PERFORMANCE MEASUREMENT AND SCALING IN SMALL INTERNAL-COMBUSTION ENGINES

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ABSTRACT
The development of a dynamometer system suitable for measuring the power output and efficiency of small internal combustion engines with masses less than 1 kg is described. These measurements are difficult because engine speeds are high ranging between 10000 and 30000 RPM while torque levels are low ranging between 5 and 200 oz-in. Tradeoffs associated with various approaches to making these measurements are discussed and the measurement capabilities of the dynamometer system are described in detail. The system is used to measure the power output of a small piston engine of a type commonly used in R/C aircraft. Characterizing the performance of these engines is important in defining the current state of the art in small engine performance and for providing insight into the processes/loss mechanisms governing engine performance at small scales.

INTRODUCTION
The power output of internal combustion engines has been shown to obey a power-law scaling of the form \( y = Ax^b \) over a remarkably wide range of sizes (figure 1). This relationship seems to hold for smaller engines weighing less than 1 kg and, if properly validated, would be a very useful tool for optimizing the design of small UAVs. Another useful tool would be a similar validated relationship for fuel consumption or efficiency. Unfortunately, recent work that has attempted to estimate efficiency from manufacturers’ published data and whose results are summarized in figure 2 shows that log-linear scaling of efficiency with engine size breaks down for smaller engines with masses less than 1 kg (35 oz).

Understanding the performance of engines weighing less than 1 kg is important for several reasons. First, it is important to researchers who are trying to build extremely small heat engines and are trying to understand the tradeoffs associated with designing and operating heat engines weighing less than 5g. One problem they have encountered is that the most recent performance estimates indicate that the efficiency of these devices will be unacceptably low (~1-5%). However, this does not necessarily mean that the microengine idea is impractical. For example, while it could be the consequence of fundamental physical limitations, it could also be a simple consequence of the fact that the technology is still immature and needs refinement. Looking for trends in the performance of model engines that represent a more mature technology may provide some insight that cannot be gained from efficiency estimates like those presented in figure 2.

Second, having a good understanding of the performance of the smallest engines currently available is important to groups that cannot wait for microengines to be developed and are trying to build small air vehicles today. For example, consider the Low-cost Unmanned Air Vehicle being developed by the U.S. Navy. With a gross weight of about 20 lb, figure 3 shows that an efficient energy storage and propulsion system is critical to achieving the target range of 1500 miles (2414 km). It also shows that acceptable performance may be available from existing model aircraft engines if they are at least 25% efficient and can be operated on JP10. Figure 2 indicates that the target efficiency may be obtainable.

The challenge that the Navy faces is two-fold. First, reliable data indicating that 25% efficiency can be achieved is unavailable – estimates computed from manufacturers’ specifications show so much scatter as
American Institute of Aeronautics and Astronautics

Figure 1 Power output vs. size for a range of piston IC engines. Figure from reference 2, some data from reference 1.

Figure 2 Efficiency vs. engine size for a range of piston IC engines. Figure from reference 2.

Figure 3 Range of a low-cost UAV as a function of the energy density of its fuel $Q_r$ (storage efficiency) and the overall efficiency of the power plant. The calculation is based on the Brequet range equation assuming a takeoff fuel mass fraction $\chi=0.45$, a propulsive efficiency of $\eta_p=0.7$ and $L/D=8$.

Several factors complicate measurement of power output and fuel consumption in these small engines. Table 1 presents some data describing model aircraft engines that could be appropriate for powering a low-cost UAV. Note that all are two-stroke machines except for engines 3 and 4.

Table 1 Data for several single cylinder model aircraft engines that could be appropriate for powering a low-cost UAV. Note that all are two-stroke machines except for engines 3 and 4.

<table>
<thead>
<tr>
<th>Wt. (oz)</th>
<th>Pwr. (HP)</th>
<th>$\eta_\omega$</th>
<th>Spd (kRPM)</th>
<th>Displ. (in$^3$)</th>
<th>Torque (oz-in)</th>
<th>Fuel (oz/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>9.5</td>
<td>1.0</td>
<td>12</td>
<td>15</td>
<td>0.40</td>
<td>67.3</td>
</tr>
<tr>
<td>2</td>
<td>13.2</td>
<td>1.62</td>
<td>16</td>
<td>16</td>
<td>0.46</td>
<td>102.1</td>
</tr>
<tr>
<td>3</td>
<td>14.2</td>
<td>0.9</td>
<td>20</td>
<td>12</td>
<td>0.52</td>
<td>75.7</td>
</tr>
<tr>
<td>4</td>
<td>22.2</td>
<td>1.6</td>
<td>21</td>
<td>12</td>
<td>0.91</td>
<td>134.5</td>
</tr>
<tr>
<td>5</td>
<td>12.2</td>
<td>1.1</td>
<td>12</td>
<td>16</td>
<td>0.40</td>
<td>69.4</td>
</tr>
<tr>
<td>6</td>
<td>15.5</td>
<td>0.93</td>
<td>20</td>
<td>12</td>
<td>0.56</td>
<td>78.2</td>
</tr>
<tr>
<td>7</td>
<td>13.2</td>
<td>1.45</td>
<td>14</td>
<td>16</td>
<td>0.46</td>
<td>91.4</td>
</tr>
<tr>
<td>8</td>
<td>13.4</td>
<td>1.54</td>
<td>13</td>
<td>17</td>
<td>0.49</td>
<td>91.4</td>
</tr>
</tbody>
</table>

The work presented here discusses the development of a dynamometer used to make measurements of the torque, speed and fuel consumption of small model aircraft engines. These are subsequently used to measure the power output and overall efficiency of these engines. Some preliminary results are presented to demonstrate the capability of the instrument. Note that the power output $P$ of the engine is given by:

$$P = \Gamma \omega$$  \hspace{1cm} (1)

where $\Gamma$ and $\omega$ are respectively the torque and speed (in radians/sec) at the output shaft. The overall efficiency (or brake thermal efficiency$^5$) of the engine is defined as follows:

$$\eta_\omega = \frac{P}{\dot{m}_f Q_r}$$  \hspace{1cm} (2)

where $P$ is the power output of the engine, $\dot{m}_f$ is the fuel mass flow rate and $Q_r$ is the energy density of the fuel.

Table 1 presents some data describing model aircraft engines as a reference for the magnitudes of the quantities we expect to measure. Note that the efficiency was estimated from manufacturers’ data.
The methods used to make these estimates are reported in reference 2. First, the table indicates that operating speeds are relatively high which means that relatively well-balanced powertrains are required for safety and to control vibration. More importantly, however, this means that the torque produced is relatively low (since power is torque times angular velocity). Second, since most of the candidate engines listed in table I are 2-stroke, single cylinder designs, vibration from the engine itself will be strong which means we could be faced with measuring a small signal against relatively large background noise. As a result, care must be taken to damp these vibrations and to avoid exciting resonances in the measurement system. Third, and finally, fuel flow rates are quite small and lie below the range of all but the most expensive commercial fluid flow meters. Moreover, even if one of these units was acquired and used, installing it in the line between the fuel tank and the engine may cause operational problems. The reason is that many of these engines bleed a small portion of exhaust gas off the muffler and use it to pressurize the fuel tank. While this pressure is adequate to move fuel through a few feet of line, it may not be enough to drive fuel through the flow meter and an auxiliary fuel pump may be necessary.

What level of accuracy is required? Figure 2 indicates that discrimination within 5% of the nominal value of the overall efficiency ought to be sufficient to detect deviations from the log-linear behavior. This level of accuracy is certainly sufficient from the point of view of the operation of low-cost UAV.

MEASUREMENTS

Torque Measurement

Figure 4 is a schematic diagram illustrating the measurement principle. The engine to be tested is mounted in a cradle that rotates about an axis coincident with the engine’s axis of rotation. The cradle is prevented from rotating by a single load cell that anchors it to the cradle support. An absorber (hysteresis brake) connected to the engine through a gear system provides a continuously variable load to the engine. As the load is applied, the engine reacts against the cradle causing a load to be exerted on the load cell. This measurement scheme has several advantages. First, by measuring the engine’s reaction at its supports, and not downstream at the absorber, no accounting needs to be made for losses in the powertrain. Second, using a load cell and adjustable moment arm as opposed to a dedicated torque sensor means that the sensitivity of the instrument may be adjusted for different engines simply by attaching the load cell to the moment arm at different radial locations. This also means that a single load cell can be used to test a relatively wide range of engines.

Figure 5 is vector plot showing the relationship between the torques and forces acting on the cradle. The x and y axes are located at the center of rotation of the cradle and the z axis extends along the cradle’s axis of rotation.

A force balance on the cradle leads to the following expression for the torque $\tau_e$ exerted by the engine on the cradle

$$\tau_e = \tau_A + \tau_B - \tau_{AB} \times F_{lc}$$  \hspace{1cm} (3)

where $F_{lc}$ is the vector associated with the force applied by the load cell on the cradle. $\tau_A$ and $\tau_B$ are respectively the parasitic torques associated with the cradle bearing and the load cell attachment points. The moment arm is represented by $\tau_{AB}$ which extends from the cradle’s center of rotation (A) to the load cell attachment point (B). The cradle is constrained by
\( \vec{F}_L \) which acts along the load cell axis \( \vec{r}_{BC} \). The objective is to use the measurements of \( |\vec{F}_L| \) made by the load cell to determine the torque produced by the engine. The difficulty is that the load cell deforms under the influence of the load allowing the cradle to deflect through an angle \( \theta - \theta_p \). While not accounting for the deflection is fine for the dynamometer system described here, other systems are equipped with an elastomeric element between the load cell and the cradle to help damp vibration. In these systems, the deflection is relatively large and must be accounted for when computing \( \vec{F}_L \) in equation 1. Accordingly, we will go through the procedure for solving the more challenging problem involving large deflections.

Determining \( \vec{F}_L \) begins by recognizing that:

\[
\vec{F}_L = \left| \vec{F}_L \right| \frac{\vec{r}_{BC}}{|\vec{r}_{BC}|} \tag{4}
\]

Since \( \left| \vec{F}_L \right| \) is known (via the measurement), the challenge is to find \( \vec{r}_{BC} \). This is accomplished as follows. First, note that \( \vec{r}_{BC} \) is constrained by the center of rotation of the cradle (A) and the load cell anchor point (C) which are fixed:

\[
\vec{r}_{BC} = \vec{r}_{AC} - \vec{r}_{AB} \tag{5}
\]

Second, since \( \vec{r}_{AB} \) is constrained to the x-y plane, it can be written in terms of a single independent variable \( \theta \), the angle of the cradle,

\[
\vec{r}_{AB} = R \cos \theta \hat{i} + R \sin \theta \hat{j} + 0 \hat{k} \tag{6}
\]

where \( \hat{i}, \hat{j}, \) and \( \hat{k} \) are unit vectors associated with the x, y and z directions respectively and R is the length of the moment arm. Equations 3 and 4 lead to similar expressions for the fixed quantities \( \vec{r}_{BC0} \) and \( \vec{r}_{AB0} \) that are associated with the initial (unloaded) position of the cradle.

\[
\vec{r}_{BC0} = \vec{r}_{AC} - \vec{r}_{AB0} \tag{7}
\]

\[
\vec{r}_{AB0} = R \cos \theta_p \hat{i} + R \sin \theta_p \hat{j} + 0 \hat{k} \tag{8}
\]

Third, \( \left| \vec{F}_L \right| \) is found using Hooke’s law

\[
\left| \vec{F}_L \right| = k_L \left( |\vec{r}_{BC}| - |\vec{r}_{BC0}| \right) \tag{9}
\]

where \( k_L \) is the stiffness of the load cell and \( \vec{r}_{BC0} \) is known and constant. Equations 5 through 8 can be solved for the cradle deflection angle \( \theta \) as a function of the magnitude of the load cell reading \( \left| \vec{F}_L \right| \). Once the deflection angle is known, it is used to calculate \( \vec{r}_{AB}, \vec{r}_{BC} \), and finally \( \vec{F}_L \) via equations 6, 5 and 3 respectively.

**Table II Parameter values for torque measurement system**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Uncertainty</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>R</td>
<td>0.25 – 5.25</td>
<td>+/- 0.002</td>
<td>in</td>
</tr>
<tr>
<td>(</td>
<td>\vec{F}_L</td>
<td>)</td>
<td>0 – 80</td>
</tr>
<tr>
<td>( \theta_p )</td>
<td>0</td>
<td>+/- 5</td>
<td>deg</td>
</tr>
<tr>
<td>( C_x )</td>
<td>0.25 – 5.25</td>
<td>+/- 0.002</td>
<td>in</td>
</tr>
<tr>
<td>( C_y )</td>
<td>-4</td>
<td>+/- 0.002</td>
<td>in</td>
</tr>
<tr>
<td>( C_z )</td>
<td>0</td>
<td>+/- 0.050</td>
<td>in</td>
</tr>
<tr>
<td>( k_L )</td>
<td>16000-40000</td>
<td>+/- 20000</td>
<td>oz/in</td>
</tr>
<tr>
<td>( \delta_{max} )</td>
<td>0.0005-0.002</td>
<td>+/- 0.001</td>
<td>in</td>
</tr>
<tr>
<td>( m )</td>
<td>26.3</td>
<td>+/- 0.25</td>
<td>lb</td>
</tr>
<tr>
<td>( r )</td>
<td>3.94</td>
<td>+/- 0.01</td>
<td>in</td>
</tr>
</tbody>
</table>

Table II shows the values of the quantities referred to in equations 3 through 9 for the dynamometer system. The moment arm length \( R \) is adjustable through 9 increments from 0.25 in to 5.75 in. The load cell attachment point \( C \) is also adjustable through the same 9 increments and is set so that \( C_x = R \). The stiffness of the load cell \( k_L \) is estimated using the maximum rated force of the load cell and the maximum deflection of the load cell \( \delta_{max} \) via:

\[
k_L = \frac{|\vec{F}_L|_{max}}{\delta_{max}} \tag{10}
\]
While the uncertainty in $k_{lc}$ is relatively large because of the relatively large range of possible load cell deflections $\delta_{\text{max}}$, this does not impact the measurement accuracy significantly because $\delta_{\text{max}}$ is still small. The parasitic torques exerted at the cradle bearings A and the load cell attachment B were estimated using the measured breakaway torque. Note that only the component of the torque along the cradle axis is considered with all other components assumed equal to zero.

Figure 6 is a contour plot of torque as a function of moment arm length $R$. The open circles indicate the moment arm lengths available on the rig.

While the uncertainty in $k_{lc}$ is relatively large because of the relatively large range of possible load cell deflections $\delta_{\text{max}}$, this does not impact the measurement accuracy significantly because $\delta_{\text{max}}$ is still small. The parasitic torques exerted at the cradle bearings A and the load cell attachment B were estimated using the measured breakaway torque. Note that only the component of the torque along the cradle axis is considered with all other components assumed equal to zero.

![Figure 6 Contours of torque (oz-in) and uncertainty in torque (%) as a function of load cell force and moment arm length R. The open circles indicate the moment arm lengths available on the rig.](image)

Dynamic Analysis

Because all of the engines to be tested are single cylinder and most are two-stroke, they are expected to have strong 1/rev or 1/2rev perturbations in their torque output. As a result, the dynamic behavior of the cradle/load-cell system is another important consideration. The analysis in this section is simplified by neglecting the vector character of the governing equations. This is reasonable given that cradle deflections are quite small because $k_{lc}$ is relatively large. A simplified force balance on the cradle gives:

$$ (\Gamma_e + A\cos(\omega t)) - \beta \dot{\theta} - k_{lc} R^2 (\theta - \theta_0) = I \ddot{\theta} $$

(11)

where $\Gamma_e$ is the steady component of the torque, $A\cos(\omega t)$ is the time-varying component of the engine torque, $\beta$ is an overall damping coefficient associated with the cradle bearings, and $I$ is the moment of inertia of the engine and cradle about the cradle’s axis of rotation. Changing variables by introducing $\phi = \theta - \theta_0 - \theta_e$ where $\theta_e$ is the angle of the cradle associated with steady state operation and noting that $\Gamma_e - k_{lc} R^2 \theta_e = 0$ in the steady state transforms equation 8 to the well known harmonic oscillator

$$ I \ddot{\phi} + \beta \dot{\phi} + k_{lc} R^2 \phi = A \cos(\omega t) $$

(12)

with natural frequency given by

$$ \omega_n = \sqrt{\frac{k_{lc} R^2}{I}} $$

(13)

The engine speed $N_{nat}$ (RPM) associated with the natural frequency of the cradle-load cell system is:

$$ N_{nat} = \frac{60}{2\pi} \omega_n $$

(14)

Ideally, the engine-cradle system would be designed so that $N_{nat}$ is significantly greater than $N_{\text{max}}$, the engine’s maximum operating speed, so that resonances of the cradle/load cell system could not be excited. In practice, however, keeping $\omega_n$ high is difficult. The physical size of the engine places a lower bound on the total moment of inertia $I$. Meanwhile, load cells, by their nature, are relatively stiff. Any structure associated with the cradle increases $I$ and lowers $\omega_n$ further.

The approach taken here is to increase $I$ by adding pairs of weights that are directly opposite the cradle’s axis of rotation (so that the center of mass is not affected) to push $\omega_n$ below the operating range of the engine. The overall moment of inertia becomes

$$ I = I_e + I_c + mr^2 $$

(15)

where $I_e$ is the moment of inertia of the engine alone, $I_c$ is the moment of inertia of the cradle, and $mr^2$ accounts...
for the additional weight added to the cradle. In the latter term, \( m \) is the mass of the extra weight added to increase the overall moment of inertia and \( r \) is the radius at which this weight acts. While this strategy is not ideal because the engine must traverse the fundamental frequency as it accelerates to operating speed, locking the cradle down and releasing it after the engine has started can prevent resonance-induced damage to the sensor. Table II shows the values for the terms in equations 13 and 15.

Figure 7 shows how the highest possible engine speed associated with the natural frequency of the cradle depends on moment arm length \( R \). Increasing the moment of inertia by adding mass to the cradle pushes \( N_{\text{nat}} \) out of the normal operating range of the engine and should permit stable operation over the entire operating range. The figure also shows that decreasing the moment arm length has the additional benefit of moving the natural frequency farther below the operating range of the engine.

![Figure 7 Engine RPM associated with the first resonant mode of the cradle-load cell system for various moment arm lengths. Increasing the cradle’s moment of inertia pushes the fundamental frequency below the operating range of the engine.](image)

**Fuel Flow Rate Measurement**

The fuel flow rate is determined by making a series of measurements of the mass of the fuel in the fuel tank over a period of time. A least squares fit is then performed to get the time rate of change of the weight of the fuel in the tank. This is the fuel mass flow rate.

**Speed Measurement**

Engine speed is measured using an ‘off the shelf’ system manufactured by ElectroSensors. It consists of a magnetic ‘pulser’ disc containing sixteen alternating poles that is attached to the power-takeoff shaft, a proximity switch that detects the passage of these poles, and a model SA420 signal conditioner that outputs an analog voltage proportional to frequency. The accuracy of this system is +/- 0.1% of the reading.

**THE DYNAMOMETER SYSTEM**

Figure 8 is a photograph of the dynamometer system showing the important components. The engine is mounted in a cradle that is supported at either end by precision bearings with very small breakaway torques. The upper end of the load cell is attached to a moment arm which, in turn, is attached to the cradle. The lower end of the load cell is attached to the cradle support. A Sensotek model 31 load cell with a full scale range of 5 lb (80oz) is used. The deflection and accuracy of the cell are given in table II.

An OS 46FX aircraft engine is used in this work to evaluate the performance of the dynamometer system. It corresponds to entry number 2 in table I and is operated on glow fuel that is a mixture of 10% nitromethane, 20% castor oil, and 70% methanol. The tip of a stainless steel sheathed K type thermocouple contacts one of the cylinder head bolts to provide an indication of cylinder head temperature. A similar thermocouple inserted into the muffler exit provides an indication of the exhaust gas temperature. While both thermocouples are attached to the cradle support, they have a negligible impact on the torque measurement. The throttle is controlled by a Futaba servo, FM receiver, and battery all attached to the cradle. This allows throttle adjustments to be made remotely without impacting the torque measurements.

The 32 oz fuel tank rests on an Ohaus model SCA210 portable scale located outside the cradle support. The scale has been modified to provide an analog signal proportional to the scale reading. Two flexible hoses connect the tank to the engine on the cradle. One carries the fuel to the needle valve inlet. The other extends from a fitting on the muffler back to the fuel tank to pressurize it. The moment induced by these hoses running off the cradle to the tank influence the torque measurement. However, most of their effect is mitigated by calibrations performed before and after every run.

The power produced by the engine is absorbed by a Magtrol model HB-880 double hysteresis brake. The maximum operating speed of the brake is 8000 RPM and so a 2.5:1 transmission connects the brake to the engine. The torque applied by the brake is continuously adjustable from zero to its maximum rating by changing the current supplied to the brake’s magnetic coils. A DC source supplies the current and can be controlled...
from a panel-mounted potentiometer or a 0-5V analog signal. The brake is rated to dissipate up to 800W of power w/o forced air cooling. More power can be dissipated on an intermittent basis or if cooling air is provided. Two infrared thermocouples monitor the temperatures of the brake drums to detect if the brakes are overheating.

The speed sensing ‘pulser’ disc is sandwiched in a special cylindrical mount. One end attaches to the shaft of the hysteresis brake while the other supports a model airplane nose cone. This nose cone receives a standard model aircraft electric starting motor and is used to start the engine. The proximity sensor sits between the ‘pulser’ disc and the hysteresis brake.

Since all of the engines currently scheduled for testing are air cooled, replacing the continuous air stream provided by the propeller is essential. Accordingly, a special duct has been constructed that directs air from a blower around the moving parts of the cradle and directly across the cylinder head. The engines will overheat rapidly without this cooling air.

Data Acquisition

A Lab View system is used to condition and log signals from all of the sensors. Analog signals are amplified and low-pass filtered using a National Instruments SCXI 1327 instrumentation amplifier. Thermocouple signals are amplified by a National Instruments SCXI 1112 thermocouple amplifier. The analog to digital conversion is performed by an eight channel, 16 bit card resident in a PC. Lab View software controls the acquisition and display of data.

Data is recorded in one of two modes. In the default strip chart mode, the channels are scanned once per second and the data written to a file which provides a record of everything that happened in the course of an experiment. This data is also written to virtual indicators and strip charts displayed on a software panel that provide real-time graphical indications of the state of the experiment. The difficulty with this mode of operation is that the timing of the scans is accomplished through software. Since other processes compete for CPU cycles, the true time elapsed between channel scans will not be constant. While this variability is small, it is not acceptable when one is interested in measuring rates (like fuel flow rate) because the true time elapsed between measurements is not be constant.

As a result, a button on the panel is provided which causes the software to enter a ‘burst’ mode of operation in which a fixed number of samples is acquired using a
hardware trigger to ensure consistent timing intervals. Each burst consists of 1000 samples per channel acquired at 1 kHz. When the ‘burst’ is complete, the software returns to ‘strip chart’ mode. The ‘burst’ mode is used to acquire a large number of samples at a single operating point.

Calibration
The torque and fuel weight measuring systems are calibrated before and after each experiment. This is accomplished by running the data acquisition program and applying a series of known weights to the scale and moment arm. A data ‘burst’ is acquired for each new application of weights.

Fuel choice and determination of heating value
The OS46FX engine is designed to operate on ‘glow fuel’ which is a mixture of methanol (C₃OH), nitromethane (CH₃NO₂), and castor oil. Different fuel mixtures are available. Mixtures with larger amounts of nitromethane and lesser amounts of castor oil generally produce more power but at the expense of engine operating life. This work uses a relatively conservative 70% methanol, 10% nitromethane, 20% castor oil mixture.

The density of the fuel mixture is given by:

$$\rho_{\text{mix}} = \rho_m \chi_m + \rho_{nm} \chi_{nm} + \rho_o \chi_o$$  \hspace{1cm} (16)

where $\chi_m$, $\chi_{nm}$, and $\chi_o$ are the volume fractions of methanol, nitromethane and castor oil in the fuel. Similarly, $\rho_m$, $\rho_{nm}$, and $\rho_o$ are the densities of the various components. The energy density of the mixture is given by:

$$Q_{r,\text{mix}} = \frac{Q_{r,m} \rho_m \chi_m + Q_{r,nm} \rho_{nm} \chi_{nm} + Q_{r,o} \rho_o \chi_o}{\rho_{\text{mix}}}$$  \hspace{1cm} (17)

Where $Q_{r,m}$, $Q_{r,nm}$, and $Q_{r,o}$ are the heating values of the various components. The values of the quantities in equations 16 and 17 are presented in table III.

Table III Properties of glow fuel constituents and overall properties of the mixture.

<table>
<thead>
<tr>
<th>Component</th>
<th>$\chi$</th>
<th>$\rho$</th>
<th>$Q_r$</th>
</tr>
</thead>
<tbody>
<tr>
<td>C₃OH</td>
<td>0.7</td>
<td>0.79</td>
<td>22.6</td>
</tr>
<tr>
<td>CH₃NO₂</td>
<td>0.1</td>
<td>1.11</td>
<td>11.6</td>
</tr>
<tr>
<td>Castor Oil</td>
<td>0.2</td>
<td>0.96</td>
<td>44.0</td>
</tr>
<tr>
<td>Mixture</td>
<td>1.0</td>
<td>0.86</td>
<td>26.0</td>
</tr>
</tbody>
</table>

Operation
After the pre-run calibration, the cradle is clamped to the cradle support to prevent damaging the load cell during engine startup. Once the engine is started, the cradle is unlocked and the engine is leaned according to the manufacturers’ instructions. This involves closing the needle valve that meters the fuel until the engine speed peaks, and then retarding the needle valve about ¼ turn to prevent overheating the engine. Failing to do this can cause the engine to operate too lean and overheat. In one case, operating too lean actually burned a hole through a piston.

With the mixture properly adjusted, the hysteresis brakes are energized to apply a load to the engine. When the engine speed has come to equilibrium, a burst of measurements is acquired. The process is repeated for a variety of loads (or brake settings) and throttle settings. The cradle is re-locked before exhausting the fuel or otherwise shutting down the engine to prevent damaging the load cell as the engine stops.

After the engine stops, the cradle is unlocked and a second calibration of the torque measuring and fuel weighing systems is performed to see how much the calibration of the system changed.

Power curves are determined by varying the applied load while measuring torque and velocity. Efficiency can be determined from the time variation in the mass of fuel and the average values of the torque and engine speed.

PRELIMINARY RESULTS

Figures 9, 10, and 11 are plots of the various sensor readings accumulated in the strip chart file over the course of an experiment. The first two curves in figure 9 show the torque and speed of the engine measured by the data acquisition system. The last curve shows the speed of the engine as indicated on the digital display of the speed sensor’s signal conditioner. Note that the speed measured by the data acquisition system reaches an upper bound because its channel on the data acquisition system was not configured properly. As a result, the values based on the reading of the digital display are the ones used to compute power. Note that the display and the measurements made by the data acquisition system are consistent for speeds below about 12600 RPM.

Figure 10 shows the weight of the fuel in the tank and a voltage proportional to the current being supplied to the hysteresis brake. This latter curve provides a good illustration of how the experiment was conducted: It began by starting the engine with the cradle locked down and zero load applied by the brake. Once the engine was operating smoothly and had been leaned
according to the manufacturer’s instructions, a data burst was acquired indicating the no-load condition. Then, a load was applied to the engine by increasing the current to the brake. The system was allowed to come to equilibrium and another data burst was acquired. This process was repeated for progressively larger and then smaller loads before the engine was shut down. Note that the trends in torque and speed mirror the trend in brake current (or applied load): As the load is increased, the torque increases and the engine speed decreases.

Figure 9  Time history of engine torque and speed. The throttle is fully open and the mixture is leaned per factory instructions.

Figure 10  Time history of brake current and fuel flow rate for the experiment of figure 9.

Figure 11  Time history of brake temperature, cylinder head temperature, and room temperature for the experiment of figure 9.

Figure 11 shows the time history of four of the temperature measurements made on the rig. The fifth, the exhaust gas temperature, is not shown because the probe failed almost immediately after engine start. The first two, the temperatures of each hysteresis brake rotor, also mirror the trend in brake current. This makes sense since increasing the applied torque increases the amount of power being dissipated by conversion to heat. The temperature of the cylinder head increases rapidly to a steady-state value indicating that the engine has come up to operating temperature. The fact that it is not too high indicates that the cooling system is doing its job. At some point, however, the tip of the thermocouple probe vibrated loose. The subsequent intermittent contact with the engine resulted in a somewhat lower but constant reading for the remainder of the experiment. Finally, the temperature of the room remained virtually unchanged during the course of the run.

Figure 12  Calibration of the torque measurement system showing consistency before and after a run.

Figure 12 shows the results of the calibration of the torque measurement system before and after a run. Note that in spite of the harsh operating environment evidenced by the noise in the plot of torque vs. time in figure 9, the calibration of the system seems to have been maintained. Figure 13 shows the results of a
similar check performed on the fuel scale. It also retained its calibration.

Figure 13 Calibration of the scale showing consistency before and after a run.

Figures 14 and 15 show torque and power as a function of engine speed. The upper curve presents the burst data acquired during the run presented in figures 9 and 10 where the engine was leaned. The lower curve presents data from an earlier experiment where the engine was allowed to operate rich. Note that while the curves show some hysteresis, the general trend is an increase in torque and power as speed decreases. This makes sense because the speed is decreased by applying a load. At some point, however, one would expect the power to begin to decrease as the load increases. While we could have explored this area of the power curve, we did not because we did not want to stall the engine and risk damaging the load cell. Future experiments will try this riskier experiment. Finally, the effect of non-ideal operating conditions is also evident: operating too rich decreases the power output of the engine.

Figure 14 Torque as a function of engine speed under leaned and rich conditions.

Figure 15 Power as a function of engine speed under leaned and rich conditions.

Figure 16 shows the mass flow rate of the fuel computed from the scale voltage for the same lean and rich runs. Note that while the data are considerably noisier and do not show a strong trend with speed, they do show a trend with mixture: Operating rich consumes more fuel. The large scatter in the fuel flow rate leads to similarly large scatter in the efficiency estimates presented in figure 17. However, the trends are as expected: Efficiency tends to decrease with increasing operating speed and operating richer is less efficient than operating leaner. Note also that this level of uncertainty (a few percent) is adequate to resolve the scatter in the efficiency estimates presented in figure 2.

Figure 16 Fuel flow rate at different operating points under leaned and rich conditions. Error bars for the rich case are the same as those for the leaned case and have been removed for clarity.

Figure 17 Overall efficiency at different operating points under leaned and rich conditions.
CONCLUSION
A dynamometer system suitable for measuring the performance of small internal combustion engines has been constructed. In the absence of vibration, the measurement system has been shown to be capable of measuring power output to within 0.5%. The first data, however, indicate that the harsh vibratory environment and other as yet un-discovered factors have reduced this capability considerably. Nevertheless, the system has shown that measurements with sufficient resolution to resolve some of the conflicts in the scaling analysis are possible. The preliminary data indicate that the power output of the engine was about 0.5 HP, considerably lower than the manufacturer’s estimate of 1.65 HP. The preliminary estimate of the efficiency was about 3.5%. This is unacceptably low for the US Navy’s Low Cost UAV.

FUTURE WORK
Future work will focus on improving the fuel flow rate measurement, adding the ability to measure air flow rate (to determine scavenging efficiency and actual mixture ratio) and manifold pressure, and by making the dynamometer system generally more robust and less affected by vibration. In addition, more runs will be performed with the current engine to see if its optimum performance has in fact been obtained. Once the reliability of the system has been established, a range of engines including much smaller ones will be tested to identify scaling trends.

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