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<u>*Statement of Originality*</u>*: "The title page containing the names above asserts that this is a wholly original work by the authors, and any shared and external contributions to this work are documented within."*

Introduction

In this laboratory exercise, the efficiency of a four-cylinder spark-ignition engine was analyzed. A 4.2 L V6 Ford engine was run at wide-open throttle with three different loads placed on it. As a result of the three loads that are acting on the engine, the engine was run at: 1320 rpm, 2600 rpm, and 4170 rpm. To acquire the preliminary data for each loading scenario of the engine, a dynamometer is used as it can produce values for engine speed, torque, power, fuel flow rate, and airflow rate in the engine. Dynamometers are a common tool used to extract power, torque, pressure, temperature, airflow, and much more data when working with rotating shafts. Typically, they are used in the engine design and manufacturing process. They can provide data that can assist with the calibration of engine management controllers and provide insight on combustion behaviour within the engine [1]. Through manipulation of the provided data, the values outlined in *Table 1* were calculated. Each value in *Table 1* contains a brief description of what the values mean in terms of this application, along with the general equation used to calculate each number.

Table 1: Table presenting the equations used in the analysis, along with the description of the values they were used to find.

Title	Equation	Description
Indicated Thermal Efficiency	$\eta_{th,i} = \frac{\dot{W}_{i,g}}{\dot{Q}_{in}} \quad (1)$	A ratio of indicated power generated by the engine with respect to energy supplied to the engine [2].
Brake Thermal Efficiency	$\eta_{th,b} = \frac{\dot{W}_b}{\dot{Q}_{in}} \quad (2)$	The ratio of brake power to input fuel energy [2].
Mechanical Efficiency	$\eta_{th,i} = \frac{\dot{W}_b}{\dot{W}_{i,g}} \quad (3)$	A comparison of indicated power to brake power [3].
Indicated Mean Effective Pressure	$imep = \frac{\dot{W}_{i,g} \cdot n_R \cdot 60}{V_d \cdot N} \quad (4)$	A constant pressure value that would produce the same power per cycle if it acted on the piston for the power stroke. Uses the indicated power value [3].
Brake Mean Effective Pressure	$bmep = \frac{\dot{W}_b \cdot n_R \cdot 60}{V_d \cdot N} \quad (5)$	A constant pressure value that would produce the same power per cycle if it acted on the piston for the power stroke. Uses the brake power value [3].
Equivalence Ratio	$\phi = \frac{AF_s}{AF} = \frac{15}{\left(\frac{\dot{m}_a}{\dot{m}_f}\right)} \quad (6)$	The stoichiometric air to fuel ratio divided by the actual fuel to air ratio [3].

All calculations and data manipulation were done in Jupyter Notebook. All resulting values can be four in the *Results* section of this report, and all plots and graphs can be found under the appendices of this report.

Apparatus

The main apparatus used in this laboratory includes a four-stroke spark-ignition engine and a dynamometer to measure engine performance. However, there are supplementary components required to allow for data to be acquired and analyzed to determine specific performance parameters. The specifications of the engine that is being used can be found below in Table *2*.

Table 2: This table displays the specifications and the parameters for the engine used in the laboratory *[3]*.

Engine Specifications	
Manufacturer	Ford Motor Company
Engine Type	4.2L OHV 12-valve V6
Horsepower (SAE net hp @ rpm)	202 @ 4,800
Torque (lb.-ft. @ rpm)	252 @ 3,400
B – Bore	96.8 mm
L – Stroke	95.0 mm
l – Connecting Rod Length	154.7 mm
a – Crank Radius	47.5 mm
r – Compression Ratio	9.2:1
V_d – Displacement Volume	699.1 cm^3
V_c – Clearance Volume	85.3 cm^3
* Data obtained from Ford 2004 Workshop Manual and the Ford Motor Company	

This engine is being tested by a SuperFlow 902 hydraulic dynamometer. This dynamometer applies a water brake and load cell to calculate torque, angular velocity, and power. A SuperFlow Airflow meter is used to measure mass airflow into the engine, and a Fuel flow rate meter is used to measure the fuel flow into the engine. This allows the exact fuel-air mixture to be controlled and monitored. The fuel used to power the engine is 87-octane gasoline with a heating value of 44.51 MJ/kg and a specific gravity of 0.74. To provide measurements for cylinder pressure, a Kistler 6117B piezoelectric spark plug integrated pressure transducer, which is a pressure sensor connected to the spark plug of the engine. Accompanying the spark plug is a Kistler 5010B charge amplifier to amplify the signal from the transducer to the Tektronix Dual Channel Oscilloscope, providing data that can be processed and analyzed. Finally, a magnetic crankshaft position sensor is used to determine the position of the crankshaft by applying the knowledge that there are 3600 teeth with one missing at the top center of the piston stroke.

Results

Shown below are the results from the simple engine efficiency analysis performed after the experimental data was obtained. The different calculated values can be found under their respective sub-headings in tabular form.

Dynamometer Data

The dynamometer data that was obtained from running the engine under the three different loads can be found below in *Table 3.*

Table 3: Presented in this table is the recorded dynamometer data for each of the three loads.

Load Number	Engine Speed [rpm]	Torque [N-m]	Power [kW]	Fuel Flow Rate [kg/s]	Air Flow Rate [kg/s]
1	1320	284.72	38.78	3.43e-3	4.22e-2
2	2600	311.84	85.01	7.19e-3	9.00e-2
3	4170	311.84	130.50	1.27e-2	1.46e-1

The pressure transducer calibration value is 10.2 bar/V. This value must be in place to change the pressure data from units of 'volts' to 'bar'. Throughout our analysis, SI units were worked with, so the pressure then had to be converted from 'bar' to 'kilopascals'.

Power

The gross indicated rate of work was found using the area under the pressure-volume curve for each load, as seen in *Appendix C*. Table *4* displays the results from the calculated indicated power for each load tested in this laboratory. These values are later used to calculate indicated thermal efficiency and indicated mean effective pressure.

Table 4: Gross indicated rate of work for each load tested.

Load Number	Indicated Power [KW]
1	43.95
2	98.60
3	167.19

Rate of Heat into the System

The calculated rate of heat into the engine for each load tested can be seen in *Table 5.* The values are later used to calculate thermal efficiencies.

Table 5: Rate of heat into the engine for each load tested.

Load Number	Rate of Heat In [KW]
1	152.49
2	319.81
3	563.39

Indicated Thermal Efficiency

The calculated indicated thermal efficiency for each load tested can be seen in *Table 6*. These values were calculated by applying *Equation 1* as seen in *Table 1*.

Table 6: Indicated thermal efficiency for each load tested

Load Number	Indicated Thermal Efficiency [%]
1	28.82
2	30.83
3	29.67

Brake Thermal Efficiency

The calculated brake thermal efficiency for each load can be seen in *Table 7.* These values were calculated by applying *Equation 2* as seen in *Table 1*.

Table 7: Brake thermal efficiency for each load tested

Load Number	Brake Thermal Efficiency [%]
1	25.43
2	26.58
3	23.16

Mechanical Efficiency

The calculated mechanical efficiency for each load can be seen in *Table 8.* The values were calculated by applying Equation 3 as seen in *Table 1*.

Table 8: Mechanical efficiency for each load tested

Load Number	Mechanical Efficiency [%]
1	88.22
2	86.22
3	78.06

Indicated Mean Effective Pressure

The indicated mean effective pressure of the engine was calculated using *Equation 4* specified in *Table 1.* This equation uses the engine's displacement volume, speed, and indicated power to yield a constant pressure value that would produce the same power per cycle if it acted on the piston for the power stroke. For this laboratory, an indicated mean effective pressure value of 1084.87 kPa was calculated. Note that the indicated mean effective pressure was only calculated for when the engine was under load two.

Brake Mean Effective Pressure

The brake mean effective pressure of the engine was calculated using *Equation 5* specified in *Table 1.* This equation uses the same variables as the indicated mean effective pressure calculation, with the replacement of indicated power with brake power. By applying this equation with the values recorded in this laboratory, a brake mean effective pressure of 935.38 kPa was calculated. Note that the brake mean effective pressure was only calculated for the second loading scenario.

Specific Heat Ratio

The experimental specific heat ratios for each of the three loading scenarios can be found below in *Table 9*.

Table 9: The calculated specific heat ratios for each respective load.

Load	Specific Heat Ratio
1	1.40
2	1.35
3	1.32

Equivalence Ratio

The equivalence ratio for each loading scenario was obtained using *Equation 6*, which can be found in *Table 1*. These calculated values can be found below in Table *10. Table 11* shows the experimental specific heat ratio values and also gives the percent that they stray from a theoretical value of 1.3.

Table 10: Displays the calculated equivalence ratios for each load.

Load	Equivalence Ratio
1	1.22
2	1.20
3	1.30

Table 11: For each load, the experimental specific heat ratios are compared to the percentage the experimental values stray from the theoretical specific heat ratios.

Load Number	Experimental Specific Heat Ratios	Percent Deviated from Theoretical Value [%]
1	1.40	7.70
2	1.35	3.85
3	1.32	1.54

Discussion

As displayed in Table 4, the indicated thermal efficiencies for loads, 1, 2, and 3 were 28.82%, 30.83%, and 29.67% respectively. The thermal efficiency is defined by the indicated power divided by the rate of heat into the cylinder. It is expected that load 2, which should correspond to peak torque RPM (but does not, given the lab manual says that peak torque RPM is 3400), would have the highest thermal efficiency. At peak engine torque, usually, the engine's volumetric efficiency is near its maximal value, thus the highest density of the air/fuel mixture is achieved, and the combustion process yields the highest dynamic pressure. The RPM where peak torque and volumetric efficiency occurs is dependent highly on parameters such as intake port sizes, compression ratio, air-fuel mixture, and any parameter that will change air intake/combustion pressure.

For the data obtained in this lab, the highest work per cycle is achieved by load 3, which is contrary to what is expected given that load 2 should correspond with peak torque. However, given that load 2 (nor load 3) is at peak torque RPM of 3400, the reason that the thermal efficiency is highest for load 2 is a result of significantly lower heat input required.

The brake thermal efficiencies are shown in *Table 7*, where it can be seen the values for loads 1, 2, and 3 are 25.43%, 26.58%, and 23.16%, respectively. The brake thermal efficiencies follow a similar pattern as the indicated thermal efficiencies, aside from the values for load 1 and load 2 being ~3.5% lower and the load 3 efficiency being ~6% lower. This is expected as the brake thermal efficiency accounts for friction. Further, it is logical that the load 3 thermal efficiency would be lowest as friction would be most significant at higher RPM.

Using the Ferguson textbook applet for the four-stroke fuel-air Otto cycle, a MATLAB code was used to return the indicated mean effective pressure for a four-stroke Otto cycle given the engine characteristics used in this laboratory. Upon running the MATLAB code, an indicated mean effective pressure of 1489.5 kPa was obtained. This value is much higher than the value of 1084.87 kPa obtained in the laboratory's results, meaning there are losses in the real-world laboratory engine test not seen in an ideal system. Losses such as incomplete fuel combustion and heat losses contribute to a decrease in the overall indicated power, therefore causing a decrease in indicated mean effective pressure as explained by the relationship in *Table 1*, *Equation 4*. Losses during engine performance do not allow for the engine to convert all reactants into products with 100% efficiency. This means the ideal maximum indicated power can never be achieved in practice but can in theoretical calculations. This laboratory exhibits this theory, yielding a theoretical indicated mean effective pressure that is 37% larger than the practical indicated mean effective pressure calculated using lab data.

The experimental ratios of specific heat can be found in *Table 9*. The calculated percentage of error for the experimental data compared to the theoretical data can be found in *Table 11.* From the data in *Table 11*, as the load acting on the engine decreases, or as the engine speed increases, the experimental specific heat ratio values converge towards the theoretical specific heat ratio value of 1.3 for this engine. This results in the most accurate, and lowest specific heat ratio occurring when the engine is operating at load 3. This trend means that the energy required to raise the temperature of the air-gas mixture increases as the engine speed goes up.

There are several reasons why the specific heat capacity is inversely proportional to the engine speed. First, as the RPM of the engine increases, the equivalence ratio does not remain constant. Because the fuel is likely to have a different heat capacity than the air/exhaust, the changing ratio of the mixture would affect the specific heat capacity. However, the most likely reason for the decrease in the specific heat capacity is the rise in the temperature of the engine. The specific heat capacity of a material is the function of enthalpy/internal energy and the temperature of the material. Furthermore, the temperature of the material and its heat capacity are inversely proportional. Because the temperature of the engine increases with higher speed, the specific heat ratio also decreases, as seen in the expansion stroke.

One reason that the calculated specific heat ratio values have error is that the theoretical value is assumed to be calculated from the engine operating at standard temperature and pressure, but the temperature and pressure would not be perfectly standard so the heat ratios would in result be influenced.

One potential source of error that would cause the computed results in this lab to have an uncertainty compared to 'perfect engine operation' could be that while the engine is running not all combustion products are exhausted from the engine before the next intake stage begins. This would result in less air intake and therefore not all the combustion products combusting. Similarly, the air-fuel mixture cannot be expected to be homogenous, especially at high engine speeds, further decreasing the combustion efficiency. This can be seen in the variation of peak pressures for cycles under the same load, most likely caused by the errors detailed above. While this is to be expected and is a part of the real-world Otto cycle, the analysis of each load was done using only 5 cycles, meaning that any variation between the cycles has a significant impact on the results. Analysis using more cycles per load would reduce the impact that these variations would have on the analysis and results.

A final source of error in the power calculations is in the process of selecting the areas used to compute the net work per cycle. Given that the plots, after interpolation, are tabulated per degree of CA, the areas were defined by the finite CA data points. This could result in slight over/underestimation of the work per cycle and thus the power.

Conclusions and Recommendations

This laboratory tested the performance of a four-cylinder engine under three different loads, using a dynamometer to record engine speed, torque, power, fuel flow rate, and airflow rate. This data was then processed and analyzed to evaluate engine power, thermal and mechanical efficiencies, indicated mean effective pressure, and brake mean effective pressure. From this laboratory, it can be concluded that the brake mean effective pressure is a more realistic calculation of engine performance than indicated mean effective pressure. As brake mean effective pressure is calculated using the engine's brake power, losses due to friction are accounted for thus yielding more applicable data resembling real-world scenarios. Although brake mean effective pressure considers friction losses, additional losses such as heat loss and incomplete combustion are not accounted for. It is because of these losses that a real-world engine performance test will never yield indicated mean effective pressure and brake mean effective pressure values of an ideal system. It can also be concluded that in theory, the peak engine torque correlates to the highest efficiencies as it achieves the highest dynamic pressure and therefore the most complete fuel combustion. However, this laboratory yielded results that did not correlate to this, showing there are other parameters such as combustion ratio, air-fuel mixture, and size of intake port that affect the engine performance.

It is recommended that the engine be run at an rpm that results in maximum torque output to confirm that it is the most efficient engine speed. Comparing these efficiencies at each load will then confirm that maximum torque output correlates with maximum thermal and mechanical efficiency. An additional recommendation is to increase the number and size of the loads applied to the engine, to further understand the relationship between rpm and mechanical/thermal efficiencies.

References

[1] T. Lish, "Setra," Setra, 28 September 2015. [Online]. Available: https://www.setra.com/blog/test-and-measurement-dynamometer. [Accessed 12 November 2021].

[2] ClubTechnical, "Engine Performance Parameters," Club Technical Mechanical Engineering Blog, 23 March 2019. [Online]. Available: https://clubtechnical.com/engine-performance-parameters. [Accessed 12 11 2021].

[3] G. Ciccarelli, "MECH 435 - IC Engine Laboratory," Queen's University - Faculty of Applied Science - Department of Mechanical and Materials Engineering, 2021.

Appendix A – Jupyter Notebook Functions

```
def interpolation(sortedData):
    iPressure = np.zeros((5,720))
    for k in range(5):
        a = sortedData[k][0]
        p = sortedData[k][1]
        for j in range(720):
            for i in range(1, len(sortedData[k][0] - 1)):
                if (a[i-1] <= j) and (a[i+1] >= j):
                    iPressure[k,j] = p[i-1] + ( (j - a[i-1]) * (p[i+1]-p[i-1]) / (a[i+1]-a[i-1]) )
    return iPressure
```

Figure 1: An image of the segment of code that was used to interpolate the pressure data for the 5 cycles in each of the three loads.

```
def volume(theta):
    V=[]
    for i in range(len(theta)):
        s = a*np.cos(theta[i]*(np.pi/180)) + np.sqrt((l**2)-(a**2)*(((np.sin(theta[i]*(np.pi/180)))**2)))
        vol = Vc + (((np.pi)*((B)**2))/4)*(l+a-s)
        V.append(vol)
    return V
```

Figure 2: An image of the segment of code that was used to find the cylinder volume data.

```
W45 = [integrate.simps(mP1[360:540], vTheta[360:540]), integrate.simps(mP2[360:540], vTheta[360:540]), integrate.simps(mP3[36
W23 = [integrate.simps(mP1[180:360], vTheta[180:360]), integrate.simps(mP2[180:360], vTheta[180:360]), integrate.simps(mP3[18
W51 = [integrate.simps(mP1[0:180], vTheta[0:180]), integrate.simps(mP2[0:180], vTheta[0:180]), integrate.simps(mP3[0:180], vT
W12 = [integrate.simps(mP1[540:720], vTheta[540:720]), integrate.simps(mP2[540:720], vTheta[540:720]), integrate.simps(mP3[54

Wnet = []
dWi = []
dQin = []
EtaI = []
EtaB = []
EtaM = []
phi = []


dWb = [52 * 0.7457, 114 * 0.7457, 175 * 0.7457]
dm = [p*(0.488)/(105.41), p*(0.488)/(50.26), p*(0.488)/(28.53)]
dma = [1.177 * 76/2118.88, 1.177 * 162/2118.88, 1.177 * 263/2118.88]

for i in range(3):
    Wnet.append((W45[i] + W23[i]) * 6)
    dWi.append(Wnet[i] * N[i] / 2)
    dQin.append(dm[i] * Qhv)
    EtaI.append(dWi[i] * 100 / dQin[i])
    EtaB.append(dWb[i] * 100 / dQin[i])
    EtaM.append(dWb[i] * 100 / dWi[i])
    phi.append(15*dm[i]/dma[i])
```

Figure 3: This image shows the segment of code used to calculate the numerical values required for the engine analysis.

The first 4 lines of the code is the integration of the cycle. The net work is then calculated by subtracting the area under the compression stroke from the area under the expansion stroke. The third block of code is, in order, the conversion of horsepower to kW and the flow rate of mass of air and fuel at each load. These values are then used to analyze the Otto cycle at each load.

Appendix B – Pressure vs. Crank Angle Plots

The pressure, in kilopascals, plotted against the crank angle, in degrees, for each of the three loads can be found in *Figure 4*, *Figure 5*, and *Figure 6* below. The diagram for each load contains five different plots for five consecutive representative cycles of engine operation (720° of crank rotation).

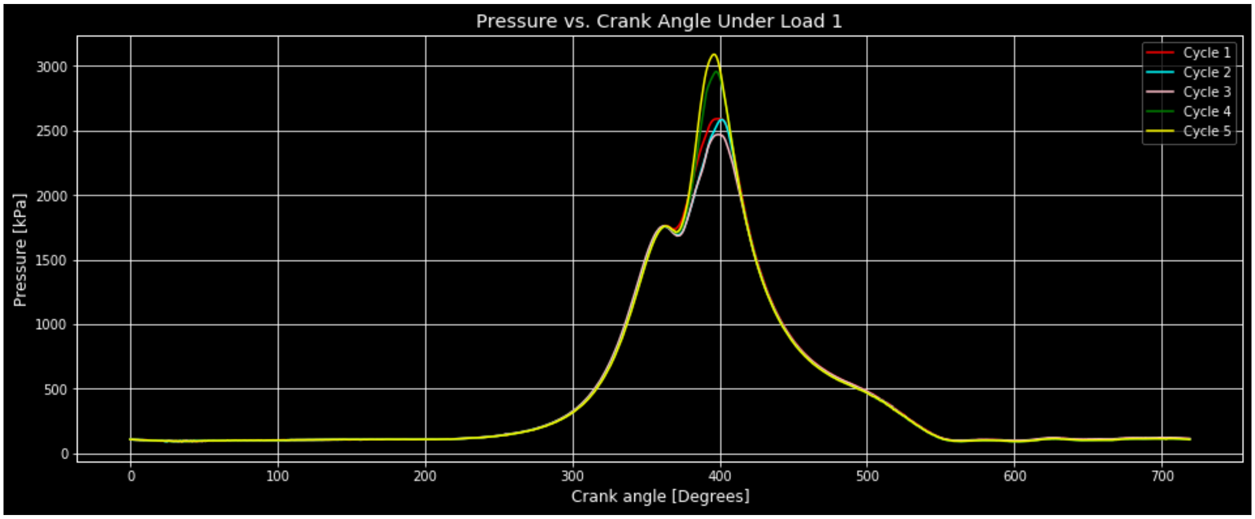

Figure 4: Pressure vs. crank angle plot for 5 consecutive cycles while the engine is under the first load.

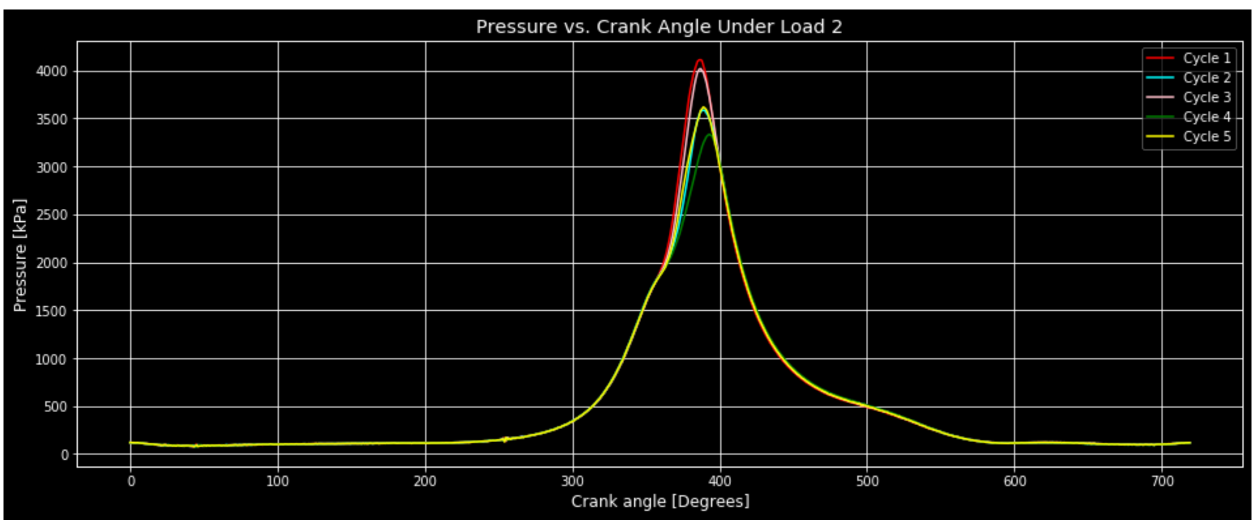

Figure 5: Pressure vs. crank angle plot for 5 consecutive cycles while the engine is under the second load.

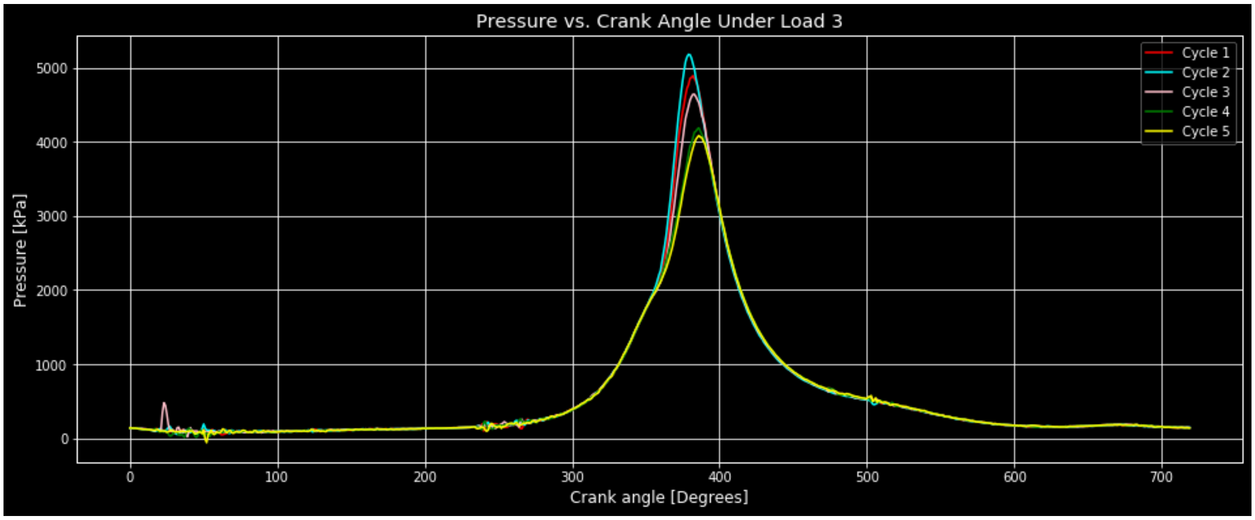

Figure 6: Pressure vs. crank angle plot for 5 consecutive cycles while the engine is under the second load.

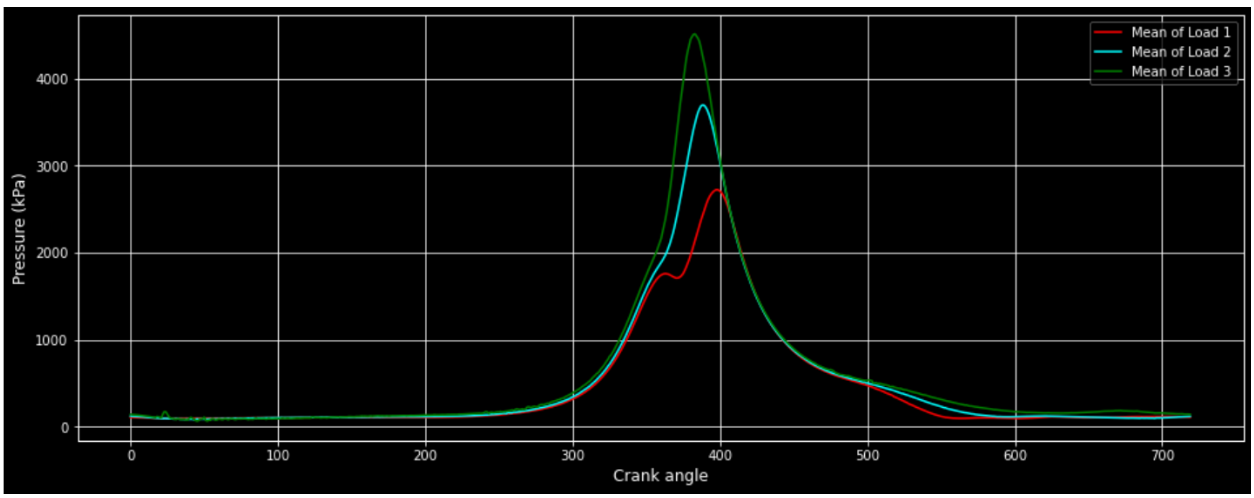

Figure 7: Plot showing the average data from each of the five cycles for each load.

To improve the accuracy of the pressures for each cycle that were used, the average pressure values were interpolated in single integer increments 1-720. Since our data was analyzed in Python, an interpolation function was created, and can be viewed in *Appendix A*, Figure *1*. An interpolation approach resulted in the ability to account for the engine speed variations per cycle [3].

A fourth plot is included in the pressure versus crank angle results, which can be referred to in Figure *7* above. This plot contains the pressures of the five operational cycles averaged for each loading scenario, then the three loading situations were plotted against each other. This plot shows that the average cycle pressures are the highest when the loading is 4500 rpm, and the average cycle pressures decrease as the loading decreases.

Appendix C – Pressure Vs. Volume Plot

Displayed below in Figure *8* is the pressure inside the cylinder plotted against the cylinder volume. This plot shape resembles that of a typical internal combustion engine.

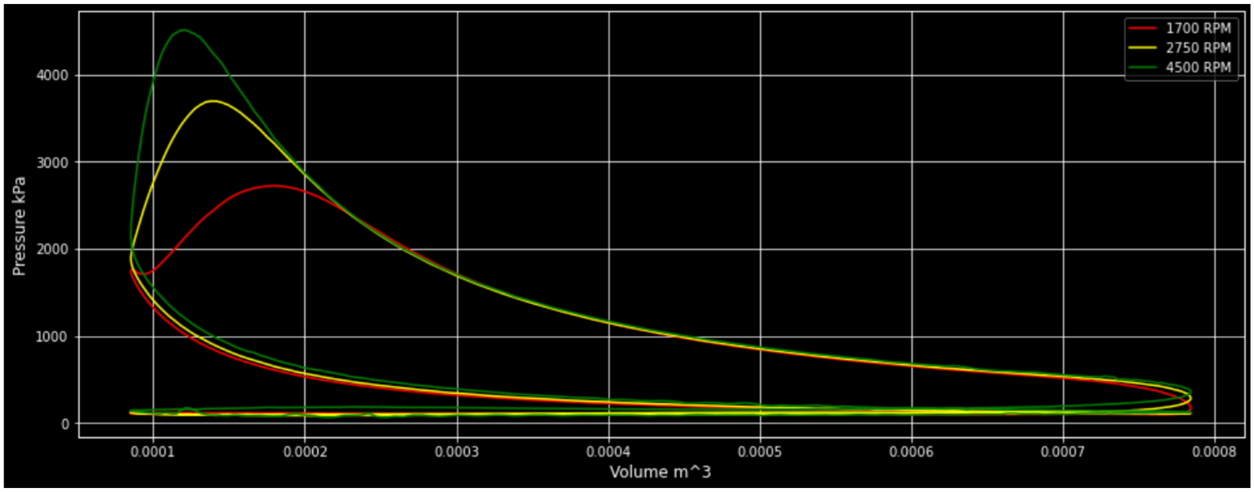

Figure 8: Pressure in kilopascals plotted against the volume on the piston in cubic meters.

To obtain the volume values, a function was created in Jupyter Notebook. An image of this function can be found in *Appendix A*, *Figure 2*.

Appendix D – Logarithmic Plot of Pressure Vs. Volume

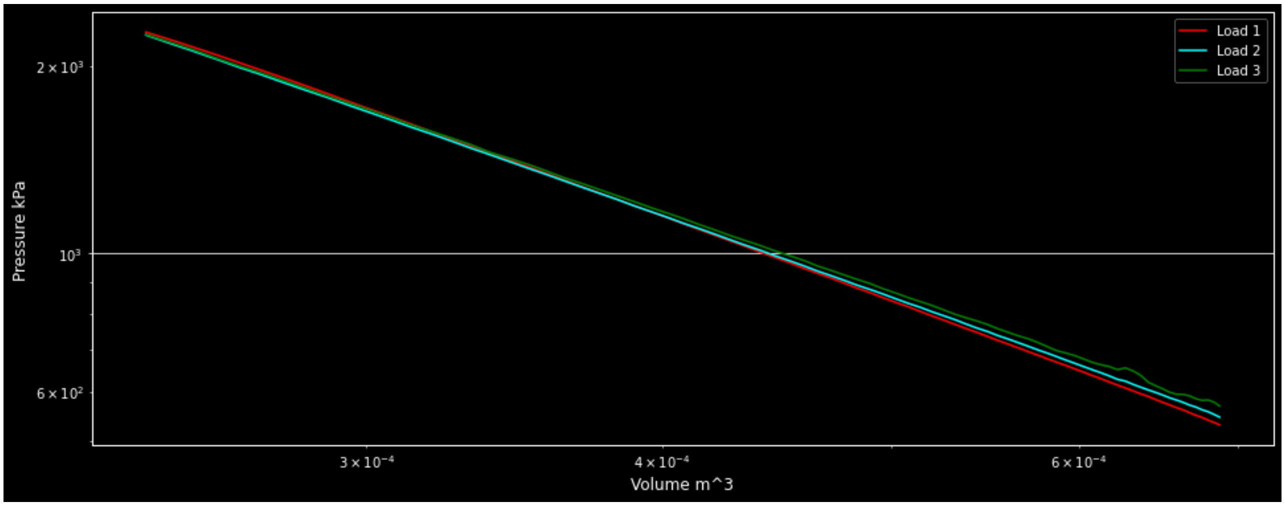

Figure 9: Logarithmic plot of the expansion stroke, pressure in kilopascals vs. volume in cubic meters.