

Investigating Combustion in a Mini Internal Combustion Engine

Asish K Sarangi, Pramod S Mehta, and A Ramesh
Indian Institute of Technology Madras

ABSTRACT

Owing to a high power-to-weight ratio, mini internal combustion engine is used in propelling an unmanned air vehicle. In comparison to the performance characteristics, the investigations on the combustion aspects of mini engines are scanty. This investigation concerns study of the combustion process of a mini engine and its variability. For this purpose, the experimental cylinder pressure histories were obtained on a laboratory set-up of a 7.45 cm³ capacity mini engine. The analyses of experimental data at different throttle settings reveal that there existed a varied range of rich and lean misfiring limits around a reference equivalence ratio that corresponds to the respective maximum indicated mean effective pressure. At the limiting equivalence ratios, cylinder pressure measurements showed a high degree of cycle-to-cycle variations. In some cases, a slow combustion or misfiring event preceded a rapid combustion. From the energy release analysis, it is observed that the combustion rate was very rapid in the initial phase which subsequently slowed down. Also, there are two peaks of energy release observed under rich mixture conditions.

CITATION: Sarangi, A., Mehta, P., and Ramesh, A., "Investigating Combustion in a Mini Internal Combustion Engine," *SAE Int. J. Aerosp.* 8(1):2015, doi:10.4271/2015-01-9002.

INTRODUCTION

Mini internal combustion (IC) engines with capacities ranging from 0.2 to 15 cm³ and power outputs varying from 20 to 2000 W find wide applications in unmanned air vehicles (UAVs). Owing to the high specific power requirements from IC engines used in mini UAVs, there is a preference for two-stroke cycle reciprocating engines with glow plug assisted ignition systems. Modern glow plug based mini engines are powered by high-energy glow fuel containing about 60-70% (by volume) of methanol, 18-25% of castor oil and the rest nitro-methane [1]. The relevant physical properties of the glow fuel constituents are listed in [Table 1](#).

The amount of fuel that can be burned in an IC engine is mainly limited by the amount of air inducted per cycle. Volumetric efficiency indicates how effective an engine is in inducting its charge. The high latent heat of vaporization of methanol (primary fuel) leads to increased charge cooling and hence greater volumetric efficiency and power. Similarly nitro-methane, when added as a fuel component, is reported to boost engine output because of its high specific heating value (the amount of heat that can be released per unit mass of air under stoichiometric mixture conditions) which is almost twice that of methanol (refer to [Table 1](#)). A significantly higher mass of nitro-methane can be burned in an engine due to its relatively low air requirement. In addition, the oxygen present in nitro-methane improves combustion rates as evident in drag racing, where nitro-methane is the primary fuel [8]. Thus, the addition of even a small fraction of nitro-methane can improve power output. Glow-plug

based mini IC engines use a resistively heated glow plug, having a platinum coil, to initiate combustion of the fuel-air charge. Once the engine warms up, the power supply to the glow plug is cut off. Continued heating is provided to the glow plug by the heat carried over from the previous cycle and the exothermic catalytic reaction between the platinum coil and the methanol in the fuel [9]. In a mini engine, the start of combustion is determined by the residual gas fraction, in-cylinder temperature, chemical kinetics of the fuel mixture and the catalytic reaction between methanol and the glow plug element. However, the physical and chemical properties of the castor oil (listed in [Table 1](#)) suggest that a fraction of it may take part in combustion.

The thermal efficiency of mini IC engines is reported to be merely 5-18% [7]. An increased surface area to volume ratio of the engine cylinder with reduced engine size results in increased thermal and frictional losses and hence a reduction in brake thermal efficiency [11, 12, 13, 14]. Higher flame quenching, short mixture residence time (owing to high engine speed), and insignificant turbulent mixing are considered responsible for the inefficiency of the mini IC engines [7, 14]. Although literature related to the performance assessment of the mini engines is available [11, 12, 13, 15, 16], investigations regarding the combustion characteristics of the mini engines have not been reported in detail, and only a few studies [10, 17, 18, 19] exist in the literature. The performance and emission characteristics of a 7.45 cm³ glow-plug-based engine was investigated and reported in an earlier publication [16]. This paper discusses the combustion characteristics of this mini engine in detail.

Table 1. Properties of glow fuel constituents [2, 3, 4, 5, 6, 7]

Fuel	Methanol	Castor Oil	Nitro-methane
Chemical formula	CH ₃ OH	C ₁₈ H ₃₄ O ₃	CH ₃ NO ₂
Molecular mass (kg/kmol)	32	298	61
Stoichiometric air–fuel ratio	6.5	11.5	1.7
Lower heating value (MJ/kg)	20	36.2	11.5
Lower specific heating value (MJ/kg stoichiometric air)	3.07	3.15	6.76
Density (kg/m ³)	796	960	1139
Self-ignition temperature (°C)	455	449	418
Latent heat of vaporization (kJ/kg)	1.16	Not available	0.627
Flame speed (m/s)	0.52	Not available	0.39-0.45

EXPERIMENTAL APPARATUS

The research facility used in this work was based on a 7.45 cm³ two-stroke, glow-plug-based mini IC engine with a fixed pitch propeller in a custom-built test rig. The specifications of the engine and the propeller are given in Table 2. The test set-up is shown schematically in Figure 1.

Table 2. Specifications of the engine and the propeller

Engine	
Type	Two-stroke glow plug based compression ignition engine
Make	OS MAX, Japan
Number of cylinders	One
Bore/Stroke	22/19.6 mm
Displacement volume	7.45 cm ³
Scavenging method	Schnuerle
Rated power	1.2 kW @ 16,000 rpm
Propeller	
Type/Make	Windsor propeller/ Master airscrew
Diameter	254 mm
Pitch	152.4 mm

A reaction type dynamometer was developed to measure the engine torque with propeller as a load. For this purpose, the engine was mounted on a cradle. A shaft, supported on two anti-friction ball bearings, was fixed to the cradle. When the engine was operated, the torque exerted on the propeller produced a reaction torque that tended

to rotate the cradle in the opposite direction. A lever of known length connecting the shaft and pressing against a load cell prevented the rotation of the cradle and hence, measured the reaction force. The reaction torque was the product of the reaction force (measured by the load cell) and the length of the lever. This dynamometer was calibrated with known torques. The entire set-up was mounted on a platform that was supported with a set of linear anti-friction ball bearings. Owing to the thrust produced, the whole platform would tend to slide forward which was resisted by calibrated springs. The deflections of the springs were measured by a calibrated linear variable differential transformer (LVDT). Engine speed was measured with a Contrinex optical speed sensor. A photograph of the dynamometer and the thrust measurement set-up is shown in Figure 2.

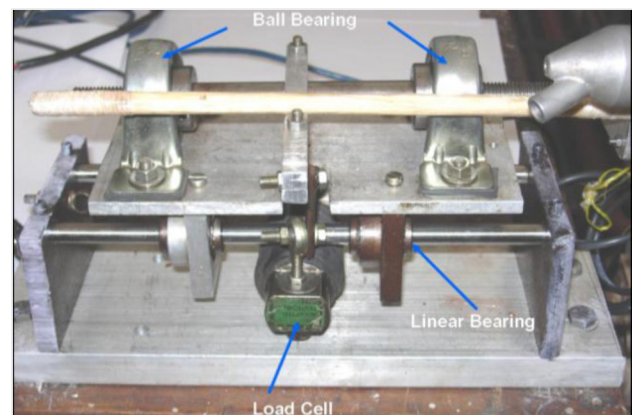
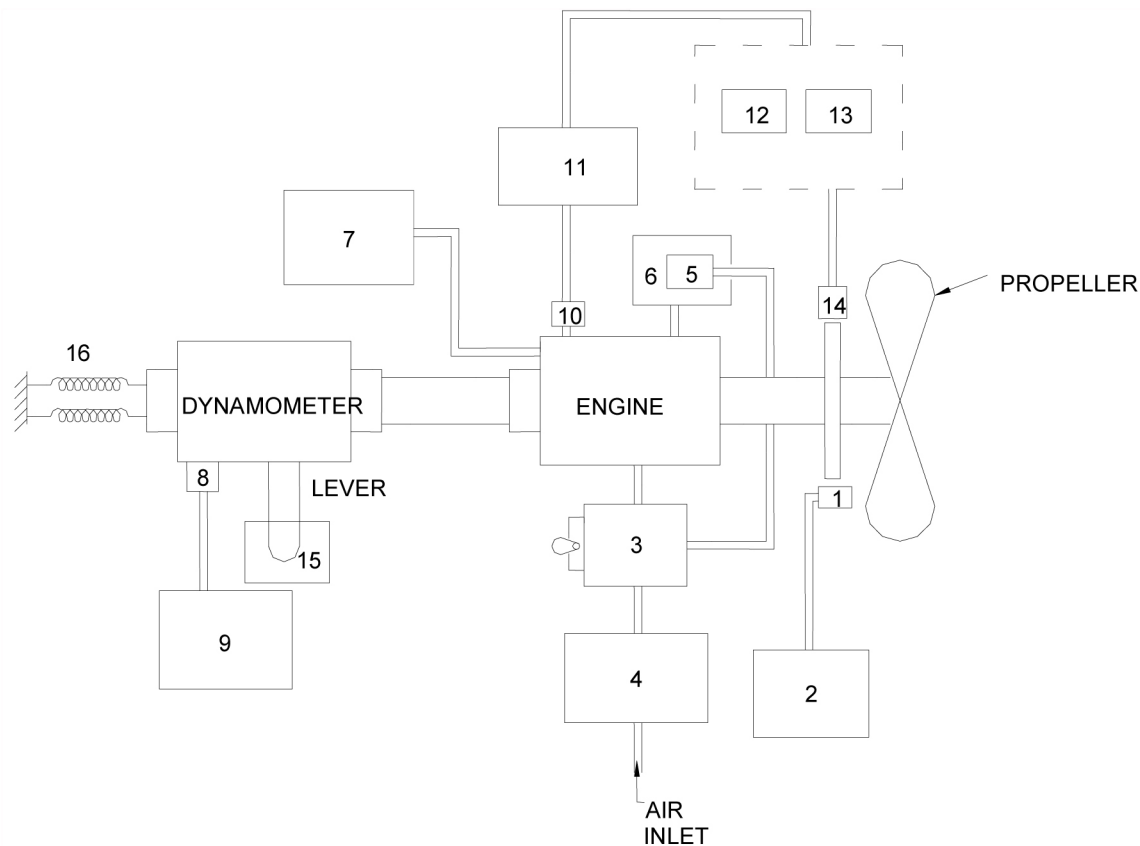


Figure 2. Torque and thrust measurement set-up for the mini engine



- | | | | |
|--------------------------------|-------------------------|----------------------|---|
| 1. Optical Speed Sensor | 2. Speed Indicator | 3. Carburetor | 4. Air Flow Meter |
| 5. Fuel Tank | 6. Weighing Balance | 7. Exhaust Analyzer | 8. Linear Variable Differential Transformer |
| 9. Carrier Frequency Amplifier | 10. Pressure Transducer | 11. Charge Amplifier | 12. Analog to Digital Card |
| 13. Personal Computer | 14. TDC Position Sensor | 15. Load cell | 16. Springs |

Figure 1. Schematic of the test rig

Airflow rate was measured with a positive displacement Invensys G4B air flow meter connected to a surge tank of 750 cm³ capacity. This surge tank arrangement considerably reduced the fluctuations experienced due to the intermittent flow of air through the engine. Flexible hoses were used to connect the air flow meter with the engine carburettor via the surge tank. At each operating condition, the time required for a fixed quantity of air flow (0.02 m³) was determined. The average gravimetric air flow rate was obtained by multiplying the average of three consistent values of the measured volumetric air flow rate with the density of the ambient air. Another long flexible hose was used to connect a fuel tank with the needle valve of the carburetor. This fuel tank was placed on a precision balance. In this work, a needle valve was used to vary the fuel flow rate, whereas the throttle was used for adjusting the air flow rate. Gravimetric fuel flow rate was estimated by finding out the time taken for the consumption of a fixed mass of fuel. The average gravimetric fuel flow rate was determined by three consecutive fuel flow measurements at the same operating condition. The parasitic

torque and the thrust associated with the hoses and wires and the frictional losses in the bearings were taken into account during the calibration of the dynamometer and the LVDT respectively.

A Kistler 601A piezo-electric, water-cooled pressure transducer was mounted on the cylinder head with a very small passage. The pressure transducer was coupled to a matching charge amplifier (Kistler 5007) and the sensitivity of the entire system was 10 bar/volt. The data acquisition system, that was used in this work, had a minimum detection capacity of 2 mV, i.e., the smallest change in pressure that could be detected was 20 mbar. In order to determine the crank shaft position, an electro-optical top dead center (TDC) position sensor was used to give a single square voltage pulse exactly at TDC. This sensor had a well aligned pair of infrared diode and photo transistor. A thin metal plate with a slot was fixed to the flywheel. The plate and the sensor were positioned in such a way that at TDC, the upper edge of the plate interrupts the light emitted by the diode and the output voltage from the photo-transistor increases to 3.5 volt from 0 volt. Voltage signals from the optical sensor were fed into the analog to digital converter and then to the data acquisition system along with

the pressure signals for recording. The crank angle location corresponding to each pressure reading was determined by linearly interpolating between each TDC pulse. The set-up for obtaining the pressure-time data is shown in Figure 3. It should be noted that during pressure crank angle data acquisition, engine performance and emissions data were not obtained.

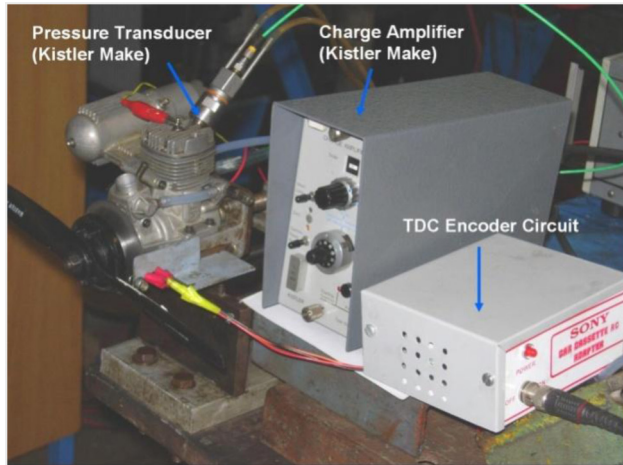


Figure 3. Piezo-electric pressure transducer with the charge amplifier

Adequate care was taken in mounting the pressure transducer to avoid the effect of acoustic frequency on pressure measurements due to the unwarranted cavity. The schematic of the pressure transducer mounting with the relevant dimensions are shown in Figure 4. The acoustic frequency in the passage at a mean gas temperature of 500 K would be ~ 14 kHz. Theoretically at this frequency, the effect on pressure values would be marginal but the phase delay could be typically of the order of $\sim 4.7^\circ\text{CA}$ at 11,000 rpm. However, this was neither established in this work nor reported in an earlier investigation of Van Aken [20]. In support of this fact, the observation of McCullough [21] is worth mentioning that it may be difficult to distinguish between the pressure traces of knock-like abnormal combustion and pressure traces due to cavity resonance with a passage mounted pressure transducer. However, no such irregularities were observed in the pressure traces acquired during the present study (refer to Figure 5). Thus, the authors believe that the pressure-time traces reported in this work are least affected by the presence of the cavity.

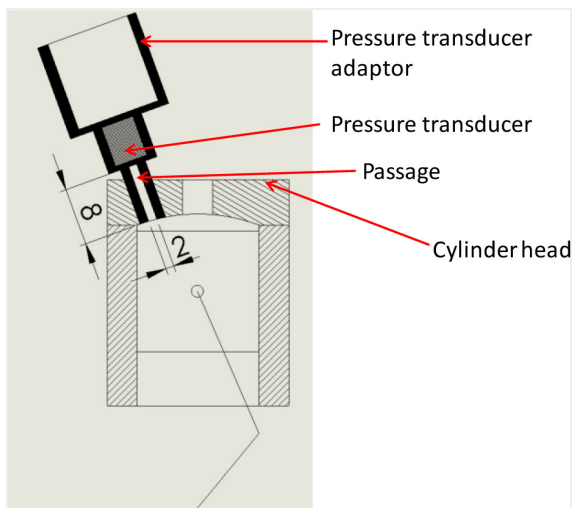


Figure 4. Schematic of the pressure transducer mounting on the cylinder head

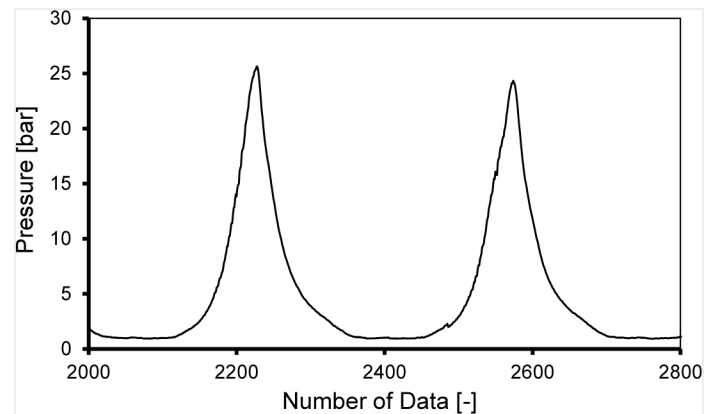


Figure 5. Plot of the pressure-time data of two cycles at wide-open throttle

EXPERIMENTAL PARAMETERS

The measured pressure-time data were transformed to pressure-crank angle data (with the data from the speed sensor and the TDC position sensor) using a software code developed in-house. Two hundred consecutive engine cycles were used to obtain the ensemble averaged in-cylinder pressure-crank angle data that was used to derive the combustion parameters. The magnitudes and occurrences of peak pressures were obtained directly from the pressure-crank angle histories. The gross indicated mean effective pressure (imep) of the engine was obtained by dividing the average work (obtained from the pressure crank angle data during the close part of the cycle after the intake and exhaust ports were closed) by the engine displacement volume. Coefficient of variation (COV) is a useful indicator of the cyclic variability. The COV of imep was calculated as the ratio of the standard deviation to the mean of the imep. Similarly, the COV of peak pressure was calculated as the ratio of the standard deviation to the mean of the peak cylinder pressure.

Based on the First Law of Thermodynamics, the instantaneous net energy release rate was determined from the averaged in-cylinder pressure-crank angle data as:

$$\frac{dQ_n}{d\theta} = \left(\frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} \right) \quad (1)$$

where θ is the crank angle, γ is the specific heat ratio, p and V are the instantaneous cylinder pressure (in Pa) and volume (in m^3) respectively. As the compression and expansion processes in the engine are not adiabatic and accompanies heat transfer, a polytropic index (1.25) evaluated from the pressure-crank angle data over a wide range of engine operating conditions is substituted for the specific heat ratio. The polytropic compression indices at different loads were determined by fitting a least squares best fit straight line to the calculated $\log p - \log V$ data during the closed part of the cycle prior to the start of combustion. Thus a mean compression polytropic index of 1.25 was obtained from a large number of measured pressure histories at different engine operating conditions investigated in this work.