

# Measurement of Loss Pathways in Small, Two-Stroke Internal-Combustion Engines

Joseph K. Ausserer and Marc D. Polanka  
Air Force Institute of Technology

Jacob A. Baranski  
Innovative Scientific Solutions, Inc.

Keith D. Grinstead  
University of Dayton Research Institute

Paul J. Litke  
Air Force Research Laboratory

## ABSTRACT

The rapid expansion of the market for remotely piloted aircraft (RPA) includes a particular interest in 10-25 kg vehicles for monitoring, surveillance, and reconnaissance. Power-plant options for these aircraft are often 10-100 cm<sup>3</sup> internal combustion engines. Both power and fuel conversion efficiency decrease with increasing rapidity in the aforementioned size range. Fuel conversion efficiency decreases from ~30% for conventional-scale engines (>100 cm<sup>3</sup> displacement) to <5% for micro glow-fuel engines (<10 cm<sup>3</sup> displacement), while brake mean effective pressure decreases from >10 bar (>100 cm<sup>3</sup>) to <4 bar (<10 cm<sup>3</sup>). Based on research documented in the literature, the losses responsible for the increase in the rate of decreasing performance cannot be clearly defined. Energy balances consisting of five pathways were experimentally determined on two engines that are representative of Group-2 RPA propulsion systems and compared to those in the literature for larger and smaller engines. The five pathways were brake power, cooling load, sensible exhaust enthalpy, incomplete combustion, and short-circuiting. The results show that incomplete combustion and (in the case of two-stroke cycle engines) short-circuiting are responsible for the decrease in fuel conversion efficiency as engine displacement decreases from greater than 100 cm<sup>3</sup> to the 10 cm<sup>3</sup> to 100 cm<sup>3</sup> size range. The results show that the percentage of heat-transfer losses in 10-100 cm<sup>3</sup> engines was not appreciably higher than those in conventional engines and that these engines have thermal efficiencies (fraction of shaft work to released energy) of ~35%, which are comparable to those of conventional-scale spark-ignition and compression-ignition engines. As engine displacement decreases below 10 cm<sup>3</sup>, heat-transfer losses increase, compounding with already large short-circuiting and incomplete combustion losses. The result is a rapid decrease in both power and fuel conversion efficiency. In addition to these scaling results, the high short-circuiting noted in the engines studied (40-50% at some wide-open-throttle conditions) underscores the benefits of employing short-circuiting management techniques such as throttle-body injection and direct injection, exhaust and intake tuning, and port optimization. The cost-benefit tradeoff of these techniques in the context of commercial-off-the-shelf (COTS)-based design is discussed.

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## INTRODUCTION

The use of small remotely piloted aircraft (RPA) is expanding rapidly in the United States. Between the years of 2015 and 2025, the industry is expected to grow by nearly 10,000 jobs and \$8.2 billion per year, providing platforms for surveillance, monitoring, and reconnaissance for both military and civilian organizations [1]. For Group 2 RPAs (~10-25 kg), two types of propulsion systems are in competition: electric and hydrocarbon-fuel internal combustion engines (ICEs). In this size range, hydrocarbon-fueled RPAs offer superior range and endurance because of the high energy density of

hydrocarbons relative to batteries [2]. For instance, the best rechargeable batteries available on the market--lithium sulfur--offer a specific energy of 600 W-hr/kg, while the average hydrocarbon offers ~13,000 W-hr/kg. Thus, the breakeven efficiency is 4-5% for small hydrocarbon engines. Theoretically, this gives hydrocarbon-fueled vehicles a 20:1 range or endurance margin over comparable electric propulsion systems. In practice, the margin is smaller. Compared to electric power plants, which have efficiencies from 77-88% [3], conventional-scale ICEs typically have fuel-to-shaft efficiencies of ~30%. The 10-100 cm<sup>3</sup> displacement engines used in Group 2 RPAs

have lower efficiencies of 4-15% [4, 5]. For an 85% efficient electric motor, a 5% ICE efficiency is the breakeven point [6], where both motor and engine are assumed to have the same specific power.

Figure 1 shows manufacturer data for a variety of engine types adapted from Menon and Cadou [5]. Overall, this Figure 1 shows fuel conversion efficiency decreasing with engine size. That decrease is driven by different phenomena in different size ranges. For example, large ship and train engines tend to be blower-scavenged, turbocharged, direct-injection, compression-ignition (CI) engines that have higher theoretical efficiencies (due to higher compression ratios) than spark-ignition (SI) highway engines. A study performed by Menon and Cadou of two-stroke, micro glow-fuel engines with <10 cm<sup>3</sup> displacement suggests that the dominant loss pathways are reversed in micro glow-fuel engines relative to larger automotive-scale engines [7]. They report that losses in micro glow-fuel engines are dominated by incomplete combustion and heat transfer from the cylinder, while losses in larger engines are dominated by sensible exhaust enthalpy, followed by heat transfer from the cylinder [7]. Therefore, it is unclear in the literature which loss pathways drive the fuel conversion efficiency of 10-100 cm<sup>3</sup> displacement engines, which are applicable to Group 2 RPAs.

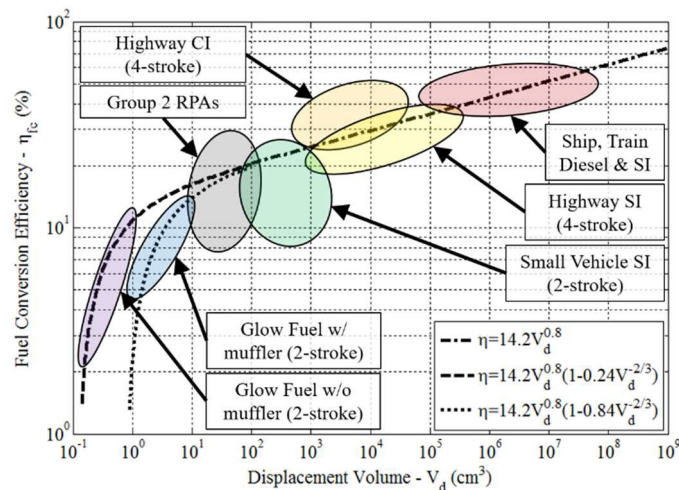


Figure 1. Manufacturer data grouped by engine type, showing the relationship between engine size and fuel conversion efficiency. Based on data compiled and fits created by Menon and Cadou [5].

Understanding the losses responsible for the observed decrease in 10-100 cm<sup>3</sup> displacement engines will provide insight into the loss mechanisms that should be targeted to improve engine efficiency and, in turn, aircraft range and endurance. The energy-loss pathways in an engine can be managed using different strategy. For example, short-circuiting could be managed through direct injection (DI), tuning the exhaust, or redesigning the ports, while thermal losses might be managed through insulation of (or providing additional cooling to) the engine. Thus, an understanding of those mechanisms and their relationship to engine size is key to prioritizing management of those losses in the unmanned aircraft industry. Furthermore, an understanding of the loss mechanisms provides a starting point for altering these small, commercial-off-the-shelf (COTS) engines that are designed to operate on high anti-knock-index (AKI) fuels (gasoline) to operate on lower AKI fuels (Diesel, JP-8, Jet-A).

Operation of small RPAs on low-AKI fuels such as Diesel, JP-8, and Jet-A is desirable to comply with DoD Directive 4140.25 [8], which prescribes the use of a single battlefield fuel.

## OBJECTIVES

While energy-balance data for engines larger than 100 cm<sup>3</sup> are readily available [9, 10] and limited data sets on smaller micro glow-fuel engines can be found [5, 7], very little energy-balance data are available for 10-100 cm<sup>3</sup> engines. Therefore, to address the issues described in the Introduction, the objective of this work was the experimental quantification of energy balances for two engines that are representative of Group 2 RPA engines and the identification of the dominant loss pathways. Based on the interest in Group 2 RPAs, the scope of this effort was limited to two-stroke spark-ignition (SI) engines between 10 and 100 cm<sup>3</sup> displacement. For the purpose of this study, an energy balance consisted of the following pathways: brake power, cooling load (and friction), sensible exhaust enthalpy, incomplete combustion, and short-circuiting. Since it is unclear from the literature which loss pathways drive the efficiency in the engines in question, identification of the dominant pathways will provide direction for future work in improving engine efficiency as well as aircraft range and performance.

## BACKGROUND AND LITERATURE REVIEW

The energy balance in this effort consisted of five energy pathways: brake power, cooling load, sensible exhaust enthalpy, incomplete combustion, and short-circuiting. Because of the measurement methodology, friction losses were included in the cooling load and sensible exhaust enthalpy. Numerous factors influence the aforementioned loss pathways, some of which correlate to engine size and some of which are relatively independent of engine size. For example, for nominally square engines, the surface-area-to-volume ratio scales with the inverse of bore, while the compression ratio is driven primarily by ignition type [SI, CI, HCCI (homogenous-charge-compression ignition)] [5]. Therefore, any discussion of loss pathways in 10-100 cm<sup>3</sup> displacement engines and how they differ from those of larger and smaller engines must consider both types of factors. This section discusses each of the energy pathways measured in this effort and the key factors influencing those pathways. The order of the discussion will mirror the order in the Results and Analysis Section: short-circuiting (scavenging), incomplete combustion, exhaust enthalpy, cooling load, and brake power.

For aeronautical applications two-stroke cycle engines are preferable to four-stroke cycle engines because they have one power stroke per revolution instead of one power stroke per every other revolution. Therefore, two-stroke cycle engines offer higher specific power, which saves weight on the aircraft. To create one power stroke per revolution, the intake and exhaust processes have significant overlap, which leads to the first loss pathway measured in this study--short-circuiting. Short-circuiting is the loss of fresh charge to the exhaust during the gas-exchange process. The degree of short-circuiting is influenced by a number of engine design factors, including scavenging layout, port design [11], fuel-injection strategy, and tuning of the exhaust and/or intake runner. Generally, a tradeoff exists between cost and performance when optimizing these design factors.

to mitigate short-circuiting. For example, rigorous computational fluid dynamics (CFD) has been used to reduce short-circuiting by 30% in a 70 cm<sup>3</sup> displacement two-stroke engine [12]; however, the research and retooling costs were amortized over 60,000 units per month. Similarly, short-circuiting in large two-stroke engines is often managed using DI. While DI systems are feasible on Group 2 RPA sized engines (and smaller), they are often impractical from a cost and weight standpoint. For example, the cost of the aftermarket throttle-body injection (TBI) system installed on the engines studied here was approximately three times that of the individual engines.

Short-circuiting is typically quantified and reported using the trapping efficiency, delivery ratio, and charging efficiency. The trapping efficiency is the percentage of fresh charge delivered to the engine that remains in the cylinder when the ports close, and unity less the trapping efficiency is short-circuiting. The delivery ratio is the amount of fresh charge delivered to the engine, divided by the cylinder capacity; the charging efficiency describes how well the fresh charge replaces the burnt charge in the cylinder. For large engines many short-circuiting models are available [11, 13], ranging from simple one- and two-zone models [14] to multi-zone models [15] and complex CFD models [16]. Often, the perfect displacement (two-zone) and perfect mixing (single-zone) models are considered bounding cases for scavenging performance. Credited originally to Hopkinson [14], the perfect-displacement model assumes that each unit of fresh charge perfectly replaces an equal volume of burnt charge and is the optimum case for scavenging. The perfect-mixing model assumes that the fresh charge and burnt charge mix instantaneously in a homogenous cylinder. It is possible to have short-circuiting greater than that of the perfect-mixing model; however, modern engines typically outperform this model; therefore, it is commonly suggested as a lower bound [11, 17].

The second loss pathway considered in this effort was incomplete combustion. Distinct from short-circuiting, incomplete combustion is defined as the fraction of the energy in the trapped fuel that is not released during combustion. Incomplete combustion may be described using the combustion efficiency, which is the amount of energy released during combustion, divided by the chemical energy contained in the fuel. A number of factors influence the degree of incomplete combustion, including ignition type, equivalence ratio, residence time (and engine speed), and thermal losses and quenching at the cylinder walls and in crevices. The latter two (residence time and quenching) are at least partially influenced by engine size. In CI engines the timing of fuel injection is used to control combustion phasing. Compared to SI engines where the cylinder contents is nominally homogenous (except in the case of stratified-charge strategies), CI engines burn globally lean but locally rich near the fuel jet(s), leading to a higher propensity to soot than SI engines. Similarly, running an SI engine at a rich condition will lead to incomplete-combustion products such as H<sub>2</sub>, CO, and unburned hydrocarbons (UHCs).

For nominally stoichiometric conditions, incomplete combustion can be caused by a number of factors, including insufficient residence time and quenching of the flame. First, consider the former. As engine size decreases, engine speeds typically increase (speeds of 20,000 rpm in engines with displacement volumes <10 cm<sup>3</sup> are common),

which decreases the residence time. It has often been hypothesized that the decreased time causes incomplete combustion [7, 18]. Comparison of estimated burn times, residence times, and practical engine studies indicates that this may not be the case. Researchers have suggested minimum burn times of 1 ms for hydrocarbons at 1 atm and much shorter times at the pressures and temperatures found in-cylinder [2]. Even at the maximum speed of 8000 rpm for the engines considered here, the residence time is still nearly 2.5 ms. Further investigation indicated that the OS 30X engine with a 4.9 cm<sup>3</sup> combustion volume had sufficient time for complete combustion of the fuel at speeds >10,000 rpm [18]. These analyses assume a well-mixed near stoichiometric charge. However, inhomogeneity in the cylinder contents coupled with short residence times could contribute to incomplete combustion.

The other common source of incomplete combustion is quenching at the cylinder walls and in crevices, which increases relative to the combustion volume with decreasing engine size. Sher and Sher developed a model for predicting the theoretical limit of miniaturizing an IC engine [19]. The model suggests a minimum attainable displacement between 0.3 and 0.4 cm<sup>3</sup> for a HCCI engine, driven primarily by thermal losses through the wall and quenching losses at the wall [19, 20, 21]. One mechanism to offset quenching driven by thermal losses from increased surface-area-to-volume ratio is increased engine speed and the burning of more fuel per unit time, which effectively maintains the ratio of combusted mass to surface area, despite decreased combustion volume. When structural limits set the maximum engine speed, this option becomes limited.

The last three energy pathways considered in this effort were heat transfer from the cylinder, exhaust-sensible enthalpy, and brake power. These pathways quantify how well an engine converts the heat released during combustion into useful work and may be described by the thermal efficiency. Factors that influence the thermal efficiency include forced induction, compression ratio, and cooling of and heat transfer from the engine. Some of the factors that influence heat-transfer scale with engine size. Both forced induction and higher compression ratios improve the availability of the energy released during combustion, leading to higher theoretical (and practical) thermal efficiencies. In Figure 1 the difference between Small Vehicle SI and Highway CI engines is largely the result of differences in compression ratio (and, thus, maximum theoretical efficiency) between SI and CI engines; the large Ship and Train type engines benefit from a high degree of forced induction. For compression ratio and forced induction, scaling driven by design is only implicitly tied to engine size. For example, forced-induction and CI engines are uncommon on small aircraft--not because they cannot be used, but because they are typically prohibitively heavy [22].

Heat rejection from the cylinder also decreases the thermal efficiency, and heat may be lost to the exhaust or through the cylinder walls, head, and piston. From the Second Law of Thermodynamics, some amount of heat rejection is necessary since the thermal energy released during combustion has an availability that is (substantially) less than unity. Ideally, this rejection would occur at the end of the expansion stroke after the combustion products have been expanded to the exhaust pressure. Practical designs fall short of this goal, and two-stroke cycle engines are typically under-expanded. It follows that

port design, specifically exhaust-port timing, plays a critical role in heat rejection to the exhaust. While heat rejection to the exhaust is a thermodynamic necessity, heat rejection from the cylinder head is not. Reducing the heat lost through the cylinder walls and head, however, will not lead to a one-to-one increase in engine power since the thermal energy rejected through the head has a thermodynamic availability that is less than unity. Instead, a modification such as insulating the head will shift some of the thermal loading from heat transfer through the head to the sensible enthalpy in the exhaust.

It is not clear in the literature how much influence engine size has on heat transfer. A study on micro glow-fuel engines showed that for engines with displacement volumes  $<1 \text{ cm}^3$ , heat transfer dissipated over one-half of the energy released during combustion [5, 7] (compared to conventional automotive-type engines where 20-30% of the released energy is dissipated to the coolant. Menon and Cadou [7] identify a number of studies suggesting that the increase in thermal losses is driven by the increase in surface-area-to-volume ratio with decreasing engine size. However, data collected by Taylor on three geometrically similar engines ranging from 241-3334  $\text{cm}^3$  displacement indicated that while larger engines were more difficult to cool, the heat transfer had little impact on efficiency [23]. Studies on the size limits of ICEs complicate the situation and suggest that the smallest possible SI engine is limited by quenching of the flame caused by heat transfer from the cylinder [19, 20, 21]. In fact, Menon and Cadou [5] predicted the minimum practical engine size, based on the rate of heat transfer from the cylinder exceeding the rate of heat generation due to combustion, and found a minimum practical engine size of 0.1  $\text{cm}^3$ , comparable to Sher's [19] result found from incomplete combustion.

Clearly, the loss pathways driving the efficiency behavior observed for 10-100  $\text{cm}^3$  displacement engines in Figure 1 are influenced by a variety of factors, some of which are, in turn, driven by engine size. Without energy balances for this size range, it is difficult to discern which loss pathways dominate engine performance and which factors have the most influence on the pathways. To provide energy balances for this size range and address these issues, energy balances were experimentally collected on two engines. The selection of those engines and the methodology for collecting the energy balances are described in the subsequent section.

## METHODOLOGY

The methodology is discussed in two subsections. The first describes the selection of the engines. The second presents the energy balance considered in this work and describes the test setup and techniques used to measure the loss pathways.

### Engine Selection

To study the loss pathways in 10-100  $\text{cm}^3$  displacement, two-stroke SI engines, two 3W Modellmotoren engines in that size range, 3W-55i and 3W-28i, were selected that were as geometrically similar as commercially possible, with surface-area-to-volume ratios of 1.46 and 1.83, respectively. The important engine characteristics are summarized in Table 1. Similar to many two-stroke engines in the 10-100  $\text{cm}^3$  displacement range, these two engines were spark-

ignited, naturally aspirated, and constructed primarily from aluminum. Both engines had a geometric compression ratio of 10. The port timings were close, but not identical. The port areas and port open times are given in Table 1. The engines had similar port layouts and were loop-scavenged. Each engine had two scavenge ports, centered 90° to either side of the single exhaust port. Opposite the exhaust port was a boost port. The 3W-28i engine had a single wide boost port, while the 3W-55i engine had two narrower boost ports located side by side. The ports were nominally rectangular, and the width and heights are also included in Table 1. To improve run-to-run repeatability and control over the equivalence ratio, the stock carburetors on both engines were replaced with Currawong Engineering throttle bodies and TBI systems (details in [24]).

Table 1. Design characteristics of 3W Modellmotoren engines selected for loss study.

| Engine                                                  | 3W-28i                                                                                | 3W-55i                |
|---------------------------------------------------------|---------------------------------------------------------------------------------------|-----------------------|
| Displacement (cc)                                       | 28                                                                                    | 55                    |
| Power (kW)                                              | 2.5                                                                                   | 3.82                  |
| Mass (kg)                                               | 1.21                                                                                  | 1.94                  |
| Power Density (kW/kg)                                   | 2.07                                                                                  | 1.97                  |
| Bore (cm)                                               | 3.61                                                                                  | 4.50                  |
| Stroke (cm)                                             | 2.84                                                                                  | 3.51                  |
| Compression ratio                                       | 10:1                                                                                  | 10:1                  |
| Cylinder surface-area-to-volume ratio (1/cm)            | 1.81                                                                                  | 1.46                  |
| Trapped compression ratio                               | 7.2:1                                                                                 | 7.1:1                 |
| Surface-area-to-swept-volume ratio ( $\text{cm}^{-1}$ ) | 1.83                                                                                  | 1.46                  |
| Exhaust port open (°ATC)                                | 106                                                                                   | 103                   |
| Scavenge port open (°ATC)                               | 125                                                                                   | 118                   |
| Boost port open (°ATC)                                  | 122                                                                                   | 110                   |
| Exhaust-port area ( $\text{cm}^2$ )                     | 1.9                                                                                   | 2.1                   |
| Scavenge-port area ( $\text{cm}^2$ )                    | 1.8                                                                                   | 2.4                   |
| Boost-port area ( $\text{cm}^2$ )                       | 1.3                                                                                   | 1.1                   |
| Exhaust port:<br>width × height (mm)                    | 1 port<br>8.5 × 22.3                                                                  | 1 port<br>9.2 × 23.3  |
| Scavenge ports:<br>width × height (mm)                  | 2 ports<br>5.6 × 16.6                                                                 | 2 ports<br>5.7 × 20.5 |
| Boost port:<br>width × height (mm)                      | 1 port<br>7.6 × 17.4                                                                  | 2 ports<br>7.2 × 7.7  |
| Engine Photograph, Ref. [25]                            |  |                       |

### Energy Balances and Small Engine Research Bench

The Small Engine Research Bench (SERB) was constructed specifically to measure the loss mechanisms and performance of 10-100  $\text{cm}^3$  engines. The description herein will focus on the aspects of the bench required to measure the energy pathways in this work; more details on the bench are available in Refs. [26, 27, 28]. A schematic of the research bench is shown in Figure 2. As mentioned previously, five energy pathways were considered in this effort: brake power, cooling load, sensible exhaust enthalpy, incomplete combustion, and short-circuiting. Because of the method used to measure cooling load and exhaust enthalpy, the friction and pumping

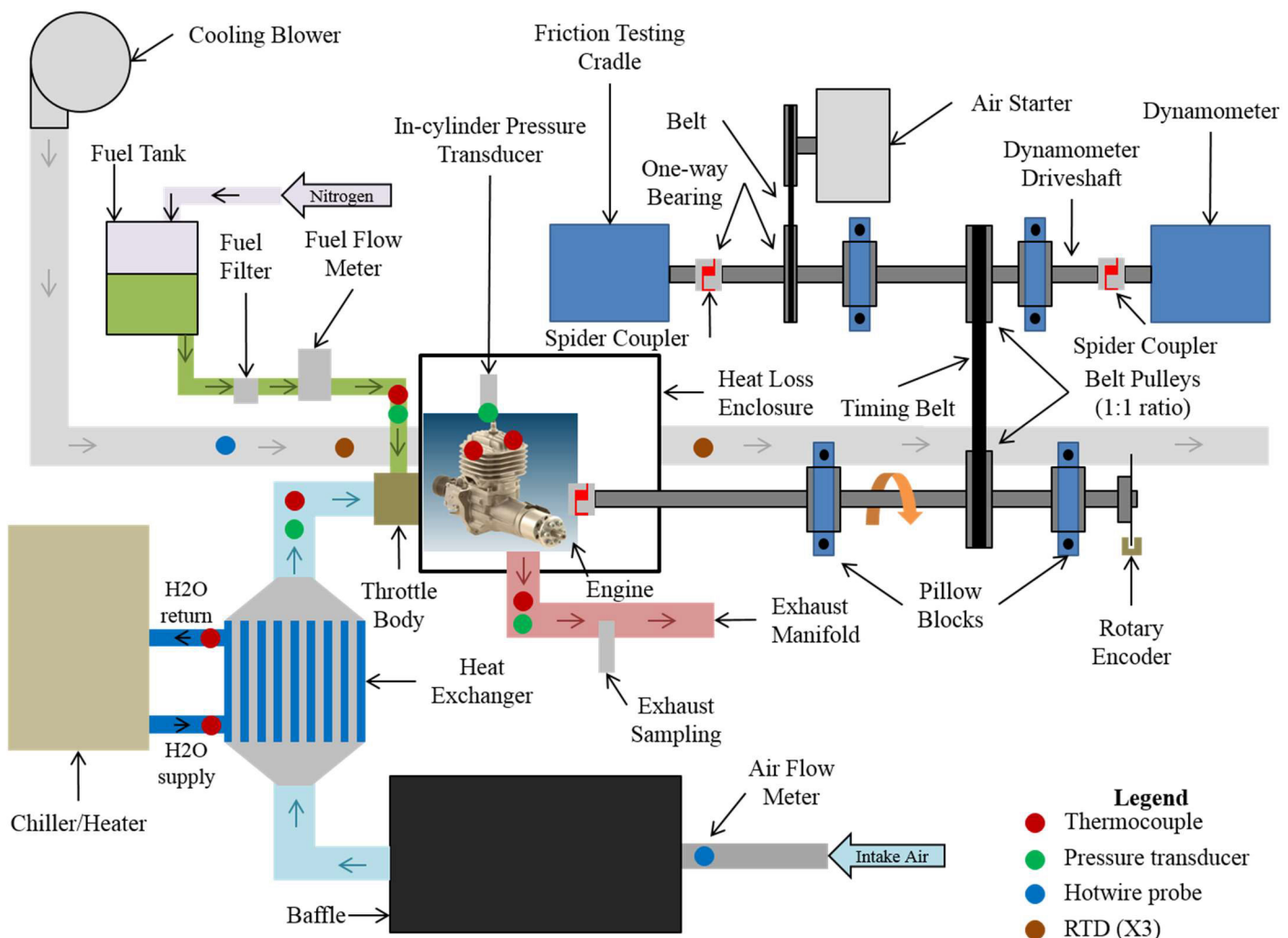


Figure 2. SERB schematic showing fluid flow pathways and measurement equipment.

losses in the engine were combined into the cooling-load and exhaust-enthalpy terms. The remainder of this section will be divided into four subsections that describe the engine control system, measurements of fuel and air delivered to the engine, brake power, cooling load, and losses in the exhaust.

### Engine Control System

The engine control system consisted of a Low Speed Data Acquisition and Control System (LSDAQ) and an AVL Indismart combustion analyzer. The LSDAQ was used to control fuel flow and engine speed as well as the engine control unit (ECU). Developed in-house, the ECU allowed on-the-fly control of spark timing, fuel-injection duration and timing, and the throttle servo [29]. Spark timing was set to maintain the CA50 mass fraction burned angle at  $8 \pm 0.5^\circ$  after top-dead center (aTDC), which was within  $2^\circ$  crank angle (CA) of maximum-brake-torque (MBT) timing for both engines. The mass fraction burned profile was calculated in real time using the AVL Indismart combustion analyzer and an in-cylinder pressure transducer, which will be described next. Based on a study of combustion phasing documented in Ref. [30], deviation of  $<2^\circ$  CA from MBT timing resulted in a  $<2\%$  decrease in torque relative to MBT.

The AVL Indismart combustion analyzer measured in-cylinder pressure, indicated mean effective pressure (IMEP), and CA50 mass fraction burned angle and monitored spark and injection timing. Engine crank angle was measured using a US Digital brand E6 optical encoder with 1800 counts per revolution connected to the AVL system. A Kistler 6113B measuring spark plug was installed in place of the stock spark plug to measure in-cylinder pressure. The measuring spark plug included a piezoelectric pressure transducer, which had a linearity of  $\pm 0.5\%$  of full scale and a sensitivity of  $\pm 1\%$  at  $200 \pm 50^\circ\text{C}$ . Ausserer *et al.* [27] documented a trade study of flush-mount and measuring-spark-plug piezoelectric pressure transducers in a 3W-55i engine that was identical to the one used here. The results showed that the measuring spark plug and stock spark plug were interchangeable and that by modifying the cylinder head for a flush-mount transducer, one risked altering engine performance, which is why the measuring spark plug was preferred for this effort. Additional details are available in Ausserer *et al.* [27].

### Air and Fuel Flow Measurements

Intake air flow was measured using an Omega Engineering FMA900 series hotwire anemometer with a range of 0.5 to 8 m/s and an accuracy of  $\pm 2\%$  of full scale. For the manufacturer's documentation,