Spark Ignition Engine — Data Analysis Notes

Background

A schematic of a four-stroke, spark ignition, internal combustion (IC) engine is shown in Figure 1. The four strokes of the cycle include:

(i) air-fuel intake into the combustion chamber,

(ii) compression of air-fuel mixture inside combustion chamber,

(iii) power generation due to explosion inside combustion chamber, and

(iv) expulsion of exhaust gases out of the combustion chamber.

The power generated during stroke (iii) is used to turn the crankshaft, which can then be used to drive the axle of an automobile, for example, to produce mechanical work. The performance characterization of such an engine involves specifying the amount of mechanical power (or brake power, $\dot{W}_b$) that can be produced as a function of the rotational speed (ω) of the crankshaft, as well as the thermal efficiency (η). These physical quantities cannot be measured directly. Therefore, in the laboratory, we measure surrogate quantities instead, and calculate the desired quantities of interest based on known physical (and theoretical) relationships.

The performance of an IC engine will depend on the type of fuel used, the manner in which the fuel is ignited (spark ignition versus compression ignition), the displacement volume of the piston (bore of the cylinder camber and stroke of the piston), and the compression ratio of the cylinder. Figure 2 shows important dimensions of the piston and cylinder camber. Note, the compression ratio (r) is the ratio of the maximum to minimum cylinder volume, and is calculated as

$$r = \frac{\mathcal{V}_{\max}}{\mathcal{V}_{\min}} = \frac{\mathcal{V}_{\mathrm{BDC}}}{\mathcal{V}_{\mathrm{TDC}}}, \tag{1}$$

where TDC and BDC refer to the "top dead center" and "bottom dead center" positions of the piston. The bore, stroke, and compression ratio for a given engine will be specified by the manufacturer. The appendix contains the values appropriate for the experiment you conducted in the lab.

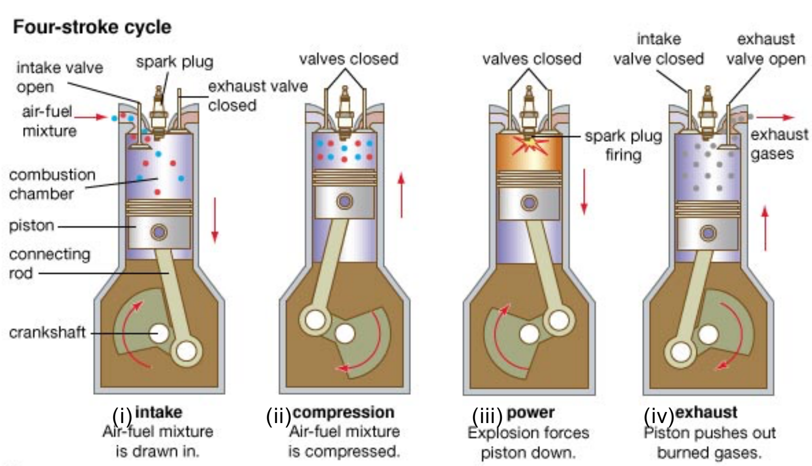

Figure 1: Illustration of a two-stroke and four-stroke internal combustion engine.

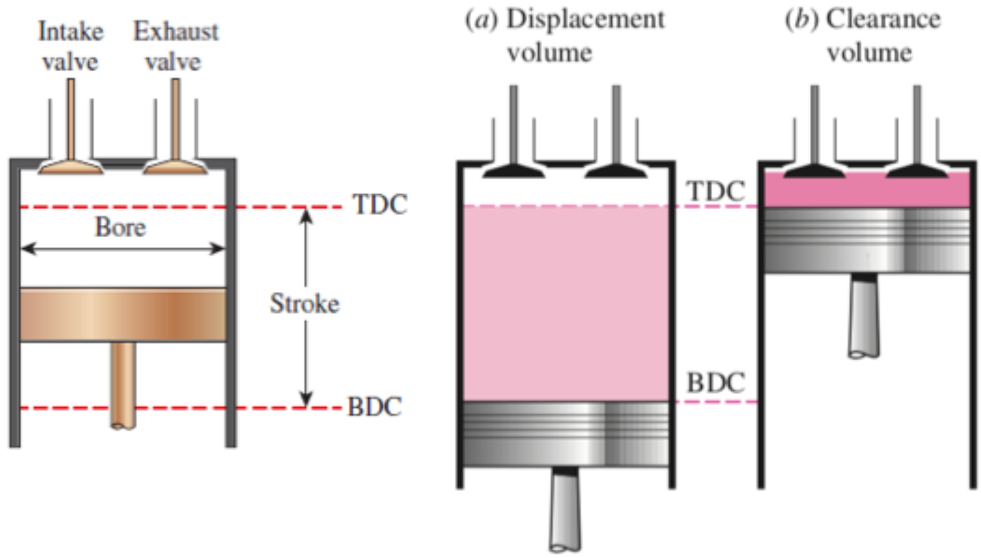

Figure 2: Piston and cylinder schematic of an internal combustion engine. The two horizontal dashed lines denote the position of the piston at its two extremes, referred to as "top dead center" (TDC) and "bottom dead center" (BDC).

Experimental Measurements

In the laboratory, instruments are available for acquiring the following measurements, expressed in their native units:

- → ω: rotational speed of the shaft [RPM],
- → τ: torque of the crankshaft [N·m],
- → t: time interval of the experiment [sec],
- → $\mathcal{V}$: volume of fuel consumed over the measured time interval, [1 cm drop of liquid level in the fuel tube equals a volume of 5.1 cm^3],
- → r: compression ratio of the cylinder [unitless]
- → T_{in}: inlet air temperature [°C],
- → P_b: inlet air pressure or barometric pressure [mmHg],
- → T_{fuel}: inlet fuel temperature [°C],
- → T_{exhaust}: exhaust gas temperature [°C].

(a) Mechanical Power

From these, the mechanical power (also referred to as brake power) is given by the rate of work done by the crankshaft when a brake is applied, and can be calculated from the following

$$\dot{W}_b = \tau\,\omega\ , \tag{2}$$

where τ and ω denote the torque about the axis of rotation of the shaft and the angular velocity of the shaft, respectively. Remember that the measured quantities will not necessarily be in the proper SI units. Therefore, it is extremely important to be mindful of your units and perform the necessary unit conversions to the SI system in your data analysis code. For example, the proper SI units on ω are rad/sec.

(b) Thermal Efficiency

The thermal efficiency of the engine is defined as the ratio of the mechanical power over the inlet power, and is calculated as

$$\eta = \frac{\dot{W}_b}{\dot{E}_{\text{in}}}, \tag{3}$$

where $\dot{E}_{\text{in}}$ denotes the rate of energy into the engine via the fuel. The amount of energy contained in the fuel is based on its Lower Heating Value (LHV), also referred to as Net Heating Value, which is a material property. A bomb calorimeter can be used to measure the LHV of a substance, as described in the appendix. The SI units associated with LHV are typically given as MJ/kg. You can lookup the LHV for any given fuel from published engineering tables. The input energy rate is then

$$\dot{E}_{\text{in}} = \dot{m}_{\text{fuel}}\, LHV, \tag{4}$$

where $\dot{m}_{\text{fuel}}$ represents the mass flow rate of fuel into the engine during the time interval of the experiment. The latter is, of course, given by

$$\dot{m}_{\text{fuel}} = \rho_{\text{fuel}}\, \dot{\mathcal{V}}_{\text{fuel}}, \tag{5}$$

where, ρ_{fuel} denotes the density of the fuel and $\dot{\mathcal{V}}_{\text{fuel}}$ is the volume flow rate of fuel into the engine. Note, ρ_{fuel} should be calculated based on the inlet temperature and pressure. The volume flow rate is simply the volume of fuel consumed divided by the time interval of the experiment.

(c) Mechanical Efficiency

The mechanical efficiency of the engine is defined as the ratio of the usable mechanical power over the total power (sum of the brake power plus the power required to overcome friction and inertia in the system),

$$\eta_{\text{mech}} = \frac{\dot{W}_b}{(\dot{W}_b + \dot{W}_f)}, \tag{6}$$

where $\dot{W}_f$ denotes the rate of energy lost due to friction and inertia of the moving piston. In the laboratory, the torque required to run the engine "dry", i.e., to drive the engine without combustion, is measured, and given by τ_d. Therefore, mechanical losses can be calculated as

$$\dot{W}_f = \tau_d\, \omega. \tag{7}$$

(d) Heat Loss

The heat rejected or lost to the surrounding air, $\dot{Q}$, can then be calculated from the First Law of Thermodynamics, which states that the rate of energy into the system must equal the rate of work done by the system plus the heat lost to the surroundings. Mathematically, we can write

$$\dot{E}_{\text{in}} = \dot{W}_b + \dot{W}_f + \dot{Q}. \tag{8}$$

The above equation is rearranged to yield the heat lost (also referred to as the miscellaneous loss rate),

$$\dot{Q} = \dot{E}_{\text{in}} - (\dot{W}_b + \dot{W}_f). \tag{9}$$

Note, the quantity $\dot{Q}$ as calculated from equation (9) represents a number of processes including heat rejected to the surrounding air through convection, radiation heat loss to the surroundings, combustion inefficiency, imperfect air-fuel mixing, and strain energy resulting in the expansion of the materials comprising the engine housing, amongst others.

(e) Mean Effective Pressure

The mean effective pressure, p_{me}, is a quantity used to characterize reciprocating engines, and essentially represents the average pressure acting on the piston during one cycle of operation (i.e., over the four strokes of the engine operation). In this manner, p_{me} provides a measure of the capacity of the engine to do work. In other words, the work done per cycle (in SI units of Joules) can be calculated as

$$W = p_{me}\,\mathcal{V}_d, \tag{10}$$

where $\mathcal{V}_d$ denotes the displacement volume of the air (i.e., $\mathcal{V}_d = \mathcal{V}_{\mathrm{BDC}} - \mathcal{V}_{\mathrm{TDC}}$). Note, the displacement volume is specified by the manufacturer, typically in units of liters. The work done per cycle is simply the brake power multiplied by t_c, the time to complete one cycle,

$$W = \dot{W}_b\,(t_c). \tag{11}$$

The time to complete one cycle is dictated by the rotational speed of the crankshaft, ω. However, we cannot forget that, in a four-stroke engine, the crankshaft makes two revolutions per cycle. Therefore, the time to complete one cycle is

$$t_c = 2\,\pi\,n_c\,\omega^{-1}. \tag{12}$$

where n_c denotes the number of revolutions per power stroke (e.g., $n_c = 2$ in a four-stroke engine), and the factor of $2\,\pi$ is needed to convert from radians to revolutions. Note, ω in equation (12) MUST be expressed in proper SI units of rad/sec. Substituting equation (12) for t_c and equation (2) for $\dot{W}_b$ into equation (13) gives

$$W = \dot{W}_b\left(\frac{2\,\pi\,n_c}{\omega}\right) = \tau\,2\,\pi\,n_c. \tag{13}$$

Equating (13) and (10) and rearranging for the mean effective pressure yields

$$p_{me} = \frac{\tau\,2\,\pi\,n_c}{\mathcal{V}_d}. \tag{14}$$

Theoretical Behavior (Otto Cycle)

It is instructive to compare the performance measurements of an actual IC engine to those of an *ideal* IC engine. Figure 3 illustrates the difference between the actual engine system compared to that of the ideal scenario. The following air standard assumptions are made in the ideal system:

→ air is the working fluid, and considered to be an *ideal gas*,

→ all of the processes (in each stroke of the cycle) are internally *reversible*,

→ the combustion process is replaced by a simple heat addition process, and

→ the exhaust process is replaced by a simple heat rejection process.

In addition, the ideal system utilizes constant specific heat values determined at standard room temperature (T=25°C).

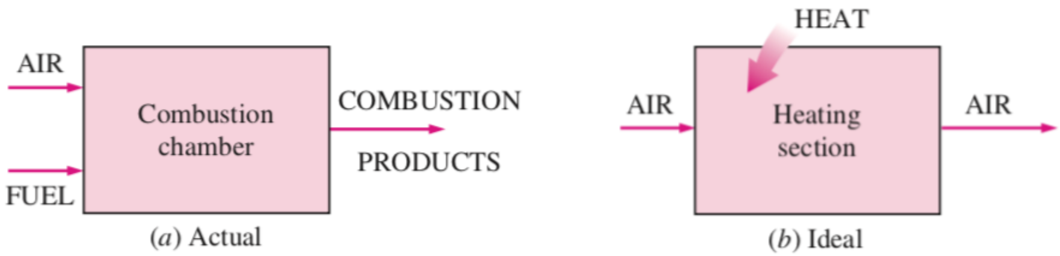

Figure 3: Comparison of the actual engine system to the ideal system.

The four strokes in an IC engine can be modeled ideally as an *Otto Cycle*. To do this, we recall the pressure-volume and temperature-entropy diagrams of an Otto cycle, as shown in Figure 4. The four-strokes of the engine are modeled with the following idealized processes:

(i) intake: isentropic compression ($s_1 = s_2$),

(ii) compression: constant-volume heat addition ($v_2 = v_3$),

(iii) power: isentropic expansion ($s_4 = s_3$),

(iv) exhaust: constant-volume heat rejection ($v_4 = v_1$),

where s and v denote the entropy and specific volume, respectively. The pressure-volume diagram of an actual IC engine is shown in Figure 5 for comparison.

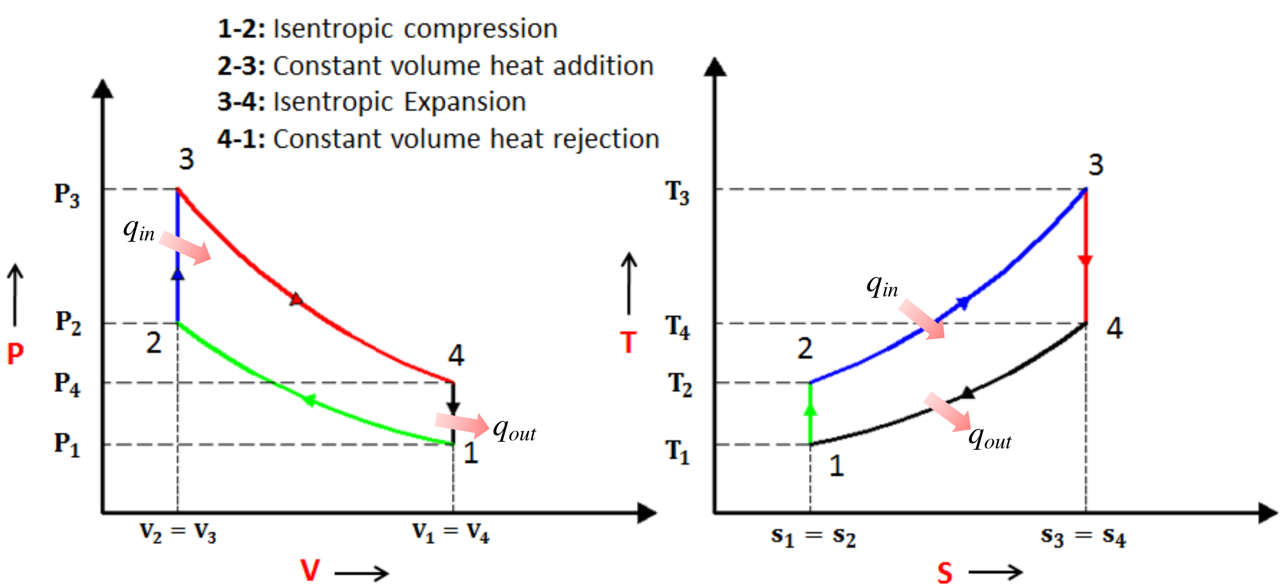

Figure 4: P-$\mathcal{V}$ and T-S diagrams of the Otto Cycle.

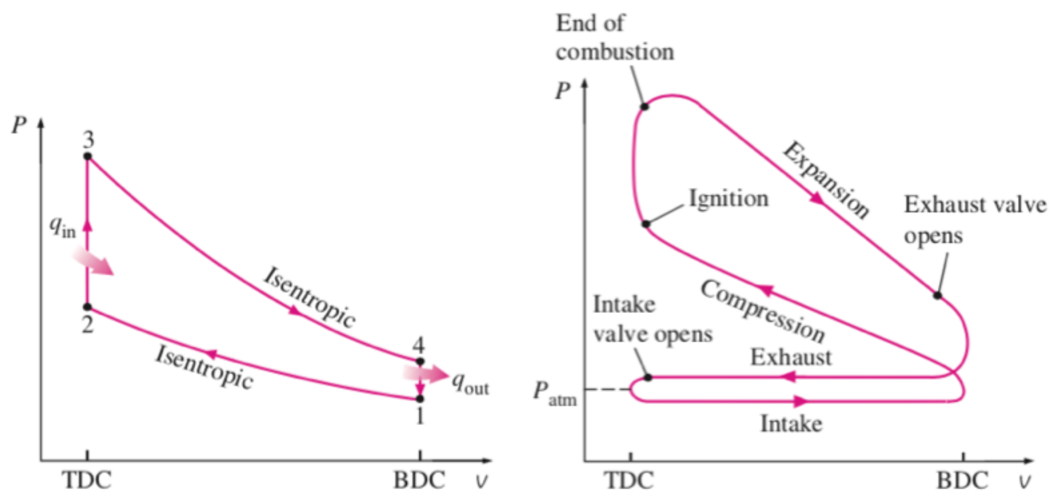

Figure 5: P-$\mathcal{V}$ diagram of the Otto cycle (left) compared to that of an actual IC engine (right).

(a) Ideal Thermal Efficiency

Thermodynamic analysis of the Otto cycle reveals that the thermal efficiency is

$$\eta_{\text{Otto}} = 1 - \frac{1}{r^{k-1}}, \tag{15}$$

where r is the compression ratio and k $(= C_p/C_v)$ is the ratio of the specific heats. In equation (15), we will use the value of k based on the measured exhaust temperature assuming standard air as the working fluid. Of course, this assumption is not quite correct, because the exhaust includes gases other than air, like CO and NO_x. The value η_{Otto} from equation (15) can be compared to that measured from equation (3) to determine how close the actual engine efficiency is to the ideal case.

(b) Ideal Work

For the ideal system (no losses), the First Law of Thermodynamics states that the rate of work done by the system is equal to the difference between the rates of heat transfered into and out of the system,

$$\dot{W}_I = \dot{Q}_{\text{in}} - \dot{Q}_{\text{out}}, \tag{16}$$

where the subscript I denotes "ideal". We assume that all of the heat available in the combustion of the fuel is transfered to the system between states 2–3 of the Otto cycle. Thus,

$$\dot{Q}_{\text{in}} = \dot{E}_{\text{in}}, \tag{17}$$

where the right hand side of the above equation is given by the expression in equation (4). This means we only need to calculate the the ideal rate of heat rejected from the system, $\dot{Q}_{\text{out}}$. The steps required to do this are outlined below.

i. Heat Added (q_{in}) and Heat Rejected (q_{out})

For an ideal system, heat transfer is given by the difference in internal energy (u) between the two thermodynamic states before and after the heat addition/rejection. Furthermore, for an ideal gas in a constant-volume process (as is the case in the Otto cycle), the difference in internal energy between two states is equal to the difference in temperature at those two states. For the Otto cycle then, we can write

$$q_{\text{in}} = u_3 - u_2 = C_v\,(T_3 - T_2) \tag{18}$$

$$q_{\text{out}} = u_4 - u_1 = C_v\,(T_4 - T_1), \tag{19}$$

where C_v is the specific heat at constant volume. Note, the quantity q in equation (18) denotes the heat per unit mass and has typical SI units of J/kg. In order to obtain the heat transfer rate, we need to multiply q by the mass flow rate of air through the system. This was NOT measured experimentally, but can be calculated as described below.

ii. Mass Flow Rate of Air ($\dot{m}_{\text{air}}$)

The mass flow rate of air into the system can be determined from the mass of air (m_{air}) drawn into the cylinder during the intake stroke divided by the time to complete one cycle (t_c),

$$\dot{m}_{\text{air}} = \frac{m_{\text{air}}}{t_c}, \tag{20}$$

where t_c is obtained from the measured rotational speed of the crankshaft as given by equation (12). The mass of air in the cylinder is given by

$$m_{\text{air}} = \rho_{1_{\text{air}}}\,\mathcal{V}_{\text{BDC}}, \tag{21}$$

where $\rho_{1_{\text{air}}}$ denotes the density of the air at state 1 (i.e., the inlet), and $\mathcal{V}_{\text{BDC}}$ is the total volume of the cylinder. The air density at the inlet is calculated using the ideal gas law based on the measured inlet temperature (T_1) and pressure (P_1).

Using the definition of the compression ratio, r, in equation (1) and the definition of the volumetric displacement, $\mathcal{V}_d = \mathcal{V}_{\text{BDC}} - \mathcal{V}_{\text{TDC}}$, we can write

$$m_{\text{air}} = \rho_{1_{\text{air}}}\,\mathcal{V}_d\,\left(\tfrac{r}{r-1}\right). \tag{22}$$

Finally, the mass flow rate of air into (and out of) the system can be written by substituting equation (22) above back into equation (20),

$$\dot{m}_{\text{air}} = \rho_{1_{\text{air}}}\,\mathcal{V}_d\,\left(\tfrac{r}{r-1}\right)\,(2\,\pi\,n_c)^{-1}\,\omega. \tag{23}$$

iii. Heat Transfer Rates ($\dot{Q}_{\text{in}}$ and $\dot{Q}_{\text{out}}$)

The rates of heat transfer into and out of the system are calculated from

$$\dot{Q}_{\text{in}} = \dot{m}_{\text{air}}\,(u_3 - u_2) \tag{24}$$

$$\dot{Q}_{\text{out}} = \dot{m}_{\text{air}}\,(u_4 - u_1)\,. \tag{25}$$

We are interested in calculating $\dot{Q}_{\text{out}}$ in order to determine the ideal work rate as given in equation (16). To do this, we need to determine the internal energy at state 1 and state 4.

iv. Internal Energy at State 1 (u_1)

Since the states of the inlet and outlet are defined (via the measured temperatures and pressures), we can use the thermodynamic table of the ideal-gas properties of air to calculate the internal energy at the inlet, u_1.

v. Internal Energy at State 4 (u_4)

In order to define the thermodynamic state of the air at point 4 in the Otto cycle, we need to use our knowledge of the idealized processes in the Otto cycle as listed earlier. The step are:

→ Define state 2 based on the measurements of T_1 and P_1. Specifically, use the relationship $T\,v^{k-1}$ = const for an adiabatic compression, along with knowledge of the compression ratio, to determine T_2. Here, v is the specific volume. Also, use the fact that the mass must remain constant, along with the ideal gas law to determine P_2. Then use the thermodynamic table of the ideal-gas properties of air to calculate u_2.

→ Determine u_3 from equation (17) based on the calculated values of u_2 and $\dot{E}_{\text{in}}$. Since process 2–3 is assumed to be constant-volume heat addition, $v_3 = v_2$. Finally, T_3 can be found from the thermodynamic table of the ideal-gas properties of air based on u_3 and v_3.

→ Process 3–4 is assumed to be an isentropic expansion. The specific volumes v_3 and v_4 are related by the compression ratio. Therefore, we can find v_4 from v_3. And, we can find T_4 from the relationship $T\,v^{k-1}$ = const using the calculated value of T_3. Finally, u_4 can be found from the thermodynamic table of the ideal-gas properties of air based on v_4 and T_4.

Once you know u_4, you can plug this value into the equations given above to calculate $\dot{Q}_{\text{out}}$ and then $\dot{W}_I$ from equation (16).

Appendix — Experimental Setup

The experimental setup used in the lab is shown in Figure 6. The important specifications of the engine that you tested in the lab are listed in Table 1. In all cases the engine is run with the throttle fully-open. The engine speed (i.e., rotational speed of the crankshaft) is controlled by applying a braking torque to the crankshaft by means of an electric dynamometer. Note, the density and lower heating value of the fuel can be looked up in thermodynamic tables based on the inlet temperature. The specific heat ratio (used in the theoretical analysis) is based on the average exhaust gas temperature, assuming standard air conditions.

Table 1: Specifications of the experimental setup

engine type :	single-cylinder, four-stroke, spark ignition IC
fuel type :	91 octane gasoline
volumetric displacement, $\mathcal{V}_d$:	0.148 L
compression ratio, r :	7 to 1
bore :	65.1 mm
stroke :	44.4 mm
fuel density, ρ_{fuel} :	726 kg/m^3
fuel lower heating value, LHV :	44.0 MJ/kg
specific heat ratio of standard air, k $(= C_p/C_v)$:	1.33

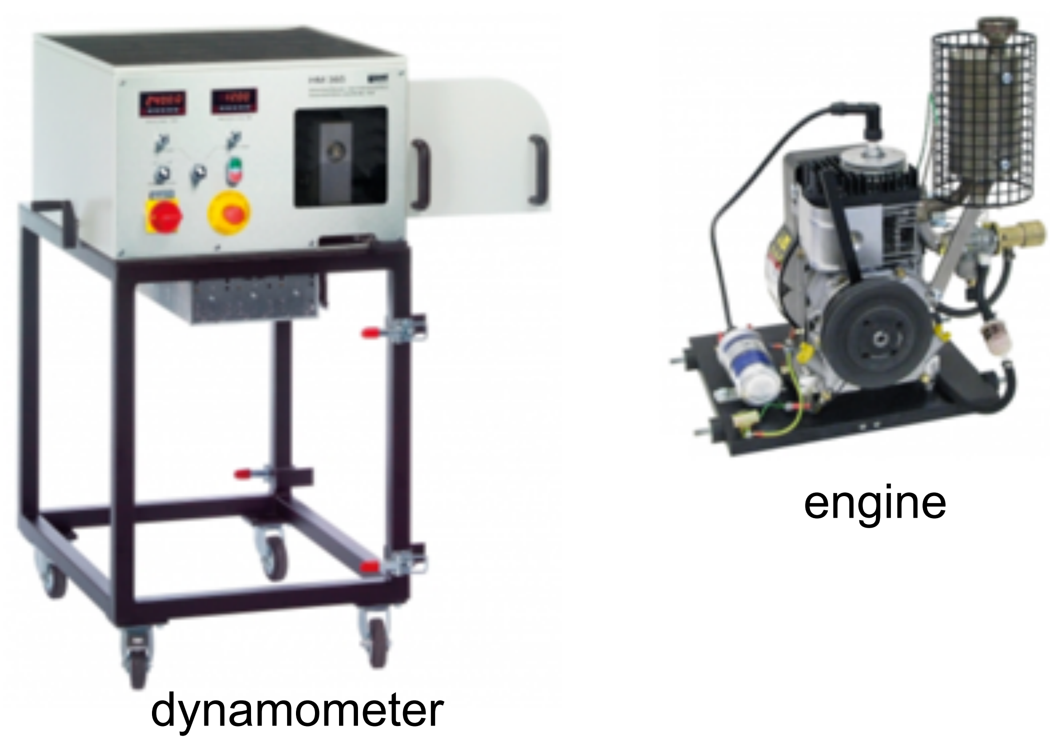

Figure 6: Experimental setup to measure the performance of a four-stroke, single-cylinder, IC engine using a dynamometer. The dynamometer shaft and engine crankshaft are connected via a belt and pulley system (not shown).

Appendix — Bomb Calorimeter

The heating value of a fuel, such as that used in an IC engine, can be determined empirically using a bomb calorimeter device, as shown in Figure 7. In this device, a combustible substance is placed inside a steel container, referred to as the "bomb". The steel bomb is submerged in an insulated water bath. The water inside the bath is stirred to ensure a homogeneous water temperature. The water bath temperature is measured with a thermometer or thermocouple. The substance inside the "bomb" is ignited via an ignition coil. Finally, the increase in water temperature is measured during combustion.

The Higher Heating Value (HHV), also referred to as the Gross Heating Value, of the combustible substance is calculated by applying the First Law of Thermodynamics to the closed system. The Lower Heating Value (LHV), also referred to as the Net Heating Value, is obtained by subtracting the latent heat of vaporization of water from the calculated HHV. This is done, because part of the chemical energy stored in the fuel is used to produce water vapor from the water in the fuel during combustion, and thus cannot be used to produce mechanical work.

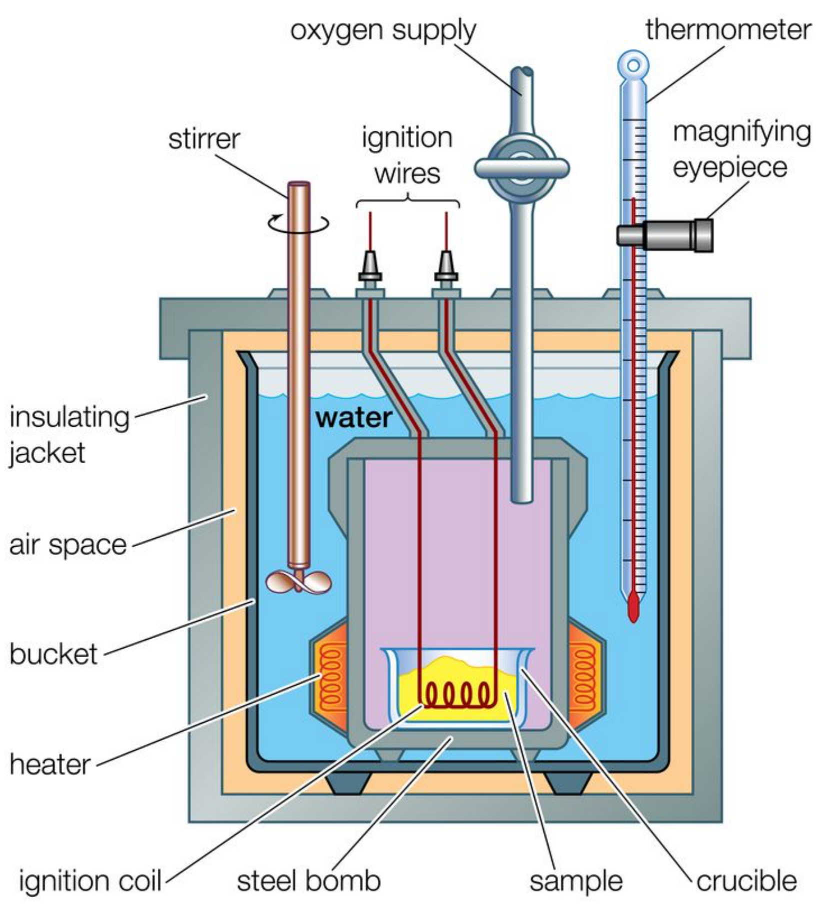

Figure 7: Schematic of a bomb calorimeter used to measure the Higher Heating Value (HHV) of a combustible substance.