel-Chromium wire suspended above the fogging fluid, which was drawn into the heating coil by a polyester felt wick. A 120 VAC variable output power supply was used to provide electrical current to the heating coil, and found to work best at approximately 9 A. This seeding method also had the additional benefit of slightly heating the air, which better mimics the temperature and kinematic viscosity of the fluid inside an engine. However, the temperature was neither precisely controlled nor measured.

![Breakout view of apparatus constructed to generate seeding particles (fog) for LDV measurements.](image)

The exact size of the suspended fog particles is not known; however, the same fogging oil was found to have a mean particle size of 2.840 µm by Brown [24], when used in a commercial fogging machine. In a separate study, particles of a similar size, formed from a vaporized water/fogging oil mixture, were found to adequately follow velocity fluctuations up to 5 kHz at a mean velocity of 30 m/s, and maintain acceptable data rates up to 100 m/s [45]. While
flow through the valve throat area will be sonic, the measurement points of interest are far enough away from the throat area that the fogging oil is believed to be adequate for the application and its intended outcomes.

8.4.5 Velocity Analysis

Stationary Flows

For stationary or non-transient flow, which will be shown in Section 8.4.6 to be the approximate case for the flat plate experiment, the flow turbulence can be interpreted as the root-mean-squared (RMS) of the fluctuating component of velocity:

\[
\text{u}_\text{rms} = \sqrt{\frac{1}{N} \sum_{i=1}^{N} (u_i - \bar{u})^2} \quad (8.13)
\]

where \( N \) is the number of discrete data points, \( \bar{u} \) is the mean flow velocity, and \( u_i \) are the instantaneous velocity fluctuations. Equation (8.13) is valid provided the mean flow is steady over the interval of interest.

Non-stationary Flows

For non-stationary turbulent flows, the mean velocity is time or phase-dependent. Thus, the fluctuating component of velocity, caused by flow instability, is superimposed with enforced changes in mean velocity. To separate the two terms, the individual velocity points were first fitted with a 10th-order polynomial curve, followed by evaluation of the turbulent fluctuations over discrete time intervals, calculated using:

\[
\text{u}_\text{rms}(\Delta t) = \sqrt{\frac{1}{N_{\Delta t}} \sum_{i:t_i \geq (t_m - \Delta t/2)}^{j:t_j < (t_m + \Delta t/2)} (u_i - \hat{u}_i)^2} \quad (8.14)
\]

where \( \Delta t \) represents the chosen time interval, \( t_m \) is the midpoint of the time interval, \( \hat{u}_i \) is the velocity at point \( i \) calculated using the polynomial regression, and \( N_{\Delta t} \) represent the number of data points that fall within the time interval. Since the Doppler bursts are acquired randomly with time, \( N_{\Delta t} \) will vary for each time window. When the magnitude of the fluctuations are also time-dependent, it is desirable to decrease the time window for improved accuracy. However, the length of the time interval is limited by the number of data points that falls within it.
Turbulence Intensity

For both stationary and non-stationary flows, the turbulence intensity, $TI$, can be calculated as a percentage according to Equation (8.15).

$$TI = \frac{u_{rms}}{u} \times 100\%$$  \hspace{1cm} (8.15)

### 8.4.6 Results of an Unconfined Flow Impinging on a Flat Plate

**Experimental Methodology**

For the unconfined case, the same plate that was used for flow imaging was placed below the outlet of the valve at a distance of 6 mm. This was the minimum height achievable without obstructing the pathway of the vertical laser component. Measurements were taken vertically at the mid point between the valve and the impingement plate, and along the radial direction in 5 mm increments, as depicted in Figure 8.13. Radial locations beyond 35 mm were not investigated since this would exceed the RPV chamber surface and this distance is also larger than the engine bore.

![Figure 8.13: Location of measurements for the flat plate LDV experiment.](image)

For each measurement point, the bandpass filter range was adjusted in the FIND™ software until the highest data rate (number of Doppler bursts) was achieved. The data was then checked for clipping by the bandpass filter, which was subsequently adjusted if necessary,
and the process was repeated. Once the correct range was established, the system was configured to capture a maximum of 10000 data points. However, only the first 100 ms of data were used to tabulate the results, which will be explained in the following section. The filter configurations and frequency shifts used in the experiment are given in Table 8.1, where \( u \) and \( v \) are the radial and axial velocity components, respectively.

### Table 8.1: LDV signal processor configuration for the flat plate experiment.

<table>
<thead>
<tr>
<th>Radial Location (mm)</th>
<th>Bandpass Filter(^1) (MHz)</th>
<th>Frequency Shift (MHz)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( u )</td>
<td>( v )</td>
</tr>
<tr>
<td>0</td>
<td>5-30</td>
<td>5-30</td>
</tr>
<tr>
<td>5</td>
<td>5-30</td>
<td>5-30</td>
</tr>
<tr>
<td>10</td>
<td>5-30</td>
<td>5-30</td>
</tr>
<tr>
<td>15</td>
<td>5-30</td>
<td>5-30</td>
</tr>
<tr>
<td>20</td>
<td>5-30</td>
<td>3-20</td>
</tr>
<tr>
<td>25</td>
<td>5-30</td>
<td>3-20</td>
</tr>
<tr>
<td>30</td>
<td>3-20</td>
<td>3-20</td>
</tr>
<tr>
<td>35</td>
<td>3-20</td>
<td>3-20</td>
</tr>
</tbody>
</table>

\(^1\) For a 10 MHz frequency shift, a low-pass filter of 23 MHz is also implemented.

### Results

The left-hand plot of Figure 8.14 is an example of the velocity points (Doppler bursts), measured over the duration of the discharge for the flat plate experiment. Initially, velocity values up to 80 m/s are shown, followed by a subsequent decrease as the pressure in the chamber depletes. The time scale of this depletion is extended due to the large volume of air in the seeding chamber. In the engine, the flow time is roughly two orders of magnitude smaller than the 2 s duration of the unconstrained discharge. Therefore, the time period used for statistical inference of the turbulence properties was limited to the first 100 ms of data. While this is still much longer than the engine flow time, the right-hand plot of Figure 8.14 shows that there are no significant changes or trends in the mean (\( \bar{u} \)) and fluctuating (\( u' \)) velocity components over time. The same is true for both velocity components, regardless of radial location. Therefore, for the early part of the discharge period, the flow can be approximated as steady. Using a longer time period for assessment also increased the number of data points available for inclusion. Data rates in excess of 5 kHz were achieved for this experiment.
The resulting mean velocities and corresponding velocity fluctuations are presented in Figure 8.15 as a function of the radial measurement location. Error bars show the standard error (SE) of the mean from three trial repetitions at each radial position. For the mean velocities, the largest SE was ±2.95 m/s in the $u$ component at a radial location of 5 mm. For the velocity fluctuations, all SE values were below ±1.65 m/s. The averaged numerical data is also summarized in Table 8.2, and includes the turbulence intensity (TI).

**Table 8.2:** Experimental LDV results for a RPV flow discharging onto a flat plate. Values averaged from three independent trials.

<table>
<thead>
<tr>
<th>Radius (mm)</th>
<th>$\bar{u}$ (m/s)</th>
<th>$\bar{v}$ (m/s)</th>
<th>$u'$ (m/s)</th>
<th>$v'$ (m/s)</th>
<th>$\text{TI}_u$ (%)</th>
<th>$\text{TI}_v$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>16.1</td>
<td>-21.0</td>
<td>16.3</td>
<td>15.4</td>
<td>101.3</td>
<td>74.0</td>
</tr>
<tr>
<td>5</td>
<td>19.4</td>
<td>-16.2</td>
<td>17.6</td>
<td>14.2</td>
<td>96.5</td>
<td>88.3</td>
</tr>
<tr>
<td>10</td>
<td>48.9</td>
<td>-32.7</td>
<td>18.9</td>
<td>16.0</td>
<td>38.6</td>
<td>49.2</td>
</tr>
<tr>
<td>15</td>
<td>37.0</td>
<td>-10.1</td>
<td>19.9</td>
<td>13.4</td>
<td>53.9</td>
<td>134.7</td>
</tr>
<tr>
<td>20</td>
<td>22.7</td>
<td>-8.4</td>
<td>16.6</td>
<td>12.2</td>
<td>74.0</td>
<td>145.2</td>
</tr>
<tr>
<td>25</td>
<td>16.6</td>
<td>-4.9</td>
<td>16.9</td>
<td>12.1</td>
<td>101.2</td>
<td>260.1</td>
</tr>
<tr>
<td>30</td>
<td>7.0</td>
<td>-0.2</td>
<td>12.4</td>
<td>9.1</td>
<td>189.6</td>
<td>2157.6</td>
</tr>
<tr>
<td>35</td>
<td>5.8</td>
<td>-0.2</td>
<td>9.2</td>
<td>8.2</td>
<td>159.2</td>
<td>1913.7</td>
</tr>
</tbody>
</table>

1 Relative to valve centerline (see Figure 8.13). Accuracy of ±0.5 mm.
Figure 8.15: Results for a pressurized flow discharging through a reverse poppet valve and impinging on a flat plate; mean velocity (top), and RMS velocity fluctuations (bottom). Error bars show standard error of three repeated measurements.

8.4.7 Results of a Flow into a Confined Volume

Experimental Methodology

To better mimic the flow field that occurs in the actual engine cylinder, velocity measurements were also taken for a flow discharging into an optically accessible, closed-volume chamber bolted to the outlet side of the RPV. A photograph of the assembly is shown in Figure 8.16. Note that the assembly is inverted (i.e. the valve is pulling downwards), which was done to prevent condensate from pooling in the measurement region. The chamber was made optically accessible for the laser by including a parallel-plane window on one side, as shown by the plan view schematic of Figure 8.17. Also shown, is the location of the 0–50 bar absolute pressure sensor (Kistler 4005B), used to acquire the chamber pressure during the flow period, and a valved outlet port used to relieve the pressure after each experiment.
Figure 8.16: Photograph of optical chamber attached to RPV flow apparatus. Note: the valve and chamber are upside-down to prevent condensate from collecting in the optical chamber.

Figure 8.17: Plan view of the chamber used for confined-flow velocity measurements.
The cross-sectional area of the optical chamber was designed to match that of the 67 mm engine bore, yielding the same volume for an identical clearance height. For the results presented in this section, the chamber height was set to 8 mm. The additional 2 mm, in comparison with the 6 mm flat plate experiment, was needed to distance the laser beam from the edge of the window, which was rounded from polishing and caused beam distortion as a result. Velocity measurements were taken on the same plane as the flat plate experiment, and the measuring point was centred vertically within the chamber. Measurements were taken at radii of 20 mm, 25 mm, and 30 mm, as this approximates the location of the spark plug relative to the crossover outlet valve and is indicative of the levels of turbulence subjected on the developing flame. Measurements closer to the valve were of no practical interest in this case, since the flame would be well established by the time it reached the crossover valve. At a radial distance of 35 mm, one of the \( u \) component beams was blocked from entering the chamber, and therefore measurements were not taken at this location.

Pressure in the optical chamber was sampled at 10 kHz, and data logging was commenced using the same digital trigger used to activate the valve solenoid. Unfortunately, the author was not able to synchronize the LDV and pressure data. Therefore, the time associated with each velocity measurement is chronologically arbitrary relative to the time scale of the pressure data. The lack of a synchronizing data marker also prevents superposition of individual trials, the implications of which will be discussed further in Section 8.4.8.

**Results**

Figure 8.18 compares the pressure measurements from the optical flow chamber with that of the engine’s expansion cylinder under motoring conditions at 1000 RPM. The engine trace has been converted from an angle-based to a time-based measurement by assuming the crankshaft speed is constant throughout the cycle, a valid assumption under motoring conditions. The two curves have been aligned based on the first visual detection of a pressure increase, although in the engine this is actually caused by compression of residual gases in advance of the crossover valve opening. Regardless, the choked flow period is occurring over a similar time frame for the two cases, since the peak pressure rise rate of the engine is about 200 times that of the flow chamber.

The highlighted region between \( t = 10 \text{ ms} \) and \( t = 20 \text{ ms} \) is of primary interest in terms of the turbulence properties. Velocity measurements taken within this region would indicate the rate of turbulent decay, along with the conditions that are present at the time of ignition. The latter is especially important regarding the early flame development period, when high mean (bulk) velocities are capable of displacing, and possibly quenching, the flame kernel.
Chapter 8: In-Cylinder Turbulence and Combustion Regime Estimation

Figure 8.18: Comparison of pressure traces: RPV flow into optical chamber versus motored engine at 1000 RPM.

The upper two plots in Figure 8.19 show the raw velocity measurements taken within the optical chamber during pressurization, for both the \( u \) (radial) and \( v \) (axial) directions. The bottom two plots show the corresponding root-mean-squared (RMS) velocity fluctuations \( (u' \text{ and } v') \), which were calculated according to Equation (8.13) over a discrete time interval of 5 ms and overlapping by 50%. Since the Doppler bursts are acquired randomly with time, the number of velocity points used to calculate each RMS data point varied, but the average number of data points in each bin was \( 30 \pm 10 \). The data used to produce Figure 8.19 was taken at a location of \( R = 25 \text{ mm} \).

8.4.8 Discussion of Results

The flat plate experiment provided insight into the magnitudes of velocity that are present during the initial discharge period, when the flow is choked. From Figure 8.15, it can be seen that mean velocities as high as 50 m/s are present in the radial direction, with fluctuations yielding localized values in excess of 70 m/s. With increasing radius, the mean radial velocity is approaching zero (i.e. \( \bar{u} \propto R^{-1} \)), which is expected, in order to satisfy the conservation of mass for an outward radial flow. Both components of velocity have peak magnitudes at \( R = 10 \text{ mm} \), which corresponds to the edge of the valve seat. The velocities at this location are higher than those at \( R = 0 \text{ mm} \) and \( R = 5 \text{ mm} \), since the flow is converging upon itself for these radii, and at some point must stop flowing inwards and reverse direction. As such, one would expect a stagnation point in the \( u \)-component at
R = 0 mm, which was shown in Figure 8.15 to not be the case. This is believed to be a biasing error introduced by bandpass filtering of the Doppler bursts, and will be addressed in the next section.

Figure 8.15 also shows the velocity fluctuations to range between 8 m/s and 20 m/s, with values slightly higher in the radial versus axial directions for a given measurement location. The fluctuations were found to be highest in the vicinity of the valve, where the flow is abruptly stopped in the axial direction and self-intersecting in the radial direction. Despite the mean axial velocity going to zero at the outer radii, the large fluctuations remain, which leads to the very large turbulence intensities (> 2000%) shown in Table 8.2.

The velocity characteristics during the main discharge period, when the flow is choked, can now be understood. However, based on the pressure traces of Figure 8.18, it is evident that the period of choked flow ends roughly 5 ms before ignition occurs. In a confined
volume, such as the engine, velocity measurements in the optical chamber indicated that high velocity magnitudes and fluctuations do still exist initially, but quickly diminish as the flow tapers off. Assuming the first Doppler burst measurements occur when the flow is nearing the end of its choked regime, by the time ignition occurs the velocity fluctuations will have reduced from $>15$ m/s to around 5 m/s. Any further delay and the level drops to 1–2 m/s, where it remains until the end of the measurement period. This highlights the significance of the ignition timing in relation to crossover valve timing. It also reveals that further advances in spark timing could potentially result in ignition difficulties, if the velocity fluctuations are increasing exponentially.

The flow chamber experiment, however, does not account for the fact that both pistons are moving in the actual engine during this time. After the initial discharge has occurred, flow into the expansion cylinder continues as the compression cylinder completes the remaining $\sim 20^\circ$CA of its up-stroke. The displaced volume is small, roughly 4% of the swept volume, but nevertheless constitutes a continuous supply of energy into the combustion chamber flow field. The crossover outlet valve is also closing throughout this period, which is likely to generate increasingly large localized velocities as the throat area reduces to zero. This is another aspect of the engine operation that is not captured by the flow chamber experiment.

Based on these observations, it can be concluded that the turbulence intensity is indeed very high during the choked flow period of fluid transfer between the crossover passage and the combustion chamber. However, if no supplementary flow is present after this initial discharge, significant turbulent decay will occur in the interim period before ignition. Therefore, the offset phasing of the cylinders is believed to play an important role in maintaining the elevated levels of turbulence within the combustion chamber, providing the time necessary for the crossover outlet valve to close, in preparation for ignition.

**Experimental Inaccuracies and Limitations**

The experiments used to evaluate the downstream turbulence characteristics of a choked flow through a reverse poppet valve were limited in their ability to replicate the true conditions that occur within the split-cycle engine. Experimental inaccuracies in the fluid properties and the flow characteristics, along with limitations of the measurement system, will now be discussed.

The temperature of the fluid entering the engine’s combustion chamber is expected to be in the vicinity of 150–200°C.\(^3\) Because the flow experiment used compressed air from the

---

\(^3\) Based on thermocouple measurements within the crossover passage.
building supply, the fluid temperature is roughly ambient or perhaps elevated slightly by the seeding apparatus (see Section 8.4.4). Using Sutherland’s law, Equation (8.16), for a temperature change from $T_0 = 298 \text{ K}$ to $T = 473 \text{ K}$, the calculated dynamic viscosity, $\mu$, of air would increase by a factor of 1.4.

\[
\mu = \mu_0 \left( \frac{T}{T_0} \right)^{3/2} \frac{T_0 + S}{T + S} \tag{8.16}
\]

where $\mu_0$ is the dynamic viscosity at $T_0$, and $S = 110.56 \text{ K}$ for air at a moderate pressure and temperature.

In addition to a higher temperature, the pressure in the engine is also roughly three times higher than that in the flow experiment. While pressure has no considerable effect on the dynamic viscosity [37], it does affect the density. The increase in density with pressure is somewhat offset by the decrease in density with temperature. Consequently, the density only increases by a factor of 1.8 from the flow experiment to the engine. Combining these two factors together, yields an overall decrease in the kinematic viscosity ($\nu = \mu/\rho$) by approximately 20%. Since the turbulent Reynolds number is inversely proportional to kinematic viscosity ($Re_T \propto \nu^{-1}$), the difference in fluid properties may result in a small underestimate of the turbulence intensity using the flow chamber experiment.

Further inaccuracy of the flow experiment is introduced by the use of a solenoid to open the valve instead of the camshaft lobe. Without knowing the displacement profile of the solenoid plunger over time, it is difficult to assess what effect, if any, this difference may have. Under choked flow conditions, the throat velocities should be the same regardless of the valve lift. However, the mass flow rate will depend on the throat area and the discharge coefficient, both of which are a function of the valve lift height. Thus, the rate at which the cylinder and chamber each fill will differ, as seen by the dissimilar slopes in Figure 8.18. The absence of piston motion and valve closure also affects the velocity field in the flow chamber, and will undoubtedly increase the turbulence levels present in the engine. Consequently, the turbulence intensity measured in the flow chamber is likely a conservative estimate.

Finally, the LDV system used to measure the flow velocity imposes additional limitations on experimental accuracy. Configuration of the LDV system parameters is increasingly difficult due to the non-stationary flow present in this application. For a typical stationary flow, many of the LDV processing parameters (e.g. bandpass filters, frequency shift, burst threshold voltage, etc.) are configured while the system is acquiring data; this allows the user to maximize the data rates and signal-to-noise ratio in real-time. The very short time scale of this experiment means adjustments must be made in iteration with post-processing.
of the data. To optimize each parameter in this manner is impractical, especially since the optical window of the flow chamber requires disassembly for cleaning every 3–4 discharges. Optimum data rates and signal-to-noise ratios have therefore not been achieved.

Restrictions on the measurable velocity range were also an issue for this experiment. In a region where very large velocity fluctuations occur in both the positive and negative flow directions, such as that below the valve, the bandpass filters cannot be adequately configured to capture the entirety of the velocity range. To acquire large positive velocities, a sacrifice in the negative velocity range must be made, effectively biasing the measurement in the positive direction. For example, a 5–30 MHz bandpass filter with a 5 MHz frequency shift will measure particle velocities from approximately 0 m/s to 90 m/s. By increasing the frequency shift to 10 MHz, velocities down to roughly −20 m/s can be measured, but the upper limited is decreased to around 50 m/s due to a 23 MHz low-pass filter. Therefore, the system is simply incapable of measuring the entire velocity range, for any of the pre-defined filter configurations, when very large velocity fluctuations are present.

One method that could be employed to circumvent this issue is re-constructing the velocity measurements through superposition. In other words, taking repeated velocity measurements at the same point, using different bandpass filters and frequency shifts to process the data, and then superimposing the results. However, since the flow is non-stationary, a reference time stamp would be required in each data set for correct temporal alignment. The author had attempted to do so using the solenoid trigger as a reference marker, but was unable to configure the LDV system to acquire the signal.

### 8.5 Estimates for Turbulence and Combustion

#### 8.5.1 Laminar Flame Properties

The parameters that will be used to evaluate the turbulence-flame interaction, and ultimately predict the combustion regime, rely on the laminar flame properties for nondimensionalization. The laminar burning velocity, $u_L$, was calculated according to Equation (8.17), which, according to Heywood [70], is appropriate for methane fuel. The laminar burning velocity at a known reference state is $u_{L,o}$, and $\rho_u/\rho_{u,o}$ is the unburned gas density ratio between the post-compression engine conditions and the reference state. A value of $n = 0.19$ was used based on the empirical results provided by Heywood [70].

$$u_L = u_{L,o} \left( \frac{\rho_u}{\rho_{u,o}} \right)^n$$  \hspace{1cm} (8.17)
Using reference laminar flame speeds of \( u_{L,o} = 0.30–0.35 \text{ m/s} \) [91], for methane-air equivalence ratios of \( \phi = 0.8–1.0 \), respectively, the density-corrected engine laminar flame speeds ranged from \( u_L = 0.55–0.57 \text{ m/s} \).

The laminar flame structure presumably consists of three layers [91, 113]:

1) The preheat layer, \( \delta_{L,ph} \), which has a non-dimensional thickness \( \mathcal{O}(1) \).

2) The fuel consumption or inner layer, where fuel is consumed and \( \text{H}_2 \) and \( \text{CO} \) are formed; approximately 1/10 of the preheat layer thickness.

3) The downstream oxidation layer, where \( \text{H}_2 \) and \( \text{CO} \) are consumed; thickness \( \ll \delta_{L,ph} \).

The preheat layer is considerably larger than the other two layers, the combination of which will hereafter be referred to as the reaction layer, \( \delta_R \). It was therefore assumed that the laminar flame thickness is approximately equal to the preheat layer thickness, \( \delta_L \approx \delta_{L,ph} \), the latter of which is based on the unburned gas properties in accordance with Equation (8.18), where \( k \) and \( c_p \) are the thermal conductivity and isobaric specific heat capacity, respectively.

\[
\delta_{L,ph} = \frac{4.6k}{c_p \rho u_L} \quad (8.18)
\]

Gas properties were weighted using the fuel and air mass fractions and calculated for the approximate conditions at the time of ignition. Values of \( \delta_L = 22–23 \mu\text{m} \) resulted, which fall within the range of those listed by Abraham et al. [5].

Finally, a characteristic time scale of the laminar flame can be approximated as:

\[
t_L = \frac{\delta_L}{u_L} \quad (8.19)
\]

The above equation can be interpreted as the time required for the flame front to move across a distance of one flame thickness. For the previously calculated values of \( \delta_L \) and \( u_L \), the laminar flame time is 39–42 \( \mu\text{s} \), depending on the equivalence ratio.

### 8.5.2 Turbulence Length Scales

Turbulent length scales can be measured directly using an LDV system, provided two simultaneous measurement volumes or an elongated measurement volume can be implemented [57]. Both of these techniques require hardware not available to the author, and therefore the length scales provided here are estimates based on general knowledge within the field of turbulence, rather than direct measurements.
The integral length scale, $l_o$, is known to be on the order of $1/6$ of the largest eddy size within the flow, which is governed by the physical boundaries of the system (e.g. combustion chamber walls). For flow into the cylinder near piston TDC, the limiting size will be the clearance height, $h_c$, and thus the integral length can be estimated from $h_c/6$ [94]. Depending on the crank angle position at the SOC, the integral length scale in the split-cycle engine is $l_o = 0.2$–$0.8\,\text{mm}$ over the period of $10$–$25^\circ\text{CA ATDC-e}$, respectively.

Energy transfer between the integral length scale and the smallest (Kolmogorov) scale is approximately linear and independent of the molecular viscosity. Therefore, the rate at which energy is transferred from the larger eddies to the smaller ones, known as the dissipation rate, $\epsilon$, can be found from:

$$\epsilon = \frac{(u'_o)^3}{l_o} \quad (8.20)$$

where $u'_o$ can be taken as the average turbulent velocity fluctuations, $u'$, at the time of ignition. Based on the flow experiment results in Sections 8.4.6 and 8.4.7, $u'_o$ is conservatively predicted to lie between $2\,\text{m/s}$ and $15\,\text{m/s}$. From the range of integral length scales previously calculated, the dissipation rate can be estimated to fall within $0.01$–$17.15\,\text{MW/kg}$, keeping in mind the mass in the cylinder is on the order of $0.2\,\text{g}$.

The Kolmogorov length scale, $l_k$, can now be estimated from Equation (8.21), using a value of $3.282\times10^{-5}\,\text{m}^2/\text{s}$ for the kinematic viscosity, $\nu$.

$$l_k = \left( \frac{\nu^3}{\epsilon} \right)^{1/4} \quad (8.21)$$

As a result, the smallest eddies in the flow field are in the range of $7$–$43\,\mu\text{m}$, primarily depending on the turbulence intensity.

### 8.5.3 Turbulence Time Scales

Similar to the length scales, time scales for the integral and Kolmogorov eddies can be found from Equations (8.22) and (8.23), respectively. The integral time scale, $t_o$, is interpreted as the average turnover time of an integral-sized eddy, based on its size ($l_o$) and velocity ($u'_o$):

$$t_o = \frac{l_o}{u'_o} \quad (8.22)$$
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The Kolmogorov time scale, \( t_k \), is based on dimensional analysis [91], and calculated from:

\[
t_k = \left( \frac{\nu}{\epsilon} \right)^{1/2}
\]  

(8.23)

The range of \( l_o \) and \( u_o' \) values result in integral and Kolmogorov time scale ranges of 13–393 \( \mu s \) and 1–57 \( \mu s \), respectively.

### 8.5.4 Combustion Regime

Based on the predicted laminar flame properties (Section 8.5.1), and the estimated turbulence characteristics (Sections 8.5.2 and 8.5.3), the combustion regime(s) within the engine can now be approximated. To do so, several non-dimensional parameters need to be defined. These are: the turbulent Reynolds number, \( Re_T \), the Damköhler number, \( Da \), and the Karlovitz number, \( Ka \); determined using Equations (8.24), (8.25), and (8.26), respectively.

The turbulent Reynolds number is based on the integral length scale eddies, \( l_o \), and the kinematic viscosity, \( \nu \), of the unburned gas:

\[
Re_T = \frac{u_o' l_o}{\nu}
\]  

(8.24)

The Damköhler number is a ratio of the macroscopic time scale to the characteristic chemical reaction time:

\[
Da = \frac{t_o}{t_L} = \left( \frac{l_o}{\delta_L} \right) \left( \frac{u_L}{u_o} \right)
\]  

(8.25)

The Karlovitz number is a relation of time scales similar to the Damköhler number, except it is between the integral time scale and the Kolmogorov time scale:

\[
Ka = \frac{t_L}{t_k} = \left( \frac{\delta_L}{u_L} \right) \left( \frac{\nu}{\epsilon} \right)^{-1/2} = \left( \frac{\delta_L}{l_k} \right)^2
\]  

(8.26)

For the aforementioned ranges of \( l_o \) and \( u_o' \), the values of \( Re_T \) were calculated to be in the range of 10–400; the \( Da \) number ranged from approximately 0.3–10; and the \( Ka \) number was determined to be between 0.7 and 28.

In combination, the non-dimensional parameters in Equations (8.24), (8.25), and (8.26) are used as boundaries for the various combustion regimes. A commonly used plot for displaying these regimes is the Borghi diagram [21], which relates the turbulence scale \( (l_o/\delta_L) \) to its intensity \( (u_o'/u_L) \). Slightly different variations of the same plot can be found throughout the literature [4, 21, 91, 113, 117, 145]. However, the author elected to use a hybrid terminology.
from Peters [113] and Law [91]. The corresponding combustion regime diagram, including the estimated region of operation for the split-cycle engine, is shown in Figure 8.20.

![Combustion phase diagram](image)

**Figure 8.20:** Combustion phase diagram adapted from Borghi [21], showing estimated regime of split-cycle engine.

The greyed rectangle in Figure 8.20 shows the split-cycle combustion regime for the range of $u'_o$ and $l_o$ estimated to be present within the engine. Variations in equivalence ratio, from $\phi = 0.8$ to $\phi = 1$, are also included, but the difference can be considered negligible relative to the overall regime uncertainty. The distribution in the x-axis (rectangle width) accounts for the change in clearance height, and consequently $l_o$, over the range of ignition timing from 10–25 °CA ATDC. Similarly, the vertical distribution encompasses the range of velocity fluctuations, from 2–15 m/s. Based on the predictions of rapid turbulence decay (see Figure 8.19), it can be expected that later spark timing will correspond to the lower right-hand corner of the regime, and earlier spark timing will correspond to the upper left-hand corner of the regime. The reader may notice that for lines of constant $Re_T$, for which only $Re_T = 1$ is drawn, the values previously listed are slightly lower than those shown in Figure 8.20. This is due to the assumptions made in the calculation of $\delta_L$ and $u_L$.

Regardless of the exact conditions, it is clearly shown that the combustion regime is predicted to lie primarily within the distributed reaction zone, bound by the lower and upper lines of $Ka = 1$ and $Da = 1$, respectively. The lower bound means $Ka > 1$ and implies
that the Kolmogorov eddy turnover time is faster than the characteristic chemical time. According to Equation (8.26), this also indicates that the smallest eddies are of sufficiently small size \( l_k < \delta_L \) to be able to penetrate into the preheat zone of the flame. As a result, the rates of mass and heat transfer from the preheat zone increase, thereby extending its thickness. At this stage, however, the inner layer of the reaction zone, \( \delta_R \), is still smaller than the Kolmogorov eddies, and therefore impenetrable. Consequently, the flame still behaves as a broadened flamelet with respect to the larger eddies in the flow.

Typical naturally aspirated SI engines operate within the corrugated flamelets regime, with some extension into the distributed reactions regime \([5, 70]\); the transition above \( Ka = 1 \) is aided through the use of turbocharging \([93]\). In either case, larger clearance heights (in comparison to the split-cycle engine) are mandated by the compression ratio and result in bigger integral length scales. As a consequence, the operating region for the average SI engine on the combustion phase diagram (Figure 8.20) is located slightly down and to the right of the split-cycle region shown. Because the compression ratio has no direct coupling to the expansion cylinder, the split-cycle engine can utilize a much smaller clearance height. It can therefore be stated that the split-cycle engine in this work possesses a smaller scale, and a similar or higher intensity, of turbulence than a conventional SI engine.

The line of \( Da = 1 \) implies the turnover time of the integral eddies is on par with the characteristic chemical reaction time. In other words, the time needed for flow induced change on a large scale is shorter than that needed for chemical change. In theory then, the flame becomes further broadened to the point where reaction sheets no longer exist. It should be explicitly stated that the portion of the split-cycle regime shown to exceed the \( Da = 1 \) line in Figure 8.20 could only be achieved in actual practice for the earliest spark timings (i.e. 16–18°CA ATDC-e), and only if \( u'_o \) was able to exceed \( \sim 12 \text{ m/s} \).

In more current publications \([91, 114]\), the dividing region between the reaction sheet regime and a perfectly stirred reactor has been further refined to include a second Karlovitz number based on the inner layer thickness of the reaction zone, \( \delta_R \):

\[
Ka_R = \frac{\delta_R^2}{l_k^2} \approx \Psi^2 Ka
\]

Equation (8.27) relates the size of the smallest eddies to the thinnest part of the reaction zone, where fuel consumption occurs. The constant \( \Psi \) represents the ratio of laminar flame thickness to the inner reaction layer thickness (i.e. \( \delta_L/\delta_R \)), which, recall from Section 8.5.1, is \( \mathcal{O}(10) \). Thus, when \( Ka_R = 1 \) the Kolmogorov eddies are of sufficiently small size to penetrate
into the inner reaction layer of the flame. From Figure 8.20 it can be seen that the split-cycle engine regime never crosses the $K\alpha_R = 1$ boundary. The general consensus amongst the literary sources cited in this section is that combustion is unlikely to be sustained for $K\alpha_R \geq 1$, due to high rate of heat diffusion from the reaction layer, leading to a drop in temperature, and subsequent extinction of the flame.

Bulk flame quenching, however, was implicitly observed in Chapter 6, with the presence of partial burning and misfiring cycles. It is no surprise that Figure 8.20 does not capture this phenomena, as the combustion regime metrics rely solely on a comparison of length and time scales that do not account for additional aspects such as thermal-diffusive flame instabilities and flame stretch. The latter can be evaluated using the Karlovitz strain or stretch factor, $K$. Abdel-Gayed and Bradley [2] developed the following expression for $K$ in terms of the turbulent velocity fluctuations, the laminar flame speed, and $Re_T$:

$$K = 0.157 \left( \frac{u_o'}{u_L} \right)^2 Re_T^{-1/2}$$  \hspace{1cm} (8.28)

Further work by the same authors [1, 3, 4] has shown that quenching in turbulent premixed flames can be predicted using the product of the Karlovitz stretch factor, $K$, and the Lewis number, $Le$. The latter is the ratio of thermal diffusivity, $\alpha$, to mass diffusivity, $D$, of the deficient reactant from within the reaction zone:

$$Le = \frac{\alpha}{D} = \frac{k}{\rho c_p D}$$  \hspace{1cm} (8.29)

where $k$ is the thermal conductivity, $\rho$ is the density, and $c_p$ is the specific heat capacity. For values of $KLe \geq 6$, turbulent flame quenching has been observed by the aforementioned studies. This value should not be taken as a hard limit, but a mere approximation where bulk quenching could be significant.

For the methane-air mixtures used in the split-cycle engine, $Le \approx 1$, and $K$ varies from approximately 0.3 to 8.2 over the range of $u_o' = 2$–15 m/s, $\phi = 0.8$–1.0, and $\theta_{ign} = 16$–25°CA ATDC-e. The change in $KLe$ with $u_o'/u_L$ is shown in Figure 8.21. The two curves represent the upper and lower limits of ignition timing that were investigated, accounting for the change in integral length scale, and its subsequent effect on $Re_T$ at the SOC. Based on this simplified analysis, and a conservative estimate of the turbulence intensity within the engine’s combustion chamber, it is shown that the upper range of the velocity fluctuations are theoretically sufficient to cause flame quenching by means of strain. This is not definitive, however, as the $KLe$ limit is a function of the turbulent burning velocity, which has been completely neglected here. Under lean conditions, the reduction in turbulent burning
velocity is expected to be significant, and therefore lean flames will be less tolerant of high strain rates. This is the most probable reason why the empirically observed lean operating limit is considerably higher than the lean flammability limit of methane-air. Figure 8.21 does not capture these effects, and simply implies that the in-cylinder conditions are at least approaching those found to extinguish flames through stretch. An accurate measure of the velocity fluctuations in the actual engine would considerably help refine these estimates.

As a final note, the reader should keep in mind that the combustion regime and other parameters presented in this section are rough estimates only. The characteristic measure of turbulence, $u'_o$, is based on a single-point measurement, in a non-stationary flow, of a marginally replicated engine environment. Besides the deficiencies of the experimental apparatus already discussed in Section 8.4.8, the flow field turbulence is not anticipated to be either isotropic or homogeneous, meaning it is likely variant in both direction and location, respectively. It follows that the parameters calculated in Section 8.5 should not be viewed as constants, but averages representing statistical distributions.

### 8.6 Summary

The purpose of this chapter was to provide some quantifying measurements relevant to the predicted turbulence properties present in the combustion chamber of the split-cycle engine. An experimental apparatus was constructed to replicate the engine conditions, and evaluate
the downstream flow field of pressurized air discharging through a reverse poppet valve. The results from three different experiments were presented: visualization of the flow, velocity measurements of the flow impinging on an unconfined plate, and velocity measurements of the flow during pressurization of a confined volume.

High speed imaging of the flow as it emerged past the valve revealed an annulus of fluid converging upon itself and creating a solid core jet. The formation of the jet required a minimum downstream distance of 5 mm, and impingement on the piston would therefore occur prior to the intersection of the annular flow in the actual engine. Coherent structures were observed within the flow, but these would decrease significantly in size in the actual engine, if they were to occur at all.

Velocity measurements taken using LDV techniques show instantaneous peaks up to 80 m/s during the period of choked flow. For the unconfined, flat plate case, mean velocities in both the axial and radial directions were found to diminish quickly as the measurement location was moved radially outward from the valve, but the velocity fluctuations remained high, averaging around 16 m/s and 13 m/s for the axial and radial components, respectively.

When the downstream flow field was confined by a volume, imitating the engine’s combustion chamber with an 8 mm clearance height, a similar magnitude of peak velocity was observed. However, these high mean velocities, and corresponding velocity fluctuations, were short-lived; lasting less than 10 ms, and followed by a rapid reduction in turbulence intensity. This highlights the importance of the secondary flow that is achieved in the actual engine by having offset cylinder phasing, which allows the compression cylinder to maintain a small amount of flow into the combustion chamber, after the initial discharge period occurs. Depending on the ignition timing, the RMS turbulent velocity fluctuations were conservatively estimated to fall within the range of 2 m/s to 15 m/s at the SOC.

These velocity measurements, along with several other estimated parameters, were then used to predict the combustion regime of the split-cycle engine. Due to the coexisting small scale and high intensity turbulence that is present, combustion is expected to fall primarily within the “reaction sheets with distributed reaction zones” regime. While conventional SI engines can also enter into this regime, the smaller integral length scales of the split-cycle engine enables the regime boundary to be crossed at a lower level of turbulent intensity. A simplified evaluation of the Karlovitz stretch factor for laminar flames indicated that flame quenching by method of strain was a definite possibility for the given flow field conditions. This is believed to be the main cause of the relatively high lean operating limit that was discovered in Chapter 6.
Chapter 9

Summary, Conclusions, and Recommendations

9.1 Summary

The objective of this research was to investigate the feasibility of using split-cycle engine architecture to overcome the three major deficiencies that currently plague natural gas powered, spark ignition engines. In no particular order, these are:

1. Poor volumetric efficiency (port-injected engines)
2. Insufficient air/fuel mixing (direct-injected engines)
3. Reduced specific output caused by slow combustion rates

It was hypothesized that the basic operating principle of the split-cycle engine would generate a sufficiently intense level of in-cylinder turbulence, immediately prior to the time of ignition, to significantly improve the rate of combustion. The split-cycle configuration also provided a means of implementing post-compression fuel injection, without locating the fuel injector inside the combustion chamber. This was expected to improve the volumetric efficiency by eliminating intake air displacement caused by port-injection. Furthermore, the author proposed the use of a novel injector location and timing strategy so that considerable air/fuel mixing could be performed upstream of the combustion chamber, in the intermediary (crossover) passage connecting the two engine cylinders. Maintaining the low exhaust gas emission characteristics that are inherent to CNG engines was a critical secondary requirement of the research.
To assess the aforementioned hypotheses, a two-cylinder split-cycle engine was designed, constructed, and tested by the author. At the time of writing, and to the author’s best knowledge, this is the only functioning split-cycle engine that has been developed within an academic institution and the only published work of a split-cycle engine operating on methane [30]. An engine-dynamometer test cell was commissioned to provide the means necessary for evaluating the engine’s performance and exhaust gas emissions. A 1-D numeric engine model was used to configure the initial valve timing of the engine, and to investigate how valve timing and crossover passage sizing affect performance.

The basic operating principles of the split-cycle engine required the development of a fast-acting, reverse-poppet valvetrain, which proved to be a considerable engineering challenge. The short valve durations led to the creation of a novel MATLAB® program that could produce an optimized cam lobe profile based on user-imposed dynamic constraints. The overall valvetrain design proved to be successful, with the exception of the valve closure force, which needs to be increased so that higher peak cylinder pressures can be achieved. This was a limiting factor for this research and will be addressed in the future.

Volumetric efficiency of the split-cycle engine was found to range from 71–75%, varying with changes in crossover passage pressure. Compared to the Kubota Z482 diesel engine platform on which the split-cycle engine was based (i.e. same bore, stroke, and intake valve diameter), the volumetric efficiency of the split-cycle engine is lower by approximately 10–15%. The deficiency is a result of the engine’s inability to transfer 100% of the mass from the compression cylinder into the crossover passage. Re-expansion of this residual gas effectively truncates the subsequent intake stroke.

Mixing between the air and the gaseous methane fuel was determined to be very good based on the following observations:

1. The engine was able to operate with relatively good stability (COV$_{IMEP}$ < 6.6%) when fuel was only injected once every three engine cycles. This indicates that significant mixing is occurring in the crossover passage, even when initial conditions are heavily stratified.

2. Under normal fuelling conditions (i.e. one injection per cycle), COV$_{IMEP}$ < 3% was achieved for all engine speeds tested. This level is considered normal for a typical SI engine [131].

3. Combustion efficiencies, based on the exhaust gas composition, were in excess of 96%, indicating most of the fuel was consumed during combustion. Typical SI engine combustion efficiencies range from 95–98% for similar air/fuel ratios to those tested. [70].
Chapter 9: Summary, Conclusions, and Recommendations

Item 1 also indicates that significant air-fuel mixing must occur within the crossover passage, and does not rely solely on the turbulent transfer period into the combustion chamber. This is believed to be a consequence of injecting fuel approximately 270°CA before the crossover outlet valve was opened, providing significantly more mixing time than a conventional port- or direct-injected engine.

Fast combustion rates were realized by the split-cycle engine, with the shortest average CA0–90 burn rates on the order of 23–24°CA, occurring for the earliest spark timings and the richest air/fuel ratio (ϕ = 1) tested. Additional spark advance will be possible once the valve retention issue is resolved, leading to the potential for even quicker burn rates; although the findings in this work indicate a possible “levelling off” in CA0–90 as ignition timing moves closer to TDC. The quickest durations are on par with turbocharged SI NG engines, and up to four times faster than naturally aspirated SI NG engines.

In a separate experimental apparatus, off-line from the engine, the flow field characteristics generated down-stream from a reverse poppet valve were investigated. Velocity measurements were used to infer the turbulent conditions present within the combustion chamber at the time of ignition. Based on these findings, the fast burn durations are supported by predictions that combustion falls within the distributed reactions regime. There is also evidence that the turbulence intensity may be sufficient to cause significant flame quenching, which may explain the relatively high lean operating limit discovered in Chapter 6.

Despite the rapid combustion, specific power output (IMEP) and fuel conversion efficiency were both modest. The former was constrained by the peak combustion pressure restriction imposed by the crossover outlet valve. This limited advances in combustion phasing, which lead to a higher-than-necessary exhaust gas enthalpy, and reduced efficiency. Average indicated fuel conversion efficiencies ranged from 22–27 %, which is relatively low compared to the current state of the art. Heat transfer from the crossover passage, late combustion phasing due to the spark timing limitation, and higher than normal blow-by losses are the major sources of inefficiency compared with conventional engines.

Low levels of harmful exhaust gas emissions, characteristic of SI NG engines, were obtained or improved in this work. The exhaust gas emissions are summarized as follows:

**NOx**: 150–800 ppm. Levels increased with equivalence ratio and engine load. The latter was limited in this work, and thus NOx levels are expected to increase once higher peak cylinder pressures are achieved. The current levels are considered low-to-average, a result of the inherently late split-cycle combustion phasing and limited engine load.
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**CO:** 1500–2000 ppm at $\phi \approx 1$, 500–700 ppm at $\phi < 0.97$. Approximately 50% less than a conventional SI engine, attributed to a well-mixed air/fuel charge.

**THC:** 1500–3000 ppm. Levels increase with spark timing advances due to a reduction in EGT, affecting post-combustion oxidation. A rise in HC emissions on approach to the lean limit is caused by incomplete combustion. Approximately 85–95% of THC emissions are NMHCs, implying they are unburned fuel from crevice volumes. The levels are considered average for an SI engine.

### 9.2 Conclusions

1. Operation of a methane-fuelled split-cycle engine, utilizing fuel injection in the intermediary (crossover) passage between cylinders, is feasible. Moreover, the development and construction of such an engine is possible under the relatively constrained budget of an academic research environment.

2. The split-cycle engine architecture used in this work is capable of generating small-scale, high-intensity turbulence within the combustion chamber at the time of ignition. As a result, very rapid combustion can be achieved, even for unfavourable burning conditions (e.g. late combustion phasing, low compression ratio), and without the use of squish volumes or organized cylinder motion.

3. Injection of gaseous fuel into the crossover passage is a viable technique for generating a homogeneous air/fuel mixture without disrupting intake air flow, or requiring the use of a fuel injector that can withstand the high temperatures of combustion. In this case, the fuel injector has been placed opposite a solid wall, which is believed to help break apart the gaseous fuel jet. Early injection also provided a longer mixing time.

4. The size of the crossover passage effectively determines the compression ratio of the split-cycle engine. The volume should be made as small as practicable to minimize heat transfer losses, and increase the compression ratio (within the limits of auto-ignition). However, significant air/fuel mixing also occurs within the crossover passage, and it is not yet known how the crossover volume affects mixture homogeneity.

5. The split-cycle engine has an inherent issue of not fully displacing all of the air out of the compression cylinder at the end of the compression stroke, resulting in a reduced volumetric efficiency. Since the density of this residual air is very high, approximately 20 times atmospheric conditions, minimizing the crevice volume(s) in the compression cylinder should be a priority for this type of engine.
6. Offset piston phasing is important for two reasons: i) it is believed to sustain the in-cylinder turbulence intensity after the initial, high-velocity filling period, and ii) it allows the expansion cylinder to minimize its volume at TDC for expulsion of residuals, and then isobarically increase in volume to adequately support combustion.

7. Creating an air/fuel mixture at elevated pressure and temperature, external to the dedicated combustion chamber, is not without risk. Backfiring into the crossover passage can be initiated under two different circumstances: i) through overly advanced spark timing, and ii) by means of autoignition. The latter was caused by very late burning cycles under lean conditions; however, the exact source of ignition is unknown.

9.3 Summary of Contributions

The following list contains the major contributions that this work has provided to the field of engineering science:

1. The author developed a unique split-cycle research engine and associated test platform (see Chapters 3 & 4). The engine features a modular reverse poppet valvetrain, a high-pressure gaseous fuel injection system, mechanically variable camshaft phasing, and an adjustable crankshaft journal offset. To the extent of the author’s knowledge, this is the only operational split-cycle engine of its type within an academic institution, and serves as an excellent foundation for future engine and combustion research.

2. The author proposed the use of a split-cycle engine as a means to overcome the major performance deficiencies of natural gas engines. As part of this effort, a novel fuel injection strategy was also employed. The experiments carried out in this work have shown that the combination of these two aspects can successfully achieve good mixture homogeneity, low emissions, and very fast combustion rates.

3. The author conducted an extensive experimental testing regime on the split-cycle engine presented in this work. The results reported from these tests reveal important operational, performance, and emissions characteristics regarding the split-cycle concept, and serve to expand the limited amount of information presently available in the literature. Furthermore, the author is not aware of any other published works demonstrating empirical results from a methane-fuelled split-cycle engine.

4. The author generated and validated a parametric 1-D split-cycle engine model using commercial software (see Chapter 7). The affect of valve timing and crossover passage volume on engine performance has been shown.
Chapter 9: Summary, Conclusions, and Recommendations

9.4 Recommendations and Future Work

9.4.1 Engine Hardware

This research has highlighted some of the deficiencies in the current split-cycle engine design. The following list details the improvements that can be made:

1. Currently, engine load is limited by the ability to maintain crossover outlet valve closure at high peak cylinder pressures. As a consequence, MBT spark timing was never achieved in this work, meaning further gains in specific output and thermal efficiency are possible. To maintain valve closure, the retention force needs to be increased. The use of a stiffer valve spring is not recommended, since the increased load on the valvetrain during the lift period is unnecessary, and may exceed the strength of one or more components. Implementation of a pneumatic spring would likely be the least invasive, only requiring replacement or modification to the valve spring bridge (see Section 3.5.8). Proper sizing of the pneumatic piston means a constant force throughout the lift period could closely be achieved.

2. The energy balance performed in this work indicates that improvements can be made regarding heat and mass losses. The added surface area of the crossover passage, and the high flow velocity into the expansion cylinder both increase the heat transfer losses. Therefore, it is recommended that a thermal barrier coating be applied to the crossover passage and combustion chamber surfaces. Additionally, mass leakage through the seals of the crossover valve guides may be contributing to the decrease in crossover pressure throughout the cycle. It is recommended that this leakage be quantified and, if required, new valve guide seals should be designed for the magnitude of pressure present in the crossover passage.

3. The numerical simulations performed in this work shows the opening duration of both crossover valves to be approximately 10°CA too long for the engine speeds investigated. This is particularly important for the crossover outlet valve; by minimizing its duration, the SOC can occur closer to the main discharge period, and closer to effective TDC. It is recommended that new cam lobes be manufactured accordingly.

4. The actual gas compression ratio of the engine was calculated to be relatively low (∼8.5:1), and should be increased to take advantage of methane’s high octane number. Simulations performed in this work shows that decreasing the crossover passage volume is one method of doing so, but will be limited in this case by practical implementation. Another means of increasing the compression ratio would be through
turbocharging, which would also capitalize on the high exhaust gas enthalpy caused by late combustion phasing.

5. Flow rate measurements have indicated that considerable blow-by is occurring in the engine, presumably within the compression cylinder (see Section 6.4.3). For maximum efficiency, the exact source of this leak should be determined and rectified.

9.4.2 Numerical Model

Model Improvements

The numerical engine model developed in AVL BOOST could use several improvements for better accuracy, and broader versatility. These are as follows:

1. Inclusion of a combustion model that accounts for changes in reaction kinetics with different unburned gas temperatures (e.g. CHEMKIN software). Further refinement could be made by coupling the model with a commercially available engine CFD code, such as AVL FIRE, which would enable turbulence effects to be predicted. The latter, however, requires considerably more computational effort.

2. Since the model was only used to predict trends, absolute values of heat transfer and mass flow rates were of no particular concern in this work. For improved accuracy, fine tuning of the heat losses within the crossover passage and ports should be performed. The valve discharge coefficients are based on flow bench measurements at a single pressure ratio. Further refinement can be made by repeating the measurements for a range of pressure ratios that are relevant to those encountered in the engine.

Future Case Studies

It is recommended that the current engine model be utilized to investigate, and possibly optimize, various design parameters. These include, but are not limited to:

1. Differing displaced volumes between the compression and expansion cylinders
2. Differing coolant temperatures for the compression and expansion cylinders
3. Magnitude of piston offset phasing
4. Regenerative heat transfer between the exhaust gas and crossover passage
9.4.3 Turbulence Measurements

Many assumptions were made regarding the validity of the velocity measurements that were used to infer turbulence and combustion characteristics (see Section 8.4.8). Because of the differences between the actual engine and the experimental apparatus used to perform these measurements, it is recommended that further investigations be performed within an actual engine environment. By doing so, the fluid properties will be correct, and dynamic effects of the piston and valve(s) will be included.
Bibliography


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ignition (SI) and compression-ignition (CI) engine performance and emissions. 


[90] G.A. Lavoie, J.B. Heywood, and J.C. Keck. Experimental and theoretical study 
of nitric oxide formation in internal combustion engines. _Combustion Science and 


[92] Y. Li, S. Liu, S.-X. Shi, M. Feng, and X. Sui. An investigation of in-cylinder tum- 
bling motion in a four-valve spark ignition engine. _Proceedings of the Institution of 
Mechanical Engineers, Part D: Journal of Automobile Engineering_, 215(2):273–284, 

[93] D. Linse, C. Hasse, and B. Durst. An experimental and numerical investigation of 
turbulent flame propagation and flame structure in a turbo-charged direct injection 

1999.

[95] F. Ma, Y. Wang, H. Liu, Y. Li, J. Wang, and S. Zhao. Experimental study on thermal 
efficiency and emission characteristics of a lean burn hydrogen enriched natural gas 

data for combustion diagnostics of HCCI engine. _Mechanical Systems and Signal 


2011.


Appendix A

CAD Drawings

The CAD drawings in this section were generated by the author to facilitate the manufacturing of numerous components for the split-cycle engine and engine test stand. The drawings have been scaled to fit the page size of this document and the indicated scale in each title block is therefore no longer valid.
Cranktrain

Crankshaft Pin
OHC Valvetrain

Valve Spring Preload Adjuster

Sheet 1 of 1

Radius of corners should allow fitment of 10mm hex drive.

M26 x 1.5 - 4h6h

Front view

Right view

Isometric view

Section view A-A

<table>
<thead>
<tr>
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<th>Spring Preload Adjuster</th>
<th>DESIGNED BY:</th>
<th>Iain Cameron</th>
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<td></td>
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<tr>
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UNLESS: X +/- 0.02
OTHERWISE: XX +/- 0.01 in.
SPECIFIED: XXX +/- 0.002
Valve Spring Preload Adjuster Jam Nut

Sheet 1 of 1

Isometric view

Front view

Right view

Section view A-A

Light Chamfer

**PART NAME:** Spring Preload Jam Nut  **DESIGNED BY:** Iain Cameron

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<td></td>
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</table>
Valve Spring Preload Adjustment Bridge (1/2)
RPV Camshaft (Shaft Only)
RPV Camshaft Lobe
Appendix A: CAD Drawings

Camshaft Bearing Cap

Notes:

Final Bore Size:
41.0007 mm MIN
41.0261 mm MAX

Bearing Clearance:
0.0006" MIN
0.0028" MAX

<table>
<thead>
<tr>
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<th>DESIGNED BY: Iain Cameron</th>
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Appendix A: CAD Drawings

OHC Front Retainer and Seal

Sheet 1 of 1

Counterbore: .4375" dia. x 4mm deep

Hole Coordinates

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</tr>
<tr>
<td>C</td>
<td>28</td>
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<td>D</td>
<td>56</td>
<td>45</td>
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<tr>
<td>E</td>
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Front View

\( \phi \ 30 \pm 0.2 \) (through-hole)
\( \phi \ 38.1 \pm 0.1 \)
1.50 (counterbore)

Isometric View

Section view A-A

<table>
<thead>
<tr>
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<th>Iain Cameron</th>
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</table>
Appendix A: CAD Drawings

RPV Lift Tappet

Sheet 1 of 1

M6x0.75 THRU
Minimum 75% thread engagement

Break sharp edges

Top View

M10 HEX

Side View

18
15°
6
9

Isometric View

PART NAME: RPV Tappet
DESIGNED BY: Iain Cameron

SCALE: 4:1 MATERIAL: Ti-6Al-4V Titanium UNLESS:
UNLESS:
OTHERWISE:
SPECFIED:
UNLESS: X
IN:
PART NO: N/A FINISH: As machined.
MFD BY: TSC

COMMENTS: Use dedicated titanium tap and Moly-Dee cutting fluid (provided).
Reverse Poppet Valve

Front view

Sheet 1 of 1

<table>
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<th>Reverse Poppet Valve</th>
<th>DESIGNED BY:</th>
<th>Iain Cameron</th>
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<tr>
<td>PART NO:</td>
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<td>SPECIFIED:</td>
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<tr>
<td>COMMENTS:</td>
<td>Threads to be ground after valves are received from Ferea.</td>
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Detail A
Scale: 4:1

M6x0.75 - 4h6h

(Approximate) 3.75

45° × 0.5

84

25

112

0.2

6

45°

19.5

25

0.02

A
RPV Spring Retainer

Sheet 1 of 1

Ensure radius is small enough that spring sits flat on adjacent surface.

Top View

Side View

Section View A-A

Isometric View

<table>
<thead>
<tr>
<th>PART NAME:</th>
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<th>Iain Cameron</th>
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<td>COMMENTS:</td>
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Pushrod Valvetrain

Camshaft Extension

Notes:
1) Internal bore can either be made from existing gear and welded to shaft or directly from billet.

2) Internal bore diameter needs to be a tight press fit onto camshaft. Use existing gear for exact measurement.

3) External dimensions must be machined after extension has been pressed onto camshaft. Total runout should be less than 0.03 mm.

---

**PART NAME:** Z482 Camshaft Extension - Rev B

**DESIGNED BY:** Iain Cameron

**SCALE:** 1:1  **MATERIAL:** 4140 Steel  **UNITS:** mm  
**DATE:** 07/05/2013  **UNLESS:** X +/- 0.02  
**PART NO:** 0.1.1  **FINISH:** None  **OTHERWISE:** .XX +/- 0.01 in.  
**MFD BY:** Andy Jenner  **SPECIFIED:** .XXX +/- 0.002  
**COMMENTS:** See notes above.
Camshaft Extension Seal Housing

Bore for 0.75" shaft seal
SKF #: 7512
McMaster-Carr: 5154T16
0.001-0.002" press-fit

Sheet 1 of 1

PART NAME: Z482 Camshaft Extension Seal Housing
DESIGNED BY: Iain Cameron

SCALE: 1:1 MATERIAL: 6061 Aluminum UNITS: mm DATE: 20/05/2013
UNLESS OTHERWISE SPECIFIED: X in.
PART NO: 0.1.1 FINISH: None MFD BY: Iain Cameron
COMMENTS: Use silicone around seal perimeter before installing.
Front Cover Modification for Camshaft Extension

ORDER OF MACHINING OPERATIONS:
1. Machine part with excess material on inner bore dimension (+1-2mm).
2. Set up engine block on mill table and indicate camshaft bore to (0,0).
3. Attach front cover to block and drill hole (2.90mm) at (0,0) location.
4. Remove front cover and weld part approximately concentric with hole.
5. Reattach front cover to block and machine inner bearing bore to size.

<table>
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<td>FINISH:</td>
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Camshaft Rear Retainer and Seal

Sheet 1 of 1

Tangent to both circles

Front view
Scale: 1:1

Chamfer to fit
M8 Flat-Head SCS

M8 clearance hole

Section view A-A
Scale: 1:1

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Rocker Shaft Modification

Notes:

1) The bolt flats were designed to be the width of a 0.5" end mill. Width tolerance is not critical.

2) Both ends of the shaft need to be sealed without impeding the through-bolt holes. Welding should be done with minimal heat input to avoid distortion of the shaft.

3) There is an existing hole in the shaft (approx. 1.4mm dia., no chamfer and not shown in the drawing), this needs to be on the same side as the wrench flats and sealed by means of welding.
Timing Belt

Idler/Tensioner Bracket
Idler/Tensioner Arm
Idler/Tensioner Pulley
Eccentric Idler Pulley Mount

TOP VIEW

Sheet 1 of 1

SIDE VIEW

FRONT VIEW

ISOMETRIC VIEW

PART NAME: Timing Belt Eccentric Pulley Mount

DESIGNED BY: Iain Cameron

SCALE: 1:1  MATERIAL: Ductile Cast Iron  UNLESS: X

UNLESS: X  OTHERWISE: JXX

UNITS: mm  SPECIFIED: JXX

FINISH: As machined.

MFD BY: L Cameron

DATE: 02/09/2014

IN.

PART NO: N/A

COMMENTS: Replaces existing breather vent plate.
Appendix A: CAD Drawings

AC Dyno Drive

Mounting Plate for Dynamometer Components
Dynamometer Torque Sensor Mount
Pillow Block Bearing Support for Dynamometer Shaft (1/2)
Pillow Block Bearing Support for Dynamometer Shaft (2/2)

Sheet 2 of 2

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<td>MFD BY:</td>
<td>Dave Tremblay</td>
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<td>COMMENTS:</td>
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Dynamometer AC Motor Shaft Coupler
Eccentric Bearing Shaft
Belt Roller

Sheet 1 of 1

Front View

Side View

Section View A-A

<table>
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2.900

2.625

Ø 2.625

1.875 Ø 2.3124

1.030

1.050

SHARP CORNER

CHAMFER EDGES

BEARING BORE
PTO Shaft Support Assembly (1/2)
Appendix A: CAD Drawings

PTO Shaft Support Assembly (2/2)

Order of Machine Operations:
1) Wire EDM mounting flange (A) and cut tubular structure (B).
2) Machine outer dimensions of bearing support (C). DO NOT machine inner bore.
3) Tack entire assembly together and then finish weld.
4) Indicate bore of mounting flange (A) and finish machine inner bore of bearing support (C).

Engine PTO Bearing Support

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<th>UNIT</th>
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<td>in.</td>
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Sheet 2 of 2

Isometric View
Scale: 11:18
Engine - General

Valve Cover
Engine Mount
Engine Flywheel

\[ m = 7.41 \text{ kg} \]
\[ I_x = 38724 \text{ kg-mm}^2 \]

---

**Engine Flywheel**

**Appendix A: CAD Drawings**

**Part Name:** Split-Cycle Flywheel

**Designed By:** Iain Cameron

**Scale:** 1:2

**Material:** 01 Tool Steel

**Units:** mm

**Date:** 28/08/2014

**Finish:** None

**MFD By:** Andy Jenner

**Comments:** To be balanced before final installation.
Appendix A: CAD Drawings

Encoder Mount Faceplate

Sheet 1 of 1

Front View

Bottom View

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UNLESS: .X

OTHERWISE: .XX

in.
Encoder Mount Side Bracket

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<th>Iain Cameron</th>
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<td>DATE:</td>
<td>25/01/2012</td>
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<td>MATERIAL:</td>
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<td>UNLESS:</td>
<td>X</td>
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<tr>
<td>UNITS:</td>
<td>in</td>
<td>OTHERWISE:</td>
<td>XX</td>
</tr>
<tr>
<td>PART NO:</td>
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<td>SPECIFIED:</td>
<td>XXX</td>
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<td>FINISH:</td>
<td>None.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>MFD BY:</td>
<td>Rezmin - Peter Guba</td>
<td></td>
<td></td>
</tr>
<tr>
<td>COMMENTS:</td>
<td>Left and right brackets are mirror images of each other.</td>
<td></td>
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Sheet 1 of 1
Throttle Linear Actuator Mount
Throttle Body Components
Throttle Body Housing (1/2)
Throttle Body Housing (2/2)
Throttle Body Mounting Flange
Intake Pipe for Throttle Body
Appendix B

Assembly Schematics and Bill of Materials

Figure B.1: Assembly drawing for intake air throttle body; refer to Table B.1 for details.
Figure B.2: Assembly drawing for intake air throttle body; refer to Table B.2 for details.
Figure B.3: Schematic of air intake system for engine; refer to Table B.3 for details.
Figure B.4: Assembly drawing of select valvetrain components; refer to Table B.4 for details.
Figure B.5: Assembly drawing of crank angle encoder components; refer to Table B.5 for details.
Figure B.6: Assembly drawing of engine PTO shaft and support components; refer to Table B.6 for details.
Figure B.7: Assembly drawing of engine-dynamometer belt tensioner; refer to Table B.7 for details.
### Table B.1: BOM for throttle body; referenced to Figure B.1.

<table>
<thead>
<tr>
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<th>Qty.</th>
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<th>Vendor</th>
<th>Description</th>
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<tr>
<td>1</td>
<td>2</td>
<td>92855A516</td>
<td>MMC</td>
<td>M5x0.8x16mm SS, low profile SHCS</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>94768A103</td>
<td>MMC</td>
<td>M5 washer, SS, 3.5 mm thick</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>94860523002</td>
<td>Siemens</td>
<td>Fuel injector, solenoid type, GDI</td>
</tr>
<tr>
<td>4</td>
<td>2</td>
<td>90965A160</td>
<td>MMC</td>
<td>M5 washer, SS, 1.0 mm thick</td>
</tr>
<tr>
<td>5</td>
<td>2</td>
<td>93655A879</td>
<td>MMC</td>
<td>M5x0.8 threaded hex stand-off, SS, 30 mm long</td>
</tr>
<tr>
<td>6</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Aluminum spacer</td>
</tr>
</tbody>
</table>

MMC = McMaster-Carr  
GDI = Gasoline direct injection

### Table B.2: BOM for throttle body; referenced to Figure B.2.

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<tr>
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<tbody>
<tr>
<td>1</td>
<td>4</td>
<td>91251A349</td>
<td>MMC</td>
<td>10-32 UNF x 1-1/4” SHCS</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>92220A174</td>
<td>MMC</td>
<td>10-32 UNF x 5/8” low profile SHCS</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>92185A198</td>
<td>MMC</td>
<td>8-32 UNC x 7/8” SHCS</td>
</tr>
<tr>
<td>4</td>
<td>1</td>
<td>91251A201</td>
<td>MMC</td>
<td>8-32 UNC x 1-1/4” SHCS</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
<td>91831A009</td>
<td>MMC</td>
<td>8-32 UNC locknut, 18-8 SS</td>
</tr>
<tr>
<td>6</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Aluminum spacer</td>
</tr>
<tr>
<td>7</td>
<td>2</td>
<td>Custom</td>
<td>N/A</td>
<td>Aluminum spacer</td>
</tr>
<tr>
<td>8</td>
<td>2</td>
<td>92671A195</td>
<td>MMC</td>
<td>10-32 UNF brass nut</td>
</tr>
<tr>
<td>9</td>
<td>1</td>
<td>L12-30-I-EXT</td>
<td>Firgelli</td>
<td>0-5VDC linear actuator</td>
</tr>
<tr>
<td>10</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Actuator arm, aluminum</td>
</tr>
<tr>
<td>11</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Butterfly mounting shaft, stainless steel</td>
</tr>
<tr>
<td>12</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Butterfly valve, brass</td>
</tr>
<tr>
<td>13</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Throttle shaft seal retainer (internal o-ring)</td>
</tr>
<tr>
<td>14</td>
<td>2</td>
<td>Custom</td>
<td>N/A</td>
<td>Bronze bushing</td>
</tr>
<tr>
<td>15</td>
<td>2</td>
<td>97184A260</td>
<td>MMC</td>
<td>M8x24mm through-dowels</td>
</tr>
<tr>
<td>16</td>
<td>2</td>
<td>91251A354</td>
<td>MMC</td>
<td>10-32 UNF x 2-1/4” SHCS</td>
</tr>
<tr>
<td>17</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>SS collar for flex hose, 2” OD</td>
</tr>
<tr>
<td>18</td>
<td>1</td>
<td>HL22571</td>
<td>Chrysler</td>
<td>Throttle position sensor</td>
</tr>
<tr>
<td>19</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Throttle body housing, aluminum</td>
</tr>
<tr>
<td>20</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Air intake runner, stainless steel</td>
</tr>
<tr>
<td>21</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Linear actuator mount, aluminum</td>
</tr>
<tr>
<td>22</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>5-bolt mounting flange, stainless steel</td>
</tr>
</tbody>
</table>

MMC = McMaster-Carr
Table B.3: BOM for air intake system; referenced to Figure B.3.

<table>
<thead>
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<th>Description</th>
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<td>1</td>
<td>1</td>
<td>042V</td>
<td>Quick Tanks</td>
<td>159L galvanized steel surge tank</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>Z50MC2-2F</td>
<td>Meriam</td>
<td>Laminar flow element with filter</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>6820K41</td>
<td>MMC</td>
<td>Rubber/SS pipe coupling, 2” x 1-1/2”</td>
</tr>
<tr>
<td>4</td>
<td>1</td>
<td>7753K159</td>
<td>MMC</td>
<td>2” x 6” one end threaded pipe, steel</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
<td>44605K328</td>
<td>MMC</td>
<td>2” x 1-1/4” steel pipe reducer</td>
</tr>
<tr>
<td>6</td>
<td>1</td>
<td>44615K467</td>
<td>MMC</td>
<td>1-1/4” x 4” threaded pipe, steel</td>
</tr>
<tr>
<td>7</td>
<td>1</td>
<td>44605K117</td>
<td>MMC</td>
<td>1-1/4” x 90 deg. elbow, steel</td>
</tr>
<tr>
<td>8</td>
<td>1</td>
<td>44615K487</td>
<td>MMC</td>
<td>1-1/4” x 6” threaded pipe, steel</td>
</tr>
<tr>
<td>9</td>
<td>1</td>
<td>9157K46</td>
<td>MMC</td>
<td>1-1/4” x 6” one end threaded pipe, 304 SS</td>
</tr>
<tr>
<td>10</td>
<td>2</td>
<td>44685K54</td>
<td>MMC</td>
<td>2” x 1-1/4” SS socket weld pipe flange</td>
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<tr>
<td>11</td>
<td>1</td>
<td>45735K118</td>
<td>MMC</td>
<td>1-1/2” x 1-1/4” SS pipe reducer</td>
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<tr>
<td>12</td>
<td>1</td>
<td>4347K352</td>
<td>MMC</td>
<td>1-1/2” x 5” 304 SS schedule 10 pipe</td>
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<tr>
<td>13</td>
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<td>45735K235</td>
<td>MMC</td>
<td>1-1/2” x 45 deg. SS elbow</td>
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<tr>
<td>14</td>
<td>1</td>
<td>4347K352</td>
<td>MMC</td>
<td>1-1/2” x 8.5” 304 SS schedule 10 pipe</td>
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<tr>
<td>15</td>
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<td>45735K215</td>
<td>MMC</td>
<td>1-1/2” x 90 deg. SS elbow, long sweep</td>
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<td>90025K388</td>
<td>MMC</td>
<td>Quad o-ring seal</td>
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<td>Custom</td>
<td>N/A</td>
<td>Throttle body assembly (see Table B.2)</td>
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MMC = McMaster-Carr
Table B.4: BOM for valvetrain components; referenced to Figure B.4

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<td>3</td>
<td>Custom</td>
<td>N/A</td>
<td>Bearing cap, 6061-T6 aluminum</td>
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<tr>
<td>3</td>
<td>4</td>
<td>5909K49</td>
<td>MMC</td>
<td>Hardened bearing washer, steel, .032” thick</td>
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<tr>
<td>4</td>
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<td>5909K36</td>
<td>MMC</td>
<td>Needle-roller thrust bearing, 1” ID</td>
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<tr>
<td>5</td>
<td>3</td>
<td>CB1479P</td>
<td>Clevite</td>
<td>2-pc hydrodynamic bearing shell</td>
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<tr>
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<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Camshaft, 1018 HR steel</td>
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<tr>
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<td>6086K116</td>
<td>MMC</td>
<td>JA style bushing, steel, .25” keyway</td>
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<tr>
<td>8</td>
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<td>P26-8MGT-20</td>
<td>Gates</td>
<td>20mm wide x 8mm pitch, ductile iron</td>
</tr>
<tr>
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<td>2</td>
<td>Custom</td>
<td>N/A</td>
<td>Camshaft lobe, 100Cr6 hardened steel</td>
</tr>
<tr>
<td>10</td>
<td>10</td>
<td>92855A616</td>
<td>MMC</td>
<td>M6x1x16mm SS, low profile SHCS</td>
</tr>
<tr>
<td>11</td>
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<td>DWC-148JJ-12</td>
<td>DWSC</td>
<td>Music wire, 42.75 N/mm</td>
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<td>Custom</td>
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<td>Spring retainer, Ti-6Al-4V titanium</td>
</tr>
<tr>
<td>13</td>
<td>2</td>
<td>91415A040</td>
<td>MMC</td>
<td>M6x0.75mm steel nut</td>
</tr>
<tr>
<td>14</td>
<td>2</td>
<td>Custom</td>
<td>N/A</td>
<td>Tappet, Ti-6Al-4V titanium, DLC coated</td>
</tr>
<tr>
<td>15</td>
<td>2</td>
<td>Custom</td>
<td>N/A</td>
<td>Rocker arm, 4340 steel, DLC coated</td>
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<td>Unknown</td>
<td>Chrysler</td>
<td>Follower bearing assembly</td>
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<td>BRG-20610</td>
<td>Jesel Inc.</td>
<td>Needle bearing, .750” OD x .561” ID x .750” L</td>
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<td>SFT-SS0004</td>
<td>Jesel Inc.</td>
<td>Rocker shaft w/ retaining rings (not shown)</td>
</tr>
<tr>
<td>19</td>
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<td>92610A385</td>
<td>MMC</td>
<td>5/16-18 x 1.25” steel SHCS, Torx-Plus drive</td>
</tr>
<tr>
<td>20</td>
<td>4</td>
<td>VS1012</td>
<td>Ferrea</td>
<td>Viton® seal (all valves)</td>
</tr>
<tr>
<td>21</td>
<td>3</td>
<td>VG1038</td>
<td>Ferrea</td>
<td>Bronze valve guide, RPVs and intake valve</td>
</tr>
<tr>
<td>22</td>
<td>2</td>
<td>Custom</td>
<td>Ferrea</td>
<td>Reverse poppet valve</td>
</tr>
<tr>
<td>23</td>
<td>2</td>
<td>16851-1328-0</td>
<td>Kubota</td>
<td>OE Valve cap</td>
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<tr>
<td>24</td>
<td>4</td>
<td>14601-1336-0</td>
<td>Kubota</td>
<td>OE split-lock collet</td>
</tr>
<tr>
<td>25</td>
<td>2</td>
<td>14601-1333-0</td>
<td>Kubota</td>
<td>OE spring retainer</td>
</tr>
<tr>
<td>26</td>
<td>2</td>
<td>14601-1324-0</td>
<td>Kubota</td>
<td>OE valve spring</td>
</tr>
<tr>
<td>27</td>
<td>1</td>
<td>VG1038</td>
<td>Ferrea</td>
<td>Bronze valve guide, shortened to 34 mm</td>
</tr>
<tr>
<td>28</td>
<td>2</td>
<td>14601-1311-0</td>
<td>Kubota</td>
<td>Intake and exhaust valve</td>
</tr>
<tr>
<td>29</td>
<td>2</td>
<td>71170</td>
<td>Dura-bond</td>
<td>Intake/exhaust valve seat, tungsten carbide</td>
</tr>
<tr>
<td>30</td>
<td>4</td>
<td>98317A231</td>
<td>MMC</td>
<td>Stainless steel external retaining ring</td>
</tr>
<tr>
<td>31</td>
<td>2</td>
<td>Custom</td>
<td>N/A</td>
<td>Modified OE Kubota rocker shaft</td>
</tr>
<tr>
<td>32</td>
<td>4</td>
<td>93070A147</td>
<td>MMC</td>
<td>M6x1x20mm steel low profile SHCS</td>
</tr>
<tr>
<td>33</td>
<td>2</td>
<td>15841-1403-6</td>
<td>Kubota</td>
<td>OE rocker arm</td>
</tr>
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</table>

MMC = McMaster-Carr
DWSC = Diamond Wire Spring Company
Table B.5: BOM for crank angle encoder assembly; referenced to Figure B.5

<table>
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<td>1</td>
<td>XH25D</td>
<td>BEI Sensors</td>
<td>Optical encoder, XH25D-SS-900-ABZC-28V/V-SM18</td>
</tr>
<tr>
<td>2</td>
<td>8</td>
<td>92196A321</td>
<td>MMC</td>
<td>1/4”-28x0.75” SHCS</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Encoder mount: faceplate, aluminum</td>
</tr>
<tr>
<td>4</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Encoder mount: right side, aluminum</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Encoder mount: left side, aluminum</td>
</tr>
<tr>
<td>6</td>
<td>1</td>
<td>SCA-06E-06E</td>
<td>GPI1</td>
<td>3/8” zero-backlash shaft coupler</td>
</tr>
<tr>
<td>7</td>
<td>1</td>
<td>15881-9103-0</td>
<td>Kubota</td>
<td>OE crank bolt modified to include 3/8” shaft</td>
</tr>
</tbody>
</table>

MMC = McMaster-Carr
GPI = Gurley Precision Instruments

Table B.6: BOM for PTO shaft and support assembly; Figure B.6.

<table>
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<th>Qty.</th>
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<th>Vendor</th>
<th>Description</th>
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</thead>
<tbody>
<tr>
<td>1 (a,b,c)</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Bearing support housing, steel weldment</td>
</tr>
<tr>
<td>2</td>
<td>2</td>
<td>97355A405</td>
<td>MMC</td>
<td>M8x24 dowel pin, steel, pull-out</td>
</tr>
<tr>
<td>3</td>
<td>7</td>
<td>01123-50820</td>
<td>Kubota</td>
<td>M8 OE flywheel cover bolt</td>
</tr>
<tr>
<td>4</td>
<td>5</td>
<td>15852-2516-0</td>
<td>Kubota</td>
<td>M10 OE flywheel bolt</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Stub shaft, 4140 steel</td>
</tr>
<tr>
<td>6</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>1/4” x 1/4” key stock</td>
</tr>
<tr>
<td>7</td>
<td>1</td>
<td>P40-8MGT-50</td>
<td>Gates</td>
<td>GT sprocket, 8mm pitch, 4” OD</td>
</tr>
<tr>
<td>8</td>
<td>1</td>
<td>2012-1</td>
<td>Gates</td>
<td>Taperlock sprocket bushing</td>
</tr>
<tr>
<td>9</td>
<td>1</td>
<td>2780T66</td>
<td>MMC</td>
<td>Ball bearing, 1” ID, 2” OD, 9/16” W</td>
</tr>
<tr>
<td>10</td>
<td>1</td>
<td>99142A590</td>
<td>MMC</td>
<td>Retaining ring, steel</td>
</tr>
</tbody>
</table>

MMC = McMaster-Carr

Table B.7: BOM for dynamometer belt tensioner; referenced to Figure B.7.

<table>
<thead>
<tr>
<th>Item</th>
<th>Qty.</th>
<th>Part No.</th>
<th>Vendor</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Tensioner base, aluminum</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>91257A736</td>
<td>MMC</td>
<td>1/2-13 x 6 UNC, grade 8 bolt</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Eccentric shaft, ground and hardened steel</td>
</tr>
<tr>
<td>4</td>
<td>2</td>
<td>7929K48</td>
<td>MMC</td>
<td>Steel needle roller bearing, 1-3/4” ID</td>
</tr>
<tr>
<td>5</td>
<td>1</td>
<td>Custom</td>
<td>N/A</td>
<td>Belt roller, aluminum</td>
</tr>
<tr>
<td>6</td>
<td>1</td>
<td>91590A141</td>
<td>MMC</td>
<td>Retaining ring, 1-3/4” ID, stainless steel</td>
</tr>
<tr>
<td>7</td>
<td>1</td>
<td>90630A125</td>
<td>MMC</td>
<td>1/2-13 UNC nylon lock nut, grade 8</td>
</tr>
</tbody>
</table>

MMC = McMaster-Carr
Appendix C

Matlab Files

Valve Spring Design and Selection

% Spring Force Calculation
clear
clc
Fpre = 360; % Spring preload force [N]
dY = 3.5; % Spring compression at full lift [mm]
SHC = 2; % Solid height clearance [in]
RPM = 1500; % Engine speed [RPM]
g = 9.807; % Acceleration due to gravity [m/s^2]

% USER DEFINED PARAMETERS
%---------------------------------------------------------------
% Prompt user for spring material selection
mat_list = {'Music Wire','Oil-Tempered Steel','Hard-Drawn Steel',...
            'Cr-Va Steel','Cr-Si Steel'};
[mat] = listdlg('SelectionMode', 'single', 'ListSize',[160 160], ...
                'PromptString','Select spring material:', 'ListString', mat_list);

% 1 - Music Wire
% 2 - Oil-tempered steel
% 3 - Hard-drawn steel
% 4 - Cr-Va steel
% 5 - Cr-Si steel
% Material properties
if mat == 1
    G = 79.5*10^-9; % Shear modulus [GPa]
w = 7860; % Density [kg/m^3]
A = 2153.5; % Sut coefficient [MPa]
m = -0.1625; % Sut exponent
elseif mat == 2
    G = 79.5*10^-9; % Shear modulus [GPa]
Appendix C: MATLAB Scripts

w = 7860; % Density [kg/m^3]
A = 1831.2; % Sut coefficient [MPa]
m = -0.1833; % Sut exponent

elseif mat == 3
G = 79.5*10^9; % Shear modulus [GPa]
w = 7860; % Density [kg/m^3]
A = 1753.3; % Sut coefficient [MPa]
m = -0.1822; % Sut exponent

elseif mat == 4
G = 79.5*10^9; % Shear modulus [GPa]
w = 7860; % Density [kg/m^3]
A = 1909.9; % Sut coefficient [MPa]
m = -0.1453; % Sut exponent

elseif mat == 5
G = 79.5*10^9; % Shear modulus [GPa]
w = 7860; % Density [kg/m^3]
A = 2059.2; % Sut coefficient [MPa]
m = -0.0934; % Sut exponent
end

end

% Prompt user to select wire diameters for evaluation (single or multiple)
d_str = {'0.092 ','0.098 ','0.105 ','0.112 ','0.125 ','0.135 ','0.148 ','0.162 ','0.177 '};
[d_sel,ok] = listdlg('SelectionMode', 'multiple', 'ListSize',[200 160],...
'PromptString','Select wire diameter(s) to evaluate:','ListString',d_str);
d_list = [0.092,0.098,0.105,0.112,0.125,0.135,0.148,0.162,0.177];
d = 25.4*d_list(d_sel);

% Prompt user to input desired (design) factor of safety
prompt1 = {'Design Factor of Safety: '};
dlg_title = 'Input desired factor of safety';
num_lines = 1;
def = {'1.2 '};
options.Resize = 'on';
nd = str2double(inputdlg(prompt1,dlg_title,num_lines,def,options));

% Prompt user for spring end design
end_str = {'Plain ends ','Plain ends ground ','Closed ends ','Closed and ground ends '};
[end_sel] = listdlg('SelectionMode', 'single', 'ListSize',[200 160],...
'PromptString','Select spring end type:','ListString',end_str);
if end_sel == 1
Ne = 0;
elseif end_sel == 2
Ne = 1;
elseif end_sel == 3 || end_sel ==4
Ne = 2;
end

% Prompt user for spring end constraints
con_str = {'Both Ends Hinged ','One Fixed , One Hinged','One Clamped , One Free ','Both Ends Fixed '};
[con_sel] = listdlg('SelectionMode', 'single', 'ListSize',[200 160],...
'PromptString','Select spring end constraints: ','ListString',con_str);
if con_sel == 1
alpha = 1;
elseif con_sel == 2
alpha = 0.7;
elseif con_sel == 3
alpha = 2;
elseif con_sel == 4
alpha = 0.5;
end

% Prompt user for coiling process
temp_str = {'Hot','Cold'};
[temp_sel] = listdlg('SelectionMode', 'single', 'ListSize',[160 160],...
                      'PromptString','Select spring forming process:','ListString',temp_str);

% Prompt user for coiling process
options.Default = 'Yes';
options.Interpreter = 'none';
preset = questdlg('Is the spring preset?','Spring Preset Options',...
                   'Yes','No',options);
switch preset
  case 'Yes'
    set = 1;
  case 'No'
    set = 2;
end

% Prompt user for shot peening process
options.Default = 'Yes';
options.Interpreter = 'none';
peened = questdlg('Is the spring shotpeened?','Shot Peening Options',...
                   'Yes','No',options);
switch peened
  case 'Yes'
    peen = 1;
  case 'No'
    peen = 2;
end

if temp_sel == 1
  if set == 1
    if peen == 1
      KU = 0.7000;
      CS = 0.7757;
      M = -0.0139;
      CE = 0.4579;
      Y = -0.0180;
    elseif peen == 2
      KU = 0.7000;
      CS = 0.7757;
      M = -0.0139;
      CE = 0.5758;
      Y = -0.0537;
    end
  elseif set == 2
    if peen == 1
      KU = 0.5600;
      CS = 0.5546;
      M = -0.0090;
    elseif peen == 2
      KU = 0.5600;
      CS = 0.5546;
      M = -0.0090;
    end
CE = 0.5021;
Y = -0.0206;
elseif peen == 2
    KU = 0.5600;
    CS = 0.5546;
    M = -0.0090;
    CE = 0.6620;
    Y = -0.0622;
end
end
elseif temp_sel == 2
    KU = 0.7400;
    CS = 0.8300;
    M = -0.0215;
    CE = 1.8080;
    Y = -0.1300;
end

% MATERIAL PROPERTIES & CALCULATIONS:
% -----------------------------------------------------------------------------
% From Table 10-4 of Shigley 2008; constants A & m are used to calculate
% the estimated tensional strength of a given material.
% For Chrome-Vanadium:
Sut = A.*(d.^m);          % Ultimate tensional strength [MPa]
Ssy = 0.4*Sut;            % Shear design strength (yield) [MPa]
Tau_max = Ssy/nd;         % Static design shear stress (at full lift)
% STATIC SPRING ANALYSIS:
% -----------------------------------------------------------------------------
c = [4:0.5:11];
% Preallocate arrays for fast loop time
D = zeros(length(c),length(d));
DD = zeros(length(c),length(d));
ID = zeros(length(c),length(d));
KB = zeros(length(c),length(d));
Fmax = zeros(length(c),length(d));
k = zeros(length(c),length(d));
Na = zeros(length(c),length(d));
Nt = zeros(length(c),length(d));
Ls = zeros(length(c),length(d));
Lf = zeros(length(c),length(d));
Li = zeros(length(c),length(d));
Lcr = zeros(length(c),length(d));
Lratio = zeros(length(c),length(d));
Fs = zeros(length(c),length(d));
Tau_pre = zeros(length(c),length(d));
Tau_s = zeros(length(c),length(d));
KS1 = zeros(length(c),length(d));
KS2 = zeros(length(c),length(d));
KE = zeros(length(c),length(d));
n = zeros(length(c),length(d));
KS2Max = zeros(length(c),length(d));
KS2ratio = zeros(length(c),length(d));
mass = zeros(length(c),length(d));
fn = zeros(length(c),length(d));
f_ratio = zeros(length(c),length(d));
pitch = zeros(length(c),length(d));

% Calculate required parameters for spring design. The methodology used % below assumes the maximum force occurs at the design shear stress level, % which occurs at maximum operational spring deflection.
for i = 1:length(c)
    for j = 1:length(d)
        % SPRING PARAMETERS AND STRESS VALUES:
        D(i,j) = c(i)*d(j);    % Mean spring diameter [mm]
        OD(i,j) = D(i,j) + d(j); % Outside diameter [mm]
        ID(i,j) = D(i,j) - d(j); % Inside diameter [mm]
        KB(i,j) = (4*c(i)+2)/(4*c(i)-3); % Bergstrassor factor (curvature)
        Fmax(i,j) = pi*d(j)^3*Tau_max(j)/(8*KB(i,j)*D(i,j)); % Static [N]
        k(i,j) = (Fmax(i,j)-Fpre)/dY;    % Spring constant [N/mm]
        Na(i,j) = G*d(j)^4/(8*10^6*k(i,j)*D(i,j)^3); % No. of active coils
        Nt(i,j) = Na(i,j)+Ne;    % Total no. of coils including ends
        Ls(i,j) = Nt(i,j)*d(j);    % Solid length [mm]
        Lf(i,j) = Ls(i,j)+SHC+dY+Fpre/k(i,j); % Free length [mm]
        Li(i,j) = Ls(i,j)+SHC+dY; % Installed length [mm]
        Lcr(i,j) = 2.63*D(i,j)/alpha; % Critical length (max)
        Lratio(i,j) = Lf(i,j)/Lcr(i,j); % Critical length ratio
        Fs(i,j) = k(i,j)*(Lf(i,j)-Ls(i,j)); % Solid force [N]
        Tau_pre(i,j) = 8*KB(i,j)*Fpre*D(i,j)/(pi*d(j)^3); % Preload [MPa]
        Tau_s(i,j) = 8*KB(i,j)*Fs(i,j)*D(i,j)/(pi*d(j)^3); % Solid [MPa]

        % FATIGUE ANALYSIS: (see Stone, 'Fatigue Lift Estimates...')
        KS1(i,j) = Tau_pre(i,j)/Sut(j);
        KS2(i,j) = Tau_max(j)/Sut(j);
        KE(i,j) = KU*(KS2(i,j)-KS1(i,j))/(2*KU-(KS2(i,j)+KS1(i,j)));
        if KE(i,j) < 0
            n(i,j) = 1;
        else
            n(i,j) = exp((1/Y)*log(KE(i,j)/CE));
        end
        KS2Max(i,j) = CS*n(i,j)^M;
        KS2ratio(i,j) = KS2Max(i,j)/KS2(i,j);

        % RESONANT FREQUENCY ANALYSIS:
        % Note: According to Shigley, the natural frequency of the spring % should be 15-20 times greater than the excitation frequency.
        mass(i,j) = pi^2*d(j)^2*D(i,j)*Na(i,j)*w/(4*10^9);
        fn(i,j) = 0.5*sqrt(k(i,j)*1000/mass(i,j)); % Natural frequency [Hz]
        f_ratio(i,j) = fn(i,j)/fres;   % This value should be > 15.

        % SPRING PITCH
        if end_sel == 1
            pitch(i,j) = (Lf(i,j)-d(j))/Na(i,j);
        elseif end_sel == 2
            pitch(i,j) = Lf(i,j)/(Na(i,j)+1);
elseif end_sel == 3
    pitch(i,j) = (Lf(i,j)-3*d(j))/Na(i,j);
elseif end_sel == 4
    pitch(i,j) = (Lf(i,j)-2*d(j))/Na(i,j);
end
end

% SPRING DESIGN AUTO SELECT
% ------------------------------------------------------------------------
% Find cell coordinates within the specified variable arrays that match
% the design requirements. These coordinates are then placed into a single
% array called 'iArray'. A count array is then produced which increments a
% counter each time a set of coordinates is read and places that count in
% the corresponding coordinate cell. The highest values in the count array
% are therefore the most suitable spring designs.

% Prompt user for design requirements and/or constraints
prompt2 = {' Maximum OD [mm]',' Minimum ID [mm]',' Minimum Free Length [mm]',' Maximum Free Length [mm]',' Min. Number of Active Coils ',' Max. Number of Active Coils ',' Frequency Ratio ',' Min. Desired Life Cycles '};
dlg_title = 'Input Design Constraints ';
um_lines = 1;
def = {'30 ','14 ','0 ','80 ','3 ','15 ','15 ','10^6 '};
options.Resize = 'on';
constr = str2double ( inputdlg ( prompt2 , dlg_title , num_lines ,def , options ));
[ ODr ODc ] = find (OD <= constr (1)); % Find OD that is less than 30mm.
iOD = [ ODr ODc ]; % Combine into single index array
[ IDr IDc ] = find (ID >= constr (2));
iID = [ IDr IDc ];
[ Lfr Lfc ] = find (constr (3) < Lf & Lf <= constr (4));
iLf = [ Lfr Lfc ];
[ Nar Nac ] = find (constr (5) <= Na & Na <= constr (6));
iNa = [ Nar Nac ];
[ kr kc ] = find (k >0);
iK = [ kr kc ];
[ Lratio Lratioc ] = find (0 <= Lratio & Lratio <1);
iLratio = [ Lratio Lratioc ];
[f_ratio f_ratioc ] = find (f_ratio >= constr (6));
if_ratio = [ f_ratio f_ratioc ];
[nr nc ] = find (n> constr (7));
in = [ nr nc ];
[ KS2ratio KS2ratio ] = find (KS2ratio >1);
iKS2ratio = [ KS2ratio KS2ratio ];
% Combine all indexing arrays into a single array
iArray = [ iOD ; iID ; iLf ; iNa ; ik; iLratio ; if_ratio ; in ; iKS2ratio ];
% Use the coordinates of the
for q = 1:length(iArray)
    count(iArray(q,1),iArray(q,2)) = count(iArray(q,1),iArray(q,2))+ 1;
end
[C1 C2] = find(count == max(max(count)));
select = [C1 C2];
% Preallocate arrays for faster computation
d_col = zeros(1,length(C1));
OD_col = zeros(1,length(C1));
ID_col = zeros(1,length(C1));
Fmax_col = zeros(1,length(C1));
Lf_col = zeros(1,length(C1));
Lratio_col = zeros(1,length(C1));
Ls_col = zeros(1,length(C1));
Li_col = zeros(1,length(C1));
Na_col = zeros(1,length(C1));
C_col = zeros(1,length(C1));
fn_col = zeros(1,length(C1));
fratio_col = zeros(1,length(C1));
k_col = zeros(1,length(C1));
mass_col = zeros(1,length(C1));
n_col = zeros(1,length(C1));
pitch_col = zeros(1,length(C1));
% Retrieve parameters for selected cases
for p = 1:length(select)
    d_col(p) = d(:,select(p,2));
    OD_col(p) = OD(select(p,1),select(p,2));
    ID_col(p) = ID(select(p,1),select(p,2));
    Fmax_col(p) = Fmax(select(p,1),select(p,2));
    Lf_col(p) = Lf(select(p,1),select(p,2));
    Lratio_col(p) = Lratio(select(p,1),select(p,2));
    Ls_col(p) = Ls(select(p,1),select(p,2));
    Li_col(p) = Li(select(p,1),select(p,2));
    Na_col(p) = Na(select(p,1),select(p,2));
    C_col(p) = c(:,select(p,1));
    fn_col(p) = fn(select(p,1),select(p,2));
    fratio_col(p) = f_ratio(select(p,1),select(p,2));
    k_col(p) = k(select(p,1),select(p,2));
    mass_col(p) = mass(select(p,1),select(p,2));
    n_col(p) = n(select(p,1),select(p,2));
    pitch_col(p) = pitch(select(p,1),select(p,2));
end
% Combine all selected cases into one array
result = [d_col' OD_col' ID_col' Fmax_col' Lf_col' Lratio_col' Ls_col' Li_col'
          Na_col' C_col' fn_col' fratio_col' k_col' mass_col' n_col' pitch_col'];
% Output selected spring design parameters to Excel sheet for viewing
xlswrite('springtable.xls', result, 'Matlab Output', 'A3');
clear all
close all
clc
global tIGN RPM XOVC
% NOTE: Crank Angle (CA) data is taken from cylinder no. 2, which is
% advanced 20 deg. from cylinder no. 1. Data referenced to cylinder no. 1
% is thus shifted to account for this

% DEFINE ENGINE PARAMETERS
% -----------------------------------------------------------------------
B = 67;       % bore [mm] 
S = 68;       % stroke [mm] 
l = 98.2;     % connecting rod length [mm] 
a = 0.5*S;    % crank throw [mm] 
hc = 0.5;     % clearance height [mm] 

% NOMINAL VALVE TIMINGS
% -----------------------------------------------------------------------
IVO = 6;      % degrees ATDC-c 
IVC = 191;    % degrees ATDC-c 
XIVO = 318;   % degrees ATDC-c 
XIVC = 3;     % degrees ATDC-c 
XOVO = 345;   % degrees ATDC-e 
XOVC = 27;    % degrees ATDC-e 
EVO = 159;    % degrees ATDC-e 
EVC = 348;    % degrees ATDC-e 

% PRELIM ENGINE CALCS
% -----------------------------------------------------------------------
Vc = pi/4*B^2*hc;     % clearance volume [mm]^3 
Vd = pi/4*B^2*S;      % displaced volume [mm]^3 
Vt = Vc + Vd;         % total volume [mm]^3 
% Including piston recesses: 
Vc1 = Vc + 450;       % clearance volume of cylinder 1 [mm]^3 
Vc2 = Vc + 450 + 565; % clearance volume of cylinder 2 [mm]^3 

% IMPORT RAW TEST DATA FOR ANALYSIS
% -----------------------------------------------------------------------
% Read raw data from TDMS files (filestr = AI data, filestr2 = TC data)
[filename, filepath] = uigetfile('*..tdms','Select a file for analysis','C:\');
filestr = strcat(filepath,filename);
filestr2 = strcat(filepath,strtok(filename,'.'),'-TC-data.tdms');
raw_data = TDMS_getStruct(filestr);
tc_data = TDMS_getStruct(filestr2);
disp(['Selected data file: ', num2str(filename)])
disp('')
disp('USER INPUTS:')
disp('--------------------------------------------------------')

% PROMPT USER TO INPUT THROTTLE POSITION
TPS = input('Enter throttle position in ''%'':
% PROMPT USER TO INPUT SPARK (IGNITION) TIMING (for MFB calc)
tIGN = input('Enter the ignition timing in deg CA after TDC:\n');
disp('-------------------------------------------------------')

% ASSIGN VARIABLE NAMES TO TDMS DATA:

CA = raw_data.Untitled.CA__deg_.data;
IPP = raw_data.Untitled.IPP__bar_.data;
ICP1_raw = raw_data.Untitled.ICP1__bar_.data;
ICP2_raw = raw_data.Untitled.ICP2__bar_.data;
XOVR = raw_data.Untitled.XOVR__bar_.data;
TRQ = raw_data.Untitled.TRQ__Nm_.data;
MFF = raw_data.Untitled.MFF__g_s_.data;
MAF = raw_data.Untitled.MAF__g_s_.data;
AFER = raw_data.Untitled.AFER__phi_.data;

if (isfield(tc_data, 'Temps__degC_')~=0)
    xovr_temp = tc_data.Temps__degC_.Xover.data;
cylhd1_temp = tc_data.Temps__degC_.CylHd1.data;
cylhd2_temp = tc_data.Temps__degC_.CylHd2.data;
cylhd3_temp = tc_data.Temps__degC_.CylHd3.data;
oilrail_temp = tc_data.Temps__degC_.OilRail.data;
tdcw_temp = tc_data.Temps__degC_.TDCW.data;
radout_temp = tc_data.Temps__degC_.RadOut.data;
intair_temp = tc_data.Temps__degC_.IntAir.data;
fuel_temp = tc_data.Temps__degC_.Fuel.data;
oilpan_temp = tc_data.Temps__degC_.OilPan.data;
tstat_temp = tc_data.Temps__degC_.Tstat.data;
exh_temp = tc_data.Temps__degC_.Exhaust.data;
RPM = tc_data.EngSpd.RPM.data;
else
    xovr_temp = 'NAN';
cylhd1_temp = 'NAN';
cylhd2_temp = 'NAN';
cylhd3_temp = 'NAN';
oilrail_temp = 'NAN';
tdcw_temp = 'NAN';
radout_temp = 'NAN';
intair_temp = 'NAN';
fuel_temp = 'NAN';
oilpan_temp = 'NAN';
tstat_temp = 'NAN';
exh_temp = 'NAN';
RPM = input('No RPM data found - please enter manually');
end

% RESHAPE DATA VECTORS INTO CYCLE-COLUMN ARRAYS

CA = reshape(CA,3600,[]);
IPP = reshape(IPP,3600,[]);
P1raw = reshape(ICP1_raw,3600,[]);
P2raw = reshape(ICP2_raw,3600,[]);
XOVR = reshape(XOVR,3600,[]);
TRQ = reshape(TRQ,3600,[]);
MFF = reshape(MFF,3600,[]);
MAF = reshape(MAF,3600,[]);
AFER = reshape(AFER,3600,[]);

% CHECK TO ENSURE DATA FILE STARTS AT TDC
% ---------------------------------------------------------------
if CA(1)== 0
    disp(['WARNING: Encoder data does not begin at TDC! 1st entry is: '...
         num2str(CA(1)) ' deg'])
    if CA(1)<180
        shift = round(CA(1)*10);
    else
        shift = round((CA(1)-360)*10);
    end
    % Adjust data so columns start at TDC
    CA = circshift(CA,[shift 0]);
    IPP = circshift(IPP,[shift 0]);
    P1raw = circshift(P1raw,[shift 0]);
    P2raw = circshift(P2raw,[shift 0]);
    XOVR = circshift(XOVR,[shift 0]);
    TRQ = circshift(TRQ,[shift 0]);
    MFF = circshift(MFF,[shift 0]);
    MAF = circshift(MAF,[shift 0]);
    AFER = circshift(AFER,[shift 0]);
end

% CHECK NUMBER OF ENGINE N IN DATA FILE
% ---------------------------------------------------------------
N = size(CA,2);
disp(['num2str(N) ' engine cycles analyzed from data file.'])

% CHECK FOR INCORRECT ENCODER VALUES IN EACH CYCLE
% ---------------------------------------------------------------
cycerr = find(any(diff(CA,1,2))==0);  % problematic cycles (columns)
if isempty(cycerr) == 0
    if size(cycerr,2) > 1
        degvect = (0:0.1:359.9)';
        CAraw = CA;
        CA = repmat(degvect,1,300);
        disp(['num2str(size(cycerr,2)) ' cycles with faulty encoder data.'])
        disp('Artificial CA data will be used. Check data for errors.')
    else
        CA(:,cycerr) =[];
        IPP(:,cycerr) =[];
        P1raw(:,cycerr) =[];
        P2raw(:,cycerr) =[];
        XOVR(:,cycerr) =[];
        TRQ(:,cycerr) =[];
        MFF(:,cycerr) =[];
        MAF(:,cycerr) =[];
        AFER(:,cycerr) =[];
        N = size(CA,2);
        disp(['Cycles ' num2str(cycerr) ' contain faulty encoder data.'])
    end
else
    disp(['Cycles ' num2str(cycerr) ' contain faulty encoder data.'])
end


```matlab
disp(['Continuing with remaining ' num2str(N) ' cycles.'])
end
end

% ROUND ENGINE RPM TO NEAREST INTEGER
% ---------------------------------------------------------------
RPM = round(mean(RPM));

% PROMPT USER TO SMOOTH PRESSURE DATA
% ---------------------------------------------------------------
P1f = lowpassfilt(P1raw, RPM, 3000);
P2f = lowpassfilt(P2raw, RPM, 3000);
Px = lowpassfilt(XOVR, RPM, 3000);

% CALULATE CYLINDER VOLUME FROM CRANK ANGLE
% ---------------------------------------------------------------
V1 = Vc1 + pi/4*B^2*(1+a-(a*cosd(CA)+(1^2-a^2*sind(CA).^2).^0.5));
V2 = Vc2 + pi/4*B^2*(1+a-(a*cosd(CA)+(1^2-a^2*sind(CA).^2).^0.5));

% PEG CYLINDER PRESSURE MEASUREMENTS
% ---------------------------------------------------------------
IPPpeg = zeros(N,1);
P2peg = zeros(N,1);
P1p = zeros(size(P1f));
P2p = zeros(size(P2f));

% Adjustable pegging parameters:
ref1 = 160;   % reference CA (Cyl.1) for IPP start of pegging
ref2 = 180;   % reference CA (Cyl.1) for IPP end of pegging

for i = 1:N
    IPPpeg(i) = mean(IPP((ref1+20:ref2+20)*10,i))-mean(P1f((ref1+20:ref2+20)*10,i));
    P1p(:,i) = P1f(:,i)+IPPpeg(i);
end

for i = 1:N
    if mean(Px(:,)) < 0  % If Px is not available use Cyl.1 for peg
        P2peg(i) = mean(P1p((1:100),i))-mean(P2f((1:100),i));
    else
        % use crossover pressure
        P2peg(i) = mean(Px((1:150),i))-mean(P2f((1:150),i));
    end
    P2p(:,i) = P2f(:,i)+P2peg(i);
end

% ENSEMBLE AVERAGE PRESSURE
P1ens = mean(P1p,2);
P2ens = mean(P2p,2);
PXens = mean(Px,2);

% PRESSURE RISE RATE CALCULATION & MISFIRE DETECTION
% ---------------------------------------------------------------
[PRR PRRmax PRRpeak PRRmaxAvg cyc_misfire] = PRRcalc(CA, P2p);
```
if (cyc_misfire>0)
    CA(:,cyc_misfire) = [ ];
    IPP(:,cyc_misfire) = [ ];
    P1p(:,cyc_misfire) = [ ];
    P2p(:,cyc_misfire) = [ ];
    Px(:,cyc_misfire) = [ ];
    TRQ(:,cyc_misfire) = [ ];
    MFF(:,cyc_misfire) = [ ];
    MAF(:,cyc_misfire) = [ ];
    AFER(:,cyc_misfire) = [ ];
    V1(:,cyc_misfire) = [ ];
    V2(:,cyc_misfire) = [ ];
    no_misfire = length(cyc_misfire);
else
    no_misfire = 0;
end

% CYLINDER PRESSURE ANALYSIS & MISFIRE DETECTION
% ------------------------------------------------------------------
[Pmax, LPP, PmaxAvg, PmaxStd, LPPAvg, LPPcov] = PMAXcalc(CA, P2p);

% POLYTROPIC INDEX CALCULATION
% ------------------------------------------------------------------
% Define start and end points for polytropic index calculations
% POLY C:
postBDC = 60; % degrees after BDC to start PolyC analysis
preXIVO = 2; % degrees before Xin valve opens to end PolyC analysis
% POLY E:
postLPP = 40; % degrees after LPP to start PolyE analysis
preEVO = 10; % degrees before EVO to end PolyE analysis
C1 = postBDC+180;
C2 = XIVO-preXIVO;
E1 = LPP+postLPP; % note: E1 is a vector
E2 = EVO-preEVO;
[nc] = POLYcalc(P1p, V1, C1, C2, 1);
[ne] = POLYcalc(P2p, V2, E1, E2, 2);

% IMEP CALCULATION
% ------------------------------------------------------------------
% Compression Cylinder:
[IMEP1, IMEP1avg, IMEP1std, IMEP1cov] = IMEPcalc(Vd, V1, P1p, 1);
% Expansion Cylinder:
[IMEP2, IMEP2avg, IMEP2std, IMEP2cov] = IMEPcalc(Vd, V2, P2p, 2);
% Net IMEP (by cycle):
IMEPnet = IMEP2+IMEP1;
disp(['The average net IMEP is ' num2str(mean(IMEPnet)) ' bar.'])
% Net IMEP average for N cycles:
IMEPnet_tm = mean(IMEPnet);
% Calculated indicated power for emissions purposes
PWRi = IMEPnet_tm*Vd*10^-5/1000^4*RPM*2*pi/60; %[kW]
% BMEP and FMEP CALCULATION
% -------------------------------------------------------------------------
BMEP = 2*pi*mean(TRQ(:))*10000/Vd; %[bar]
FMEP = IMEPnet_tm-BMEP; %[bar]

% FUEL CONVERSION EFFICIENCY CALCULATION
% -------------------------------------------------------------------------
[nt] = TEFFcalc(IMEPnet, Vd, RPM, MFF);
ntAvg = mean(nt); % overall average fuel conversion efficiency
ntMax = max(nt); % overall maximum conversion efficiency

% VOLUMETRIC EFFICIENCY CALCULATION
% -------------------------------------------------------------------------
if exist('IntAir_temp','var')
    [VE] = VEcalc(IntAir_temp, Vd, RPM, MAF);
else
    [VE] = VEcalc(25, Vd, RPM, MAF);
end

% AVERAGE AIR AND FUEL FLOW RATE FOR DATA SET:
% -------------------------------------------------------------------------
MAFtm = mean(MAF(:));
MFFtm = mean(MFF(:));

% INTAKE PORT PRESSURE ANALYSIS
% -------------------------------------------------------------------------
[IPPcm IPPtm IPP_ensAvg IPP_ensAvg_min] = IPPcalc(CA, IPP, RPM);

% MASS FRACTION BURNED ANALYSIS
% -------------------------------------------------------------------------
[xb, xb_start, xb_end, CA10, CA50, CA90, CA10_90, PR, PRN, PRN10, PRN50,...
    PRN90, PRN10_90] = MFBcalc(P2p, V2, EVO, nc, ne);
CAxb = CA(xb_start:xb_end,:); % Crank angle for MFB curves

% MFB mean statistics
CA10tm = mean(CA10);
CA50tm = mean(CA50);
CA90tm = mean(CA90);
CA0_10tm = mean(CA10-tIGN);
CA10_90tm = mean(CA10_90);
CA10_90min = min(CA10_90);
CA10_90max = max(CA10_90);
CA10_90cov = std(CA10_90)/CA10_90tm*100;

% PRN mean statistics
PRN10tm = mean(PRN10);
PRN50tm = mean(PRN50);
PRN90tm = mean(PRN90);
PRN0_10tm = mean(PRN10-tIGN);
PRN10_90tm = mean(PRN10_90);
PRN10_90min = min(PRN10_90);
PRN10_90max = max(PRN10_90);
PRN10_90cov = std(PRN10_90)/PRN10_90tm*100;
PRN0_10cov = std(PRN0_10-tIGN)/PRN0_10tm*100;
PRN0_90 = PRN90-tIGN;

% AIR FUEL RATIO ANALYSIS
% -------------------------------------------------------------------
[AIR_cm AFR_tm AFRER_cm AFRER_tm] = AFRcalc(MFF, MAF); %<-- calculated AFR
% cm = cycle mean
% tm = total dataset mean
AFER_horiba_tm = mean(AFER(:)); %<-- measured AFR
disp(['The avg. measured equivalence ratio is: ' num2str(AFER_horiba_tm)])

% TEMPERATURE AVERAGING
% -------------------------------------------------------------------
EGT = mean(exh_temp);
IAT = mean(intair_temp);
CHT = mean((cylhd1_temp+cylhd2_temp+cylhd3_temp)/3);
XOT = mean(xovr_temp);
FLT = mean(fuel_temp);

table = {filename RPM tIGN AFR_horiba_tm AFRER_tm N no_misfire TPS AEAP IMEP1avg...
IMEP2avg IMEPnet_tm IMEP1cov IMEP2cov IPPtm IPP_ensAvg_min MAFtm MFFtm EGT...
IAT CHT FLT XOT mean(Px(:)) max(Px(:)) PmaxAvg max(Pmax) LPPAvg LPPcov...
PRRmaxAvg PRRpeak mean(nc) mean(ne) CA10tm CA50tm CA90tm CA10_90tm...
CA10_90min CA10_90max CA10_90cov PRN10tm PRN50tm PRN90tm PRN10_90tm...
PRN10_90min PRN10_90max PRN10_90cov VE ntAvg ntMax PWRi CA0_10tm PRN0_10tm...
PRN0_10cov BMEP};
xlswrite('.../matlab-results.xlsx',table)

PRR Calculation and Misfire Detection

function [PRR PRRmax PRRpeak PRRmaxAvg cyc_misfire] = PRRcalc(CA, P2p)
global tIGN c1 c2

C1 = tIGN*10; % start analysis at time of ignition
C2 = 1800; % end analysis at 180 deg. ATDC

% Extract encoder resolution from crank angle data
res = mean(diff(CA(:,1)));%
% Pressure Rise Rate [bar/deg]
PRR = (diff(P2p(c1:c2,:),1,1))/res;% Misfire detection based upon max PRR not going above 0.1 bar/deg
k=1;
for j=1:size(P2p,2)
    if max(PRR(50:end,j))<0.1
        cyc_misfire(k)=j;
        k=k+1;
    end
end
disp('-------------------------------------------------------------------')
if exist('cyc_misfire','var')
disp(['num2str(length(cyc_misfire)) ... 
     ' misfire(s) detected in the following cycle(s): '...
     num2str(cyc_misfire)])
    % Remove misfire cycle from PRR array
    PRR(:,cyc_misfire)=[];
else
    disp('No misfires occured in this data set.')
cyc_misfire = 0;
end
% Maximum PRR for each cycle [bar/deg]
PRRmax = max(PRR);
% Maximum overall PRR
PRRpeak = max(PRRmax);
% Average maximum PRR
PRRmaxAvg = mean(PRRmax);

Pmax and LPP Determination

function [Pmax , LPP , PmaxAvg , PmaxStd , LPPAvg , LPPcov ] =...
    PMAXcalc ( crank_angle , pressure)
global tIGN
    % PMAXcalc calculates the maximum pressure in a cycle and the average
    % maximum for any given number of cycles. Data should be entered as one
    % column per cycle.
    % ------------------------------------------------------------------------
    % LPP = the location of Pmax in deg CA after TDC
    % PmaxAvg = the average Pmax for given number of cycles
    % PmaxStd = the standard deviation of Pmax
    % LPPAvg = the average location of Pmax in deg CA
    % LPPrelSpk = the location of Pmax relative to spark timing
    % ------------------------------------------------------------------------
    ca1 = tIGN*10+100;  % start evaluation of Pmax 10 deg after spark timing
    ca2 = 1800;         % end Pmax evaluation at BDC
    [Pmax CA_index ]= max(pressure(ca1:ca2,:));
    for i=1:length(CA_index)
        LPP(i) = crank_angle(CA_index(i)+ca1,i);
        LPPrelSpk(i) = crank_angle(CA_index(i)+ca1-tIGN*10,i);
        if LPP(i) == crank_angle(CA_index(i)+ca1) && Pmax(i) < pressure(100,i)
            disp('LPP check being performed...')
            [Pmax2 CA_index2] = max(pressure(ca1+100:ca2,i));
            Pmax(i)=Pmax2;
            LPP(i) = crank_angle(CA_index2+ca1+100,i);
            LPPrelSpk(i) = crank_angle(CA_index2+ca1+100-tIGN*10,i);
        end
    end
    % Compute statistics of maximum pressure
Appendix C: MATLAB Scripts

PmaxAvg = mean(Pmax);
PmaxStd = std(Pmax);
LPPAvg = mean(LPP);
LPPstd = std(LPP);
LPPcov = LPPstd/LPPAvg*100;
end

IMEP Calculation

function [IMEP, IMEPavg, IMEPstd, IMEPcov] = IMEPcalc(Vd, volume, pressure, CylNo)
% IMEPcalc calculates the IMEP for each cycle as well as the average for
% the total number of cycles input into the function.
% Vd = displacement volume of cylinder
% IMEP is 1 row x N columns (where N is the number of cycles)
% IMEPavg is a scalar value (average of N cycles)
if CylNo==1
    pressure = circshift(pressure, -200);
end
if size(volume) == size(pressure)
    Wcycle = zeros(1,size(pressure,2));
    IMEP = zeros(1,size(pressure,2));
    for i = 1:size(pressure,2)
        Wcycle(i) = simps(volume(:,1), pressure(:,i)); % Cycle work [bar-mm^-3]
        IMEP(i) = Wcycle(i)/Vd;
    end
    IMEPavg = mean(IMEP); % Average IMEP for data set
    IMEPstd = std(IMEP); % Standard deviation of IMEP data
    IMEPcov = abs(IMEPstd/IMEPavg*100); % IMEP coefficient of variance
    disp('---------------------------------------------------------------')
    disp(['For cylinder number ' num2str(CylNo) ':'])
    disp(['The average IMEP is ' num2str(IMEPavg) ' bar.'])
    disp(['The standard deviation of IMEP is ' num2str(IMEPstd) ' bar.'])
    disp(['The COV of IMEP is ' num2str(IMEPcov) ' %.'])
else
    IMEP = 0;
    IMEPavg = 0;
    disp('--------------------------------------------------------------------------------')
    disp('IMEP calculation error: Input arguments are of different lengths')
end
end

Volumetric Efficiency Calculation

function [VE] = VEcalc(Vd, RPM, MAF)
rho_air = 1200; % (101325/(287*(273+intakeTemp)))*1000; % Air density [g/m^-3]
MAF_ideal = rho_air*Vd/(60/RPM*1000^-3);
VE = (mean(MAF(:))/MAF_ideal)*100; % Volumetric efficiency [%]
end
Appendix C: MATLAB Scripts

Intake Port Pressure Statistics

function [IPPcm IPPtm IPP_ensAvg IPP_ensAvg_min] = IPPcalc(CAD, IPP, RPM)

datasize = size(IPP);

% Calculate mean IPP for each cycle
IPPcm = mean(IPP);

% Calculate the mean IPP for all cycles
IPPtm = mean(IPP(:));

% Calculate ensemble average of N-cycles (3600 data points)
IPP_ensAvg = mean(IPP,2);

% Find ensemble average minimum:
IPP_ensAvg_min = min(IPP_ensAvg);

disp('-------------------------------------------------------------------')
disp(['The average intake port pressure is: ' num2str(IPPtm) ' bar.'])
disp(['The minimum ensemble average IPP is: ' num2str(IPP_ensAvg_min) ' bar.'])

Mass Fraction Burned Calculation

function [xb, xb_start, xb_end, CA10, CA50, CA90, CA10_90, PR, PRN, PRN10, PRN50,...
PRN90, PRN10_90] = MFBcalc(P, Vol, EVO, nc, ne)

global tIGN RPM XOVC

% Define start and end point of MFB calculation

xb_start = tIGN*10;  % Row index for MFB calc. start
xb_end = EVO*10-10;  % Row index for MFB calc. end

% Decrease filter band to smooth pressure data
P = lowpassfilt(P, RPM, 500);

% Preallocate for processing speed
V0 = zeros(1, size(P,2));
Vf = zeros(1, size(P,2));
P0 = zeros(1, size(P,2));
Pf = zeros(1, size(P,2));

for i = 1:size(P,2)
    VO(i) = Vol(xb_start, i);  % Volume at ignition
    PO(i) = P(xb_start, i);  % Pressure at ignition
    Vf(i) = Vol(xb_end, i);  % Volume at EVO
    Pf(i) = P(xb_end, i);  % Pressure at EVO
end

xb = zeros(xb_end - xb_start, size(P,2));  % Preallocate for speed

PR = zeros(xb_end - xb_start, size(P,2));

if RPM<875
    Pmot = importdata('Pmotor850.mat');
elseif RPM<925
    Pmot = importdata('Pmotor900.mat');
elseif RPM<975
    Pmot = importdata('Pmotor950.mat');
elseif RPM<1025
    Pmot = importdata('Pmotor1000.mat');
else
    Pmot = importdata('Pmotor1075.mat');
end

else
    Pmot = importdata('Pmotor1000.mat');
else
    Pmot = importdata('Pmotor1075.mat');
end

end

if RPM<875
    Pmot = importdata('Pmotor850.mat');
elseif RPM<925
    Pmot = importdata('Pmotor900.mat');
elseif RPM<975
    Pmot = importdata('Pmotor950.mat');
elseif RPM<1025
    Pmot = importdata('Pmotor1000.mat');
else
    Pmot = importdata('Pmotor1075.mat');
end
Pmot = importdata('Pmotor1050.mat');
elseif RPM < 1125
    Pmot = importdata('Pmotor1100.mat');
elseif RPM < 1175
    Pmot = importdata('Pmotor1150.mat');
elseif RPM < 1225
    Pmot = importdata('Pmotor1200_Aug20.mat');
else
    disp('Error: RPM exceeds allowable range. '
         'Pmot = []; end
% Low pass filter motoring data
Pmot = lowpassfilt(Pmot, RPM, 500);
% MFB calculation based on Ball, Stone, Raine equation.
% ---------------------------------------------------------------------
for j = 1: size(P, 2) % j is cycle no.
k = 1;
    for i = (xb_start): (xb_end) % i is data point in cycle
        % IMPROVED R-W METHOD
        xb(k, j) = (P(i, j)^(1/ ne(j)) * (Vol(i, j) * P(i, j)^(1/ nc(j)) -...
            VO(j) * P0(j)^(1/ nc(j))) / (Vf(j) * Pf(j)^(1/ ne(j)) * P(i, j)^(1/ nc(j)) -...
            VO(j) * P0(j)^(1/ nc(j)) * P(i, j)^(1/ ne(j)));
        % ORIGINAL R-W METHOD: (Note: Vf should actually be EOC)
        xbo(k, j) = (Vol(i, j) * P(i, j)^(1/ ne(j)) - VO(j) * P0(j)^(1/ ne(j))) /...
            (Vf(j) * Pf(j)^(1/ ne(j)) - VO(j) * P0(j)^(1/ ne(j)));
        % PRESSURE RATIO
        PR(k, j) = P(i, j) / Pmot(i) - 1;
        k = k + 1;
    end
% NORMALIZED PRESSURE RATIO
PRN(:, j) = PR(:, j) / max(PR(:, j));
end
% Calculate burn durations (CA10, CA50, CA90)
% ---------------------------------------------------------------------
CA10 = zeros(size(xb, 2), 1);
CA50 = zeros(size(xb, 2), 1);
CA90 = zeros(size(xb, 2), 1);
for i = 1: size(xb, 2)
    CA10(i) = (find(xb(:, i) >= 0.1, 1) + xb_start) / 10;
    CA50(i) = (find(xb(:, i) >= 0.5, 1) + xb_start) / 10;
    CA90(i) = (find(xb(:, i) >= 0.9, 1) + xb_start) / 10;
    PRN10(i) = (find(PRN(:, i) >= 0.1, 1) + xb_start) / 10;
    PRN50(i) = (find(PRN(:, i) >= 0.5, 1) + xb_start) / 10;
    PRN90(i) = (find(PRN(:, i) >= 0.9, 1) + xb_start) / 10;
end
PRN10 = PRN10';
PRN50 = PRN50';
PRN90 = PRN90';
CA10_90 = CA90 - CA10;
PRN10_90 = PRN90 - PRN10;
disp(' ---------------------------------------------------------------')
disp(['The average CA10 location is: ' num2str(mean(CA10)) ' deg.'])
disp(['The average CA50 location is: ' num2str(mean(CA50)) ' deg.'])
disp(['The average CA90 location is: ' num2str(mean(CA90)) ' deg.'])
disp(['The CA10-90 combustion duration is: ' num2str(mean(CA10_90)) ' deg.'])
disp(['The average PRN10 location is: ' num2str(mean(PRN10)) ' deg.'])
disp(['The average PRN50 location is: ' num2str(mean(PRN50)) ' deg.'])
disp(['The average PRN90 location is: ' num2str(mean(PRN90)) ' deg.'])
disp(['The PRN10-90 combustion duration is: ' num2str(mean(PRN10_90)) ' deg.'])
end

AFR/AFER Calculation

function [AFRcalc_cm AFRcalc_tm AFERcalc_cm AFERcalc_tm] = AFRcalc(MFF, MAF)
%AFRcalc calculates the air-to-fuel ratio and corresponding equivalence ratio based on measured flow rates of fuel and air.
%AIR Cycle Mean:
AFRcalc_cm = mean(MAF)./mean(MFF);
%AIR Cycle Mean:
AFERcalc_cm = 17.16./AFRcalc_cm;
%AIR Total Dataset Mean:
AFRcalc_tm = mean(MAF(:))/mean(MFF(:));
%AIR Total Dataset Mean:
AFERcalc_tm = 17.16/AFRcalc_tm;
disp('---------------------------------------------------------------------')
disp(['The calculated AFR is ' num2str(AFRcalc_tm),...
     ' which corresponds to an equivalence ratio of ' num2str(AFERcalc_tm)]);
end
Appendix D

Calculations and Derivations

Design-Stage Uncertainty Example Calculation

**Equipment:**

1. *Sierra Mass Flow Meter*
   - Accuracy: ±1.0% of full scale
   - Repeatability: ±0.2% of full scale
   - Full scale: 0.418 g/s of methane
   - 4–20 mA output converted to 1–5 V through 250 Ω resistor
   - ±0.01% resistance uncertainty

2. *National Instruments USB-6356 Data Acquisition Device*
   - Full-scale resolution voltage \( E_{FSR} \) = ±10 V
   - ADC resolution \( M \) = 16-bit
   - AI absolute accuracy = ±2.5 μV

**Mechanical Sensitivity:**

\[
\frac{\text{Mechanical Units}}{\text{Volt}} = \frac{0.418 - 0}{5 - 1} \left[ \frac{\text{g/s}}{\text{V}} \right] = 0.1045 \frac{\text{g/s}}{\text{V}}
\]
Appendix D: Calculations

Uncertainties:

(i) DAQ Device Uncertainty (zero order and bias):

\[
(u_0)_{daq} = \pm \frac{1}{2} \frac{E_{FSR}}{2^M} = \pm \frac{20}{2 \cdot 2^{16}} = \pm 152.6 \, \mu V
\]

\[
(u_c)_{daq} = \pm 2.5 \, \mu V
\]

\[
(u_d)_{daq} = \pm \sqrt{152.6^2 + 2.5^2} = \pm 152.62 \, \mu V \cdot 0.1045 \, g/s/V = \pm 1.6 \times 10^{-5} \, g/s
\]

(ii) Mass Flow Meter Uncertainty:

\[
(u_d)_{fm} = \pm 0.418 \, g/s/V \sqrt{0.01^2 + 0.002^2} = \pm 0.004263 \, g/s
\]

(iii) mA to V Resistor Uncertainty:

\[
(u_d)_{res} = (0.0001)(250 \, \Omega)(20 \times 10^{-3} \, A)(0.1045 \, g/s/V) = \pm 5.2 \times 10^{-5} \, g/s
\]

Overall Uncertainty in Mass Fuel Flow Rate:

\[
U_{MFF} = \pm \sqrt{(u_d)_{daq}^2 + (u_d)_{mf}^2 + (u_d)_{res}^2}
\]

\[
= \pm \sqrt{(1.6 \times 10^{-5})^2 + (0.004263)^2 + (5.2 \times 10^{-5})^2}
\]

\[
= \pm 4.26 \times 10^{-3} \, g/s
\]

The overall uncertainty in the mass fuel flow rate of methane is:

\[
U_{MFF} = \pm 4.26 \, mg/s
\]
Calculation of Equivalence Ratio Based on Exhaust Gas Composition

The generalized global combustion reaction can be written as:

\[ C_n H_m + \frac{n_{O_2}}{\phi} (O_2 + 3.773N_2) = n_P (\tilde{x}_{C_a H_b} C_a H_b + \tilde{x}_{CO} CO + \tilde{x}_{CO_2} CO_2 + \tilde{x}_{O_2} O_2 + \tilde{x}_{N_2} N_2 + \tilde{x}_{NO} NO + \tilde{x}_{NO_2} NO_2 + \tilde{x}_{H_2O} H_2O + \tilde{x}_{H_2} H_2) \]  

where:

- \( n_{O_2} \) is the number of moles of oxygen required for stoichiometric combustion
- \( \phi \) is the air/fuel equivalence ratio
- \( n_P \) is the total number of moles in the exhaust products
- \( \tilde{x}_i \) is the wet mole fraction of species \( i \) in the exhaust products

For combustion of methane, \( CH_4 \), \( n = 1, m = 4 \), the number of moles of oxygen for complete combustion, \( n_{O_2} \), is 2. \( \tilde{x}_{C_a H_b}, \tilde{x}_{CO}, \tilde{x}_{CO_2}, \tilde{x}_{O_2}, \) and \( \tilde{x}_{NO_x} \) are measured values and thus the remaining 7 unknowns are:

- \( \tilde{x}_{N_2}, \tilde{x}_{H_2O}, \tilde{x}_{H_2}, n_P, \phi, b, a \).

From the mass balance of elements in Equation (D.1):

- **C:** \( n = n_P (a \tilde{x}_{C_a H_b} + \tilde{x}_{CO} + \tilde{x}_{CO_2}) \)  
- **H:** \( m = n_P (b \tilde{x}_{C_a H_b} + 2 \tilde{x}_{H_2O} + 2 \tilde{x}_{H_2}) \)  
- **O:** \( \frac{2 n_{O_2}}{\phi} = n_P (\tilde{x}_{CO} + 2 \tilde{x}_{CO_2} + 2 \tilde{x}_{O_2} + \tilde{x}_{NO_x} + \tilde{x}_{H_2O}) \)  
- **N:** \( \frac{7.546 n_{O_2}}{\phi} = n_P (2 \tilde{x}_{N_2} + \tilde{x}_{NO_x}) \)

Note: \( \tilde{x}_{NO_x} \) has been used to show that it represents both \( \tilde{x}_{NO} \) and \( \tilde{x}_{NO_2} \), since they are measured as a combined entity. For calculation purposes, however, \( NO_x \) was treated as NO.

The sum of the mole fractions must equate to one:

\[ 1 = (\tilde{x}_{C_a H_b} + \tilde{x}_{CO} + \tilde{x}_{CO_2} + \tilde{x}_{O_2} + \tilde{x}_{N_2} + \tilde{x}_{NO_x} + \tilde{x}_{H_2O} + \tilde{x}_{H_2}) \]  

Beyond this point, two assumptions will be used to generate the remaining two equations required to solve for \( \phi \) in Equation (D.1):
1. The measured hydrocarbons in the exhaust gases have the same H:C ratio as the fuel. Based on the low values of NMHC measured in this work, this is a valid assumption. The assumption can be written mathematically as:

\[
\frac{b}{a} = \frac{m}{n} \quad (D.7)
\]

2. The relationship between CO, CO₂, H₂O, and H₂ can be described by the following equation:

\[
\frac{\tilde{x}_{CO} \tilde{x}_{H_2O}}{\tilde{x}_{CO_2} \tilde{x}_{H_2}} = K \quad (D.8)
\]

Equation (D.8) takes on the form of an equilibrium constant for the water-gas shift reaction (WGSR)¹, and in fact the values typically used for \( K \) (2.5–4.5), correspond to those of the WGSR equilibrium constant at temperatures between 1500–2000 K. According to Heywood [70], however, \( K \) is an empirically derived value. In this work, the author has observed that changing \( K \) from 2.5 to 4.5 resulted in equivalence ratio differences on the order of \( \phi = \pm 0.001 \).

Since CO, CO₂, O₂, and NOₓ are measured on a dry² basis, their mole fractions can be converted from dry mole fractions, \( \tilde{x}^*_i \), to wet mole fractions, \( \tilde{x}_i \), through the following equation:

\[
\tilde{x}_i = (1 - \tilde{x}_{H_2O}) \tilde{x}^*_i \quad (D.9)
\]

Equation (D.9) can be combined with Equation (D.8) and subsequently rearranged to solve for \( \tilde{x}_{H_2} \):

\[
\tilde{x}_{H_2} = \frac{\tilde{x}^*_CO \tilde{x}_{H_2O}}{\tilde{x}^*_CO_2 K} \quad (D.10)
\]

Similarly, the number of moles in the products, \( n_P \), can be found by substitution of Equation (D.9) into Equation (D.2) and rearranging:

\[
n_P = \frac{n}{\tilde{x}_{CH_{b/a}} + (1 - \tilde{x}_{H_2O})(\tilde{x}^*_CO + \tilde{x}^*_CO_2)} \quad (D.11)
\]

Note that the \( a \cdot \tilde{x}_{C_aH_b} \) term has been replaced with \( \tilde{x}_{CH_{b/a}} \) since hydrocarbon emissions are measured as \( C_1 \).

¹ Water-gas shift reaction: \( CO_2 + H_2 \leftrightarrow CO + H_2O \)
² Water in the exhaust gas sample is condensed in a chiller at 4°C and removed. The remaining dessication is done with filtering techniques.
Rearranging Equation (D.3) to solve for $\tilde{x}_{H_2O}$ yields:

$$\tilde{x}_{H_2O} = \frac{1}{2} \left[ \frac{m}{n_P} - (b \tilde{x}_{C_aH_b} + 2 \tilde{x}_{H_2}) \right]$$  \hspace{1cm} (D.12)

Substitution of Equation (D.11) into Equation (D.12) leads to:

$$\tilde{x}_{H_2O} = \frac{1}{2} \left[ \frac{m}{n} (a \tilde{x}_{C_aH_b} + (1 - \tilde{x}_{H_2O})(\tilde{x}_{CO}^* + \tilde{x}_{CO_2}^*)) - (b \tilde{x}_{C_aH_b} + 2 \tilde{x}_{H_2}) \right]$$  \hspace{1cm} (D.13)

Furthermore, $\tilde{x}_{H_2}$ can be eliminated with substitution of Equation (D.10). The $\tilde{x}_{C_aH_b}$ terms cancel by making use of the assumption for $b/a$ from Equation (D.7), and with some arithmetic manipulation yields:

$$\tilde{x}_{H_2O} = \frac{m}{2n} \left( \frac{\tilde{x}_{CO}^* + \tilde{x}_{CO_2}^*}{1 + \frac{\tilde{x}_{CO}^*}{\tilde{x}_{CO_2}^* K} + \frac{m}{2n} (\tilde{x}_{CO}^* + \tilde{x}_{CO_2}^*)} \right)$$  \hspace{1cm} (D.14)

With all of the mole fractions known, Equation (D.4) can be rearranged to solve for $\phi$, with substitution of Equation (D.9) for the measured dry mole fractions:

$$\phi = \frac{2n_{O_2}}{n_P \tilde{x}_{H_2O} + n_P(1 - \tilde{x}_{H_2O})(\tilde{x}_{CO}^* + \tilde{x}_{CO_2}^* + \tilde{x}_{O_2}^* + \tilde{x}_{NO_x}^*)}$$  \hspace{1cm} (D.15)

Either Equation (D.5) or Equation (D.6) can be used to solve for $\tilde{x}_{N_2}$, if desired. These equations will give slightly different answers due to the assumptions that have been made.
Calculation of exhaust gas temperature (EGT)

The following EGT calculation method was taken from Stone [131].

The temperature in the cylinder before the exhaust valve opens can be found from the ideal gas law:

\[ T_1 = \frac{p_1 V_1}{m_1 R} \]  \hspace{1cm} (D.16)

Where \( p_1 \) is the measured cylinder pressure at the corresponding cylinder volume, \( V_1 \), \( R \) is the gas constant for the exhaust products, and \( m_1 \) is the mass in the cylinder, which can be approximated from the air and fuel mass flow rate measurements.

Assuming the gas expansion in the cylinder during the blow-down process is isentropic, the temperature in the cylinder after blow-down can be found Equation (D.17), where \( p_2 \) is the measured or assumed exhaust port pressure, and \( \gamma \) is the specific heat ratio of the gas.

\[ T_2 = T_1 \left( \frac{p_2}{p_1} \right)^{\gamma - 1/\gamma} \]  \hspace{1cm} (D.17)

The mass in the cylinder after blow-down can then be found either through the ideal gas law or using the ratios of temperature and pressure from state 1 to 2:

\[ \frac{m_2}{m_1} = \left( \frac{T_1}{T_2} \right) \left( \frac{p_2}{p_1} \right) \]  \hspace{1cm} (D.18)

The temperature in the exhaust port after the exhaust stroke (assuming well-mixed with blow-down gases) can be calculated from:

\[ T'_2 = \frac{T_1}{\gamma} + T_2 \left( 1 - \frac{1}{\gamma} \right) \frac{m_2}{m_1} \]  \hspace{1cm} (D.19)

\( T'_2 \) can be used to provide an estimate of the true exhaust gas temperature leaving the engine cylinder.
Exhaust Energy Balance

From Equation (6.2) in Section 6.7, the energy balance for the control volume of an engine (Figure 6.45) can be written as:

\[ \dot{m}_a h_a + \dot{m}_f h_f = \dot{Q} + \dot{W}_b + \dot{m}_b h_b + \dot{m}_e h_e \] (D.20)

The blow-by term (\( \dot{m}_b h_b \)) can be lumped into the mass air flow term (\( \dot{m}_a h_a \)), by assuming the blow-by gas consists only of air. Also, the mass flow rates can be replaced with molal flow rates by using the total molar specific enthalpy, denoted by \( \tilde{h} \). The resulting equation is therefore:

\[ \dot{n}_a \tilde{h}_a + \dot{n}_f \tilde{h}_f = \dot{Q} + \dot{W}_b + \dot{n}_e \tilde{h}_e \] (D.21)

The total enthalpy terms can be separated into their formation and sensible enthalpies, \( \tilde{h}_f^\circ \) and \( \tilde{h}_s^\circ \), respectively. The right-hand term of Equation D.21 can then be written as the sum each exhaust species, denoted by subscript \( i \), and the corresponding mole fraction, \( \tilde{x}_i \):

\[ \dot{n}_e \tilde{h}_e = \dot{n}_e \sum x_i (\tilde{h}_f^\circ + \tilde{h}_s^\circ)_i \] (D.22)

Equation (D.22) can be expanded to show each species in the exhaust:

\[ \dot{n}_e \tilde{h}_e = \dot{n}_e \left[ \tilde{x}_{CO_2} (\tilde{h}_{f,CO_2} + \tilde{h}_{CO_2}^\circ) + \tilde{x}_{CO} (\tilde{h}_{f,CO} + \tilde{h}_{CO}^\circ) + \tilde{x}_{O_2} (\tilde{h}_{f,O_2} + \tilde{h}_{O_2}^\circ) + \tilde{x}_{H_2} (\tilde{h}_{f,H_2} + \tilde{h}_{H_2}^\circ) + \tilde{x}_{CH_4} (\tilde{h}_{f,CH_4} + \tilde{h}_{CH_4}^\circ) + \tilde{x}_{NO} (\tilde{h}_{f,NO} + \tilde{h}_{NO}^\circ) + \tilde{x}_{N_2} (\tilde{h}_{f,N_2} + \tilde{h}_{N_2}^\circ) \right] \] (D.23)

By regrouping the sensible enthalpy terms and removing \( \tilde{h}_{f,O_2}^\circ \), \( \tilde{h}_{f,H_2}^\circ \), and \( \tilde{h}_{f,N_2}^\circ \), which all have values of zero according to the JANAF tables [104], and noting that \( \tilde{x}_{NO} \approx 0 \); Equation (D.23) becomes:

\[ \dot{n}_e \tilde{h}_e = \dot{n}_e \left[ \sum (\tilde{x}_i \tilde{h}_i^\circ) + \tilde{x}_{CO_2} \tilde{h}_{f,CO_2}^\circ + \tilde{x}_{CO} \tilde{h}_{f,CO}^\circ + \tilde{x}_{H_2O} \tilde{h}_{f,H_2O}^\circ + \tilde{x}_{CH_4} \tilde{h}_{f,CH_4}^\circ \right] \] (D.24)
For complete combustion of methane-air \((\text{CH}_4 + \frac{2}{\phi} (\text{O}_2 + 3.76\text{N}_2) \longrightarrow \text{CO}_2 + 2\text{H}_2\text{O} + 7.52\text{N}_2)\), the lower heating value is the difference between the product and reactant enthalpies:

\[
Q_{LHV,\text{CH}_4} = - (\Delta H^o) = -(H_P^o - H_R^o) \\
= \left( x_{\text{CH}_4} \tilde{h}^o_{f,\text{CH}_4} + x_{\text{O}_2} \tilde{h}^o_{f,\text{O}_2} + x_{\text{N}_2} \tilde{h}^o_{f,\text{N}_2} \right) \\
- \left( x_{\text{CO}_2} \tilde{h}^o_{f,\text{CO}_2} + x_{\text{H}_2\text{O}} \tilde{h}^o_{f,\text{H}_2\text{O}} + x_{\text{N}_2} \tilde{h}^o_{f,\text{N}_2} \right) \\
= \tilde{h}^o_{f,\text{CH}_4} - \tilde{h}^o_{f,\text{CO}_2} - 2\tilde{h}^o_{f,\text{H}_2\text{O}} \tag{D.25}
\]

Similarly, for the complete combustion of CO-air \((\text{CO} + \frac{1}{2} (\text{O}_2 + 3.76\text{N}_2) \longrightarrow \text{CO}_2 + 7.52\text{N}_2)\), the lower heating value is:

\[
Q_{LHV,\text{CO}} = - (\Delta H^o) = -(H_P^o - H_R^o) \\
= \left( x_{\text{CO}} \tilde{h}^o_{f,\text{CO}} + x_{\text{O}_2} \tilde{h}^o_{f,\text{O}_2} + x_{\text{N}_2} \tilde{h}^o_{f,\text{N}_2} \right) \\
- \left( x_{\text{CO}_2} \tilde{h}^o_{f,\text{CO}_2} + x_{\text{N}_2} \tilde{h}^o_{f,\text{N}_2} \right) \\
= \tilde{h}^o_{f,\text{CO}} - \tilde{h}^o_{f,\text{CO}_2} \tag{D.26}
\]

Rearranging Equations (D.25) and (D.26) in terms of \(\tilde{h}^o_{f,\text{CH}_4}\) and \(\tilde{h}^o_{f,\text{CO}}\) on the left-hand side, respectively, followed by substitution into Equation (D.24), results in:

\[
\dot{n}_e \tilde{h}_e = \dot{n}_e \left[ \sum \left( x_i \tilde{h}^o_i \right) + \tilde{x}_{\text{CH}_4} Q_{LHV,\text{CH}_4} + \tilde{x}_{\text{CO}} Q_{LHV,\text{CO}} + \tilde{h}^o_{f,\text{CO}_2} \left( \tilde{x}_{\text{CO}} + \tilde{x}_{\text{CO}_2} + \tilde{x}_{\text{CH}_4} \right) + \tilde{h}^o_{f,\text{H}_2\text{O}} (2\tilde{x}_{\text{CH}_4} + \tilde{x}_{\text{H}_2\text{O}}) \right] \tag{D.27}
\]

For the incomplete methane-air reaction, the chemical equation is below, where \(n_i\) is the number of moles for each species:

\[
n_{\text{CH}_4} \text{CH}_4 + \frac{2}{\phi} n_{\text{CH}_4} (\text{O}_2 + 3.76\text{N}_2) \longrightarrow n_{\text{CO}_2} \text{CO}_2 + n_{\text{CO}} \text{CO} + n_{\text{O}_2} \text{O}_2 + n_{\text{H}_2\text{O}} \text{H}_2\text{O} + n_{\text{CH}_4} \text{CH}_4 + n_{\text{N}_2} \text{N}_2
\]

Atomic balances for carbon and hydrogen yield Equations (D.28) and (D.29), respectively. \(n_{\text{CH}_4}^*\) has been used to distinguish \(\text{CH}_4\) as a reactant from \(\text{CH}_4\) in the exhaust products.

\[
n_{\text{CH}_4}^* = n_{\text{CO}_2} + n_{\text{CO}} + n_{\text{CH}_4} \tag{D.28}
\]

\[
2n_{\text{CH}_4}^* = n_{\text{H}_2\text{O}} + 2n_{\text{CH}_4} \tag{D.29}
\]
By definition $n_i = \bar{x}_i \cdot n_P$, where $n_P$ is the number of moles in the products. Substitution and rearrangement yields:

$$\frac{n_{\text{CH}_4}^*}{n_P} = \bar{x}_{\text{CO}_2} + \bar{x}_{\text{CO}} + \bar{x}_{\text{CH}_4}$$  \hspace{1cm} (D.30)$$

$$2 \frac{n_{\text{CH}_4}^*}{n_P} = \bar{x}_{\text{H}_2\text{O}} + 2 \bar{x}_{\text{CH}_4}$$  \hspace{1cm} (D.31)$$

By noting that $\frac{n_{\text{CH}_4}^*}{n_P} = \frac{\dot{n}_{\text{CH}_4}}{\dot{n}_P}$ and that $\dot{n}_P = \dot{n}_e$, Equations (D.30) and (D.31) can be substituted into Equation (D.27) to produce:

$$\dot{n}_e \bar{h}_e = \dot{n}_e \left[ \sum \left( \bar{x}_i \bar{h}_i^\circ \right) + \bar{x}_{\text{CH}_4} Q_{\text{LHV,CH}_4} + \bar{x}_{\text{CO}} Q_{\text{LHV,CO}} + \bar{h}_{f,\text{CO}_2} \left( \frac{\dot{n}_{\text{CH}_4}^*}{\dot{n}_e} \right) + \bar{h}_{f,\text{H}_2\text{O}} \left( 2 \frac{\dot{n}_{\text{CH}_4}^*}{\dot{n}_e} \right) \right]$$  \hspace{1cm} (D.32)$$

Since both the fuel and air enter the engine at approximately 298 K, the sensible enthalpies of both gases are zero (for tables with $T_{\text{ref}} = 298$ K). In addition, the formation enthalpy of air is also zero. Thus, for CH$_4$ as the fuel, Equation (D.21) can be written as:

$$\dot{n}_{\text{CH}_4} \bar{h}_{f,\text{CH}_4} = \dot{Q} + \dot{W}_b + \dot{n}_e \bar{h}_e$$  \hspace{1cm} (D.33)$$

Substituting Equation (D.25) into Equation (D.33) yields:

$$\dot{n}_{\text{CH}_4} \left( Q_{\text{LHV,CH}_4} + \bar{h}_{f,\text{CO}_2} + 2 \bar{h}_{f,\text{H}_2\text{O}} \right) = \dot{Q} + \dot{W}_b + \dot{n}_e \bar{h}_e$$  \hspace{1cm} (D.34)$$

Substituting Equation (D.32) into Equation (D.34) provides the final equation, which can be rearranged to solve for the only unknown term, $\dot{Q}$:

$$\dot{n}_{\text{CH}_4} Q_{\text{LHV,CH}_4} = \dot{Q} + \dot{W}_b + \dot{n}_e \left[ \sum \left( \bar{x}_i \bar{h}_i^\circ \right) + \bar{x}_{\text{CH}_4} Q_{\text{LHV,CH}_4} + \bar{x}_{\text{CO}} Q_{\text{LHV,CO}} \right]$$  \hspace{1cm} (D.35)$$

where:

- $(\dot{n}_{\text{CH}_4} Q_{\text{LHV,CH}_4})$ is the input fuel energy
- $\dot{Q}$ is the total heat loss
- $\dot{W}_b$ is the engine brake power
- $\dot{n}_e \sum \left( \bar{x}_i \bar{h}_i^\circ \right)$ is the sensible exhaust energy
- $(\bar{x}_{\text{CH}_4} Q_{\text{LHV,CH}_4} + \bar{x}_{\text{CO}} Q_{\text{LHV,CO}})$ is the unburned fuel energy in the exhaust
Appendix E

LabVIEW™ Programming

DAQ and Low-Level Control VI Block Diagram

The complete block diagram code for DAQ and low-level control is shown in Figure E.1. Due to its physical size, the code has been sectioned according to Figure E.2, with the resulting sections enhanced in Figures E.3 to E.7. The function of each section is as follows:

(A) Temperature and engine speed monitoring along with low speed data logging
(B) Analog output control for dynamometer speed, fan operation, and throttle position
(C) High speed acquisition of encoder position and analog channel data
(D) Data scaling and processing
(E) Data display and logging to disk

The corresponding front panel is shown in Figure E.8.

Engine Control VI

The engine control VIs were adapted from the “DI Calibrator” VI that came with the fuel injection control module. The configuration of the FPGA target VI block diagram is shown in Figure E.9. The corresponding real time VI block diagram is shown in two parts: the top half in Figure E.10 and the bottom half in Figure E.11. The RT VI front panel is shown in Figure E.12.
Figure E.1: Block diagram of DAQ and low-level control VI.
Figure E.2: Breakdown of VI segments shown in Figures E.3 to E.7.
Figure E.3: Block diagram of section A.
Figure E.4: Block diagram of section B.
Figure E.5: Block diagram of section C.
Figure E.6: Block diagram of section D.
Figure E.7: Block diagram of section E.
Figure E.8: Screenshot of LabVIEW™ VI used for engine monitoring, DAQ, and low-level control.
Figure E.9: Block diagram of engine control VI on the FPGA target.
Figure E.10: Block diagram of engine control VI on the RT target. (Top; page 1/2)
Figure E.11: Block diagram of engine control VI on the RT target. (Bottom; page 2/2)
Figure E.12: Screenshot of LabVIEW™ VI used for engine fuel and spark control.
Appendix F

Dynamometer Calibration

To determine and continually verify the calibration of the dynamometer’s rotary torque transducer, a dead-weight calibration arm was fabricated. The arm, shown in Figure F.1, was manufactured from aluminum plate and holes were drilled along its length to reduce off-axis gravitational errors. A bubble vial attached to the top provides a means of ensuring the arm is level and the dynamometer drive shaft is neutrally balanced before adding calibration weight.

![Figure F.1: Dead-weight dynamometer calibration arm.](image)

To calibrate the torque transducer, the arm is first attached to the end of the dyno shaft in a horizontally level position, as shown in Figure F.2. A rigid piece of steel is inserted into the flexible coupler that is closest to the AC motor and braced against the mounting plate, preventing the drive-line from turning. To ensure accuracy, the engine drive belt must be loose, or preferably removed from the drive pulley, and the transducer/signal conditioner warmed up for a minimum of 1 hour.

The remaining calibration procedure is as follows:
Step 1 Set torque reading to zero
Step 2 Add full-scale weights (of known value) to calibration arm
Step 3 Adjust conditioner gain until output reading corresponds with applied torque
Step 4 Remove weights and check/re-adjust zero
Step 5 Repeat steps 1–4 until no further adjustment is required
Step 6 Check linearity at random fractional loads

Figure F.2: Photograph of dynamometer being calibrated with dead-weight torque arm.
**Vita Auctoris**

**Name:** Iain A.S. Cameron

**Year of birth:** 1987

**Place of birth:** Fredericton, New Brunswick, Canada

**Secondary education:**
Fredericton High School
Fredericton, New Brunswick, Canada (2001–2005)

**Post-secondary education:**
Bachelor of Applied Science, Honours Mechanical Engineering – Automotive Option, with Great Distinction
University of Windsor
Windsor, Ontario, Canada (2005–2009)

**Post-graduate education:**
Doctor of Philosophy
Mechanical Engineering, University of Windsor
Windsor, Ontario, Canada (2009–2015)