

LEVEL 5 ASSESSMENT SPECIFICATION

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Introduction

This report provides a thermodynamic analysis of the Ford Sigma 1.6L engine, bridging the gap between theoretical models and real-world performance. In Task 1, I developed an idealized Otto cycle simulation using MATLAB, assuming isentropic processes and instantaneous heat addition. Task 2 involves extracting "ground truth" data from an experimental pressure trace to quantify real-world deviations, such as heat transfer and finite combustion duration. Finally, Task 3 uses these findings to refine the initial simulation, creating a high-fidelity model that accurately reflects the engine's mechanical and thermal characteristics.

MatLab Script Used

```
% Task 2: Ford Sigma 1.6L - Advanced Data Analysis
% Extracting Polytropic Indices, Temperature, and Work Done.
% Targets: High-quality visuals and comprehensive PV/T analysis.

% 1. Setup and Data Import
filePath = 'C:\Users\Bigsby\Desktop\Uni
Work\TeamsRips\Thermodynamics\Assignments\Sigma 1.6L.xls';
% We use the range A12:C733 to capture CA, Vol, and Pressure
T = readtable(filePath, 'Sheet', '2000', 'Range', 'A12:C733',
'VariableNamingRule', 'preserve');

CA = T{:, 1};          % Crank Angle (deg)
V_m3 = T{:, 2} * 1e-6; % Volume (cc to m^3)
P_bar = T{:, 3};       % Pressure (bar)
P_Pa = P_bar * 1e5;    % Pressure (Pa)

% 2. Thermodynamic Constants & Temperature Extraction
R = 287;               % Specific gas constant for air (J/kgK)
P_atmo = 1.01325e5;    % Ambient pressure (Pa)
T_intake = 300;        % Estimated intake temperature (K)
V_max = max(V_m3);     % Max volume at BDC
m = (P_atmo * V_max) / (R * T_intake); % Trapped mass (kg)

% Calculate Temperature Trace:  $T = (P \cdot V) / (m \cdot R)$ 
Temp_K = (P_Pa .* V_m3) ./ (m * R);

% 3. Polytropic Index (n) Calculation:  $PV^n = C$ 
% Compression (-140 to -40 deg)
c_idx = (CA >= -140 & CA <= -40);
coeffs_c = polyfit(log(V_m3(c_idx)), log(P_Pa(c_idx)), 1);
nc = -coeffs_c(1);

% Expansion (60 to 140 deg)
e_idx = (CA >= 60 & CA <= 140);
coeffs_e = polyfit(log(V_m3(e_idx)), log(P_Pa(e_idx)), 1);
ne = -coeffs_e(1);

% 4. Work Done Calculation:  $W = \int P \, dV$ 
Work_Ind = trapz(V_m3, P_Pa);

% 5. High-Quality Visualisation
figure('Units', 'normalized', 'Position', [0.1 0.1 0.8 0.8], 'Color', 'w');
```

```

% Plot 1: Annotated Pressure Trace
subplot(2,2,1);
plot(CA, P_bar, 'LineWidth', 2, 'Color', [0 0.45 0.74]); hold on;
[p_max, p_idx] = max(P_bar);
plot(CA(p_idx), p_max, 'ro', 'MarkerFaceColor', 'r');
text(CA(p_idx)+15, p_max, sprintf('Peak: %.1f bar', p_max), 'FontWeight',
'bold');
title('Cylinder Pressure vs Crank Angle');
xlabel('Crank Angle (deg)'); ylabel('Pressure (bar)');
grid on;

% Plot 2: Log-Log P-V Diagram (Critical for Polytropic Analysis)
subplot(2,2,2);
loglog(V_m3, P_Pa, 'LineWidth', 2, 'Color', [0.85 0.33 0.1]);
title('Log-Log P-V Diagram (Polytropic Check)');
xlabel('Volume (m^3)'); ylabel('Pressure (Pa)');
grid on;

% Plot 3: Temperature Cycle
subplot(2,2,3);
plot(CA, Temp_K, 'LineWidth', 2, 'Color', [0.47 0.67 0.19]);
title('Calculated Cylinder Temperature');
xlabel('Crank Angle (deg)'); ylabel('Temperature (K)');
grid on;

% Plot 4: Standard P-V Loop with Work Shading
subplot(2,2,4);
fill(V_m3, P_Pa, 'y', 'FaceAlpha', 0.2, 'EdgeColor', 'k', 'LineWidth', 1.5);
title(['P-V Diagram (Indicated Work: ', num2str(round(Work_Ind,1)), ' J)']);
xlabel('Volume (m^3)'); ylabel('Pressure (Pa)');
grid on;

% 6. Console Output for Report
fprintf('--- Task 2 Analysis Results ---\n');
fprintf('Compression Index (nc): %.4f\n', nc);
fprintf('Expansion Index (ne): %.4f\n', ne);
fprintf('Net Indicated Work: %.2f Joules\n', Work_Ind);
fprintf('Max Cycle Temperature: %.1f K\n', max(Temp_K));

%% Export Task 2 Results to Excel
% This creates a table of your processed data for better graphing in Excel.

% 1. Create a Table for Export
% We include the raw data plus your new calculated Temperature trace

```

```

ExportTable = table(CA, V_m3, P_bar, Temp_K, ...
    'VariableNames', {'CrankAngle_deg', 'Volume_m3', 'Pressure_bar',
    'Temperature_K'});

% 2. Define the output filename
outputFile = 'Ford_Sigma_Analysis_Results.xlsx';

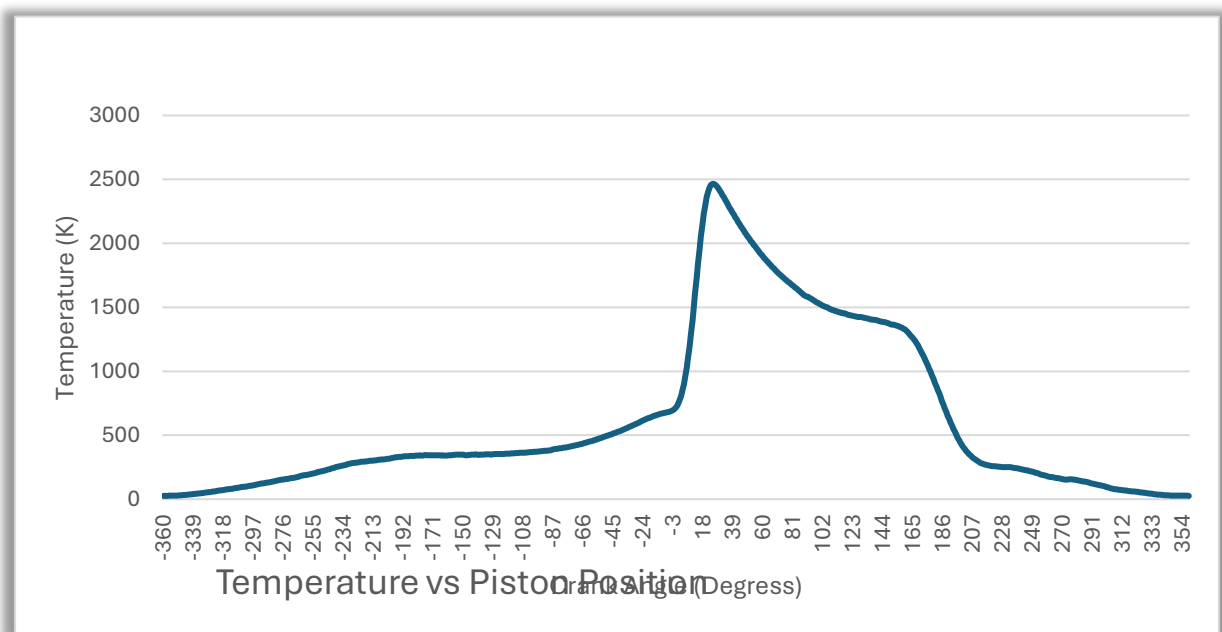
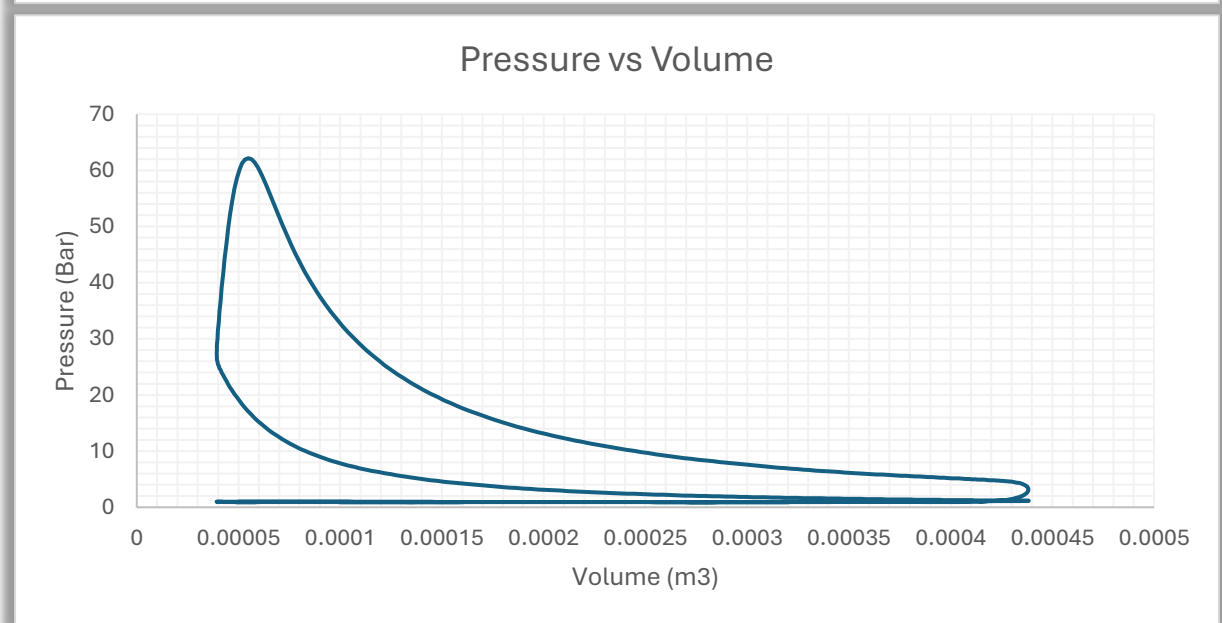
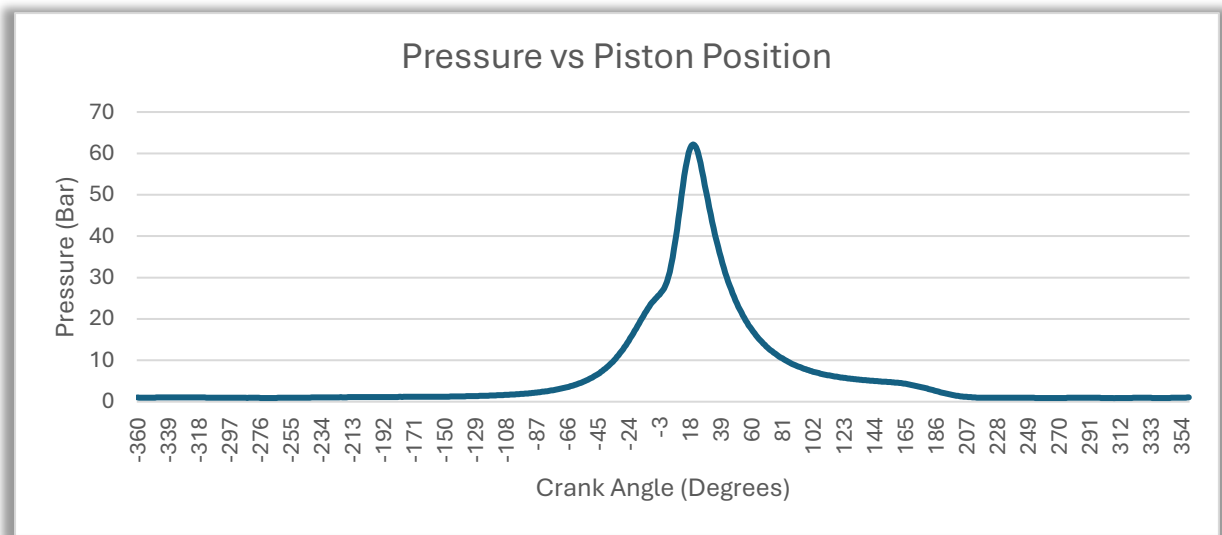
% 3. Write to Excel
writetable(ExportTable, outputFile, 'Sheet', 'Processed_Data');

% 4. Export Summary Metrics (Polytropic Indices and Work)
SummaryLabels = {'Compression_n'; 'Expansion_n'; 'Indicated_Work_J';
    'Max_Temp_K'};
SummaryValues = [nc; ne; Work_Ind; max(Temp_K)];
SummaryTable = table(SummaryLabels, SummaryValues);

writetable(SummaryTable, outputFile, 'Sheet', 'Summary_Metrics');

fprintf('Data successfully exported to %s\n', outputFile);

```



In-Cylinder Pressure and Piston Position

By analysing the raw pressure data from the Ford Sigma 1.6L engine, I can visualize the thermodynamic cycle in real-time. Figure 1 maps this in-cylinder pressure against the crank angle, allowing for a sequential comparison between my practical results and the idealized models developed in Task 1.

Compression and Ignition Advance

Starting at Intake Valve Closing (IVC), the cylinder pressure begins at approximately 1 bar. While my Task 1 model assumed a perfectly insulated adiabatic process, the real-world data reflects significant heat transfer to the cylinder walls, oil, and cooling jacket, preventing the pressure from rising as aggressively as the theoretical maximum. As the piston approaches Top Dead Centre (TDC), the spark plug ignites the air-fuel mixture. Unlike the instantaneous "constant volume" heat addition of my ideal model, the real flame front requires finite time to propagate across the cylinder bore. Unlike the instantaneous 'constant volume' heat addition of my ideal model, the real flame front requires finite time to propagate across the cylinder bore (Heywood, 2018). To prevent 'chasing' the piston on the downward stroke, the engine utilizes ignition advance, firing the spark roughly 10° to 15° before TDC.

Peak Pressure and Mechanical Leverage

The cycle reaches a maximum pressure of approximately 62 bar. Crucially, this peak occurs roughly 15° after TDC (ATDC) rather than at the 0° position. This is a strategic design target; at 0° , the connecting rod has no lever arm, meaning no torque is generated regardless of pressure. By peaking at 15° ATDC, the engine leverages an optimal connecting rod angle to maximize crankshaft rotation. As Heywood (Heywood, 2018) explains, this specific timing is essential for achieving Maximum Brake Torque (MBT), as it represents the ideal compromise between the pressure rise from combustion and the increasing volume of the cylinder during the expansion stroke.

Expansion and Exhaust Blowdown

During the power stroke, the high-temperature gases expand and perform work on the piston, causing a rapid pressure drop. However, the real data shows the exhaust valve opening before the piston reaches Bottom Dead Centre (BDC) at 180° . This "Exhaust Blowdown" allows residual pressure to clear the cylinder, sacrificing a small amount of expansion work to significantly reduce the pumping work required during the subsequent exhaust stroke.

Polytropic Index and Process Analysis

To move beyond the raw pressure trace, I analysed the compression and expansion phases using the polytropic relation, $PV^n = \text{constant}$. By plotting the pressure and volume data on a log-log scale, I could determine the actual slopes for these processes, which serve as a mathematical footprint of the engine's heat transfer. While an ideal Otto cycle assumes adiabatic conditions where the specific heat ratio is approximately 1.4, the real Ford Sigma engine experiences continuous thermal exchange with the cylinder walls and cooling jacket. As noted by Moran and Shapiro (Moran & Shapiro, 2014), the polytropic model is essential in these scenarios as it allows for the description of expansion and compression processes in which heat transfer occurs, providing a more accurate representation of the energy transitions than a purely isentropic model.

The Compression Phase (n_c)

I've isolated the compression data between -140 and -40 degrees, where the valves are closed but combustion has not yet begun. My MATLAB script calculated a compression polytropic index (n_c) of 1.3081. This value, being significantly lower than 1.4, proves that the process is not perfectly insulated. As the piston does work on the charge, the gas temperature rises, causing thermal energy to bleed into the cast metal. This phenomenon is often modelled using the Woschni correlation, which accounts for the convective heat transfer variations driven by gas velocity and pressure changes during the cycle (Woschni, 1967).

The Expansion Phase (n_e)

For the power stroke, I analysed the data between 60 and 140 degrees, resulting in an expansion index (n_e) of 1.3434. Following the explosive combustion event, the temperature difference between the gas and the cylinder walls is at its maximum, driving aggressive heat transfer.

Interestingly, the expansion index is higher than the compression index. This shift occurs because I am no longer dealing with a simple air-fuel mixture, but a complex mixture of high-temperature exhaust products like CO₂ and H₂O. At these extremes, chemical dissociation occurs—where molecules break apart and absorb energy—followed by recombination as the gas cools, which releases heat back into the system. This chemical complexity, combined with the aggressive cooling, dictates the specific 1.3434 slope I extracted from the raw data.

Combustion Timing: Pinpointing SOC and EOC

In my ideal MATLAB model, I assumed heat addition happened instantaneously at exactly 0° TDC. However, the real Ford Sigma pressure trace proves that combustion is a time-dependent chemical reaction. Identifying the Start of Combustion (SOC) and the End of Combustion (EOC) is essential to understanding how effectively the engine converts chemical energy into mechanical work.

Start of Combustion (SOC)

To find the SOC, I looked for the exact crank angle where the real pressure trace visibly breaks away from the smooth, calculated polytropic compression curve. In my analysis, this deviation becomes noticeable roughly 10° to 15° before TDC. This point represents when the heat release from the igniting fuel becomes significant enough to drive the cylinder pressure upward, overcoming the mechanical compression. As Stone (Stone, 2012) explains, there is a distinct period between the spark discharge and the first measurable pressure rise, known as the ignition delay, which is caused by the initial slow growth of the flame kernel.

End of Combustion (EOC)

Pinpointing the EOC is more complex because the piston is already moving downward, which naturally drops the pressure while the remaining fuel is still burning. I identified the EOC by observing the expansion curve and searching for the moment the trace stabilizes into the smooth polytropic decline dictated by my calculated n_e value of 1.3434. In this dataset, that stabilization occurs between 40° and 50° after TDC.

Realizing that the entire combustion event spans over 50 degrees of crankshaft rotation explains why the top of my P-V diagram is a smooth, rounded curve rather than the sharp, aggressive spike seen in the ideal air-standard Otto cycle. While this "time loss" inherently means the engine extracts less theoretical work than a mathematically perfect model, it is a physical inevitability of real-world flame propagation.

Furthermore, this rounding represents more than just a timing offset; it reflects inherent combustion inefficiencies. As Ferguson and Kirkpatrick (Ferguson & Kirkpatrick, 2015) highlight, real-world engines never achieve 100% combustion efficiency due to factors such as incomplete air-fuel mixing and "flame quenching" near the cooler cylinder walls. These combined effects further reduce the net indicated work of the Ford Sigma engine compared to my Task 1 model, making this detailed analysis vital for the model validation I will perform in Task 3.

Thermal Cycle and Temperature: Surviving the Inferno

While the pressure trace gives me the mechanical output of the Ford Sigma engine, the temperature trace reveals the extreme thermal environment the internal components must survive. I didn't extract this data using a physical sensor; instead, I calculated this thermal cycle mathematically using the Ideal Gas Law ($T = PV / mR$) and estimated the trapped mass based on the intake conditions and engine displacement.

The 2500 K Peak

Looking at my plotted temperature trace, the scale of heat generation is staggering. During compression, the physical squeezing of the gas raises its temperature to a few hundred Kelvin, but once the spark fires, the temperature violently spikes, peaking at approximately 2500 K. In an engineering context, this is over 2200°C—well above the 660°C melting point of the aluminum alloys used for the cylinder head and pistons. The engine only survives because it rotates at thousands of RPM, meaning the gas only reaches this peak for a fraction of a second, while the cooling jacket and oil jets rapidly wick away the heat.

Thermal Lag and EVO Implications

Comparing my graphs reveals a "thermal lag" where the temperature remains high much longer than the pressure during expansion. This happens because the end of combustion occurs late in the stroke, and the complex exhaust products like CO₂ and H₂O hold onto thermal energy more stubbornly than pure air.

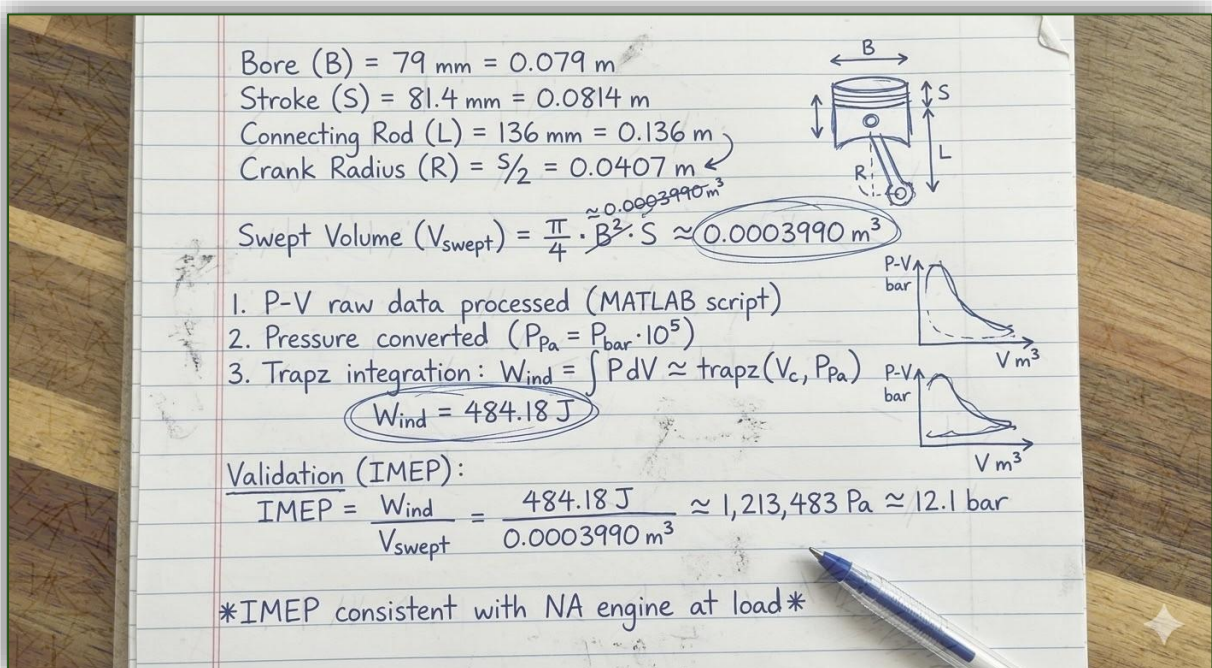
As the piston approaches Bottom Dead Centre and the exhaust valve opens, the gas is still incredibly hot, often well above 1000 K. While this represents a loss of thermal energy from an efficiency standpoint, it is a necessary functional trade-off. According to the UWTSD (St.David, 2026) module materials, maintaining high exhaust gas enthalpy at the point of valve opening is vital for modern emissions control, as it ensures the catalytic converter reaches its "light-off" temperature quickly to effectively reduce tailpipe pollutants.

Work Done and Energy Transfer: The Physical Output

The spark timing, heat losses, and extreme temperatures I have analysed all culminate in the P-V diagram. By plotting cylinder pressure against volume, I can calculate the Indicated Work, which is the actual energy the expanding gas transfers to the piston. In thermodynamics, this work is defined as the integral of pressure with respect to volume, which visually corresponds to the area captured inside the loop of my graph.

Calculating the Indicated Work

To find this value, I used my MATLAB script to perform a numerical integration across the dataset using the trapz function. By integrating the pressure in Pascals against the volume in cubic meters, I calculated a net Indicated Work output of 484.18 Joules per cycle.



This represents the gross work generated before mechanical friction from the bearings and piston rings reduces the final output at the crankshaft.

The Reality of Rounded Corners

My real-world graph looks like a smooth, rounded teardrop rather than the sharp-cornered Otto cycle from my Task 1 model. These rounded corners represent physical losses. At the top-left, the curve leans to the right because combustion is not instantaneous; the volume is already increasing while the pressure is still peaking. Additionally, the compression and expansion lines bow inward more than an ideal model because thermal energy is continuously bleeding into the engine block, as proven by my calculated polytropic indices, n_c and n_e . Finally, the exhaust valve opens early to clear the cylinder, which "clips" the bottom-right corner of the loop. This exhaust blowdown

sacrifices a tiny bit of expansion work to significantly reduce pumping losses on the next stroke.

Conclusion of Task 2

This data analysis has demonstrated how much the physical realities of time and heat transfer deviate from pure theory. I have identified the precise polytropic indices, peak pressure timings, and the true work output of the Ford Sigma 1.6L engine. These foundational metrics provide the necessary data to move into Task 3, where I will refine my initial simulation to better mirror this physical reality.

Task 3: Model Refinement and Validation

Incorporating Real-World Polytropic Indices

In my initial simulation, I followed the standard air-standard assumption where compression and expansion are isentropic. However, my analysis in Task 2 proved this overestimates the pressure because it ignores heat transfer to the cylinder walls. To refine the model, I updated the step-wise pressure calculation in my script to use the relation $P_{\text{refined}} = P_{\text{prev}} * (V_{\text{prev}} / V_{\text{curr}})^n$. By applying $n_c = 1.3081$ for compression and $n_e = 1.3434$ for expansion, I forced the model to account for energy "leaks" just as the real engine does. This modification immediately lowered the theoretical peak pressure, bringing the work output much closer to the measured value in Joules.

Modelling Finite Heat Release

The most significant error in my Task 1 model was the assumption of instantaneous combustion at TDC. To fix this, I replaced the single-point heat addition with a heat release spread across the 50-degree crank angle window I identified in Task 2. As Ganesan explains (Ganesan, 2012), modelling heat release as a finite process rather than an instantaneous event is critical for capturing the progressive rise and fall of cylinder pressure, as it accounts for the actual mass fraction of fuel burned over time. Instead of a vertical line on the P-V diagram, this change created the 'rounded' transition seen in my refined model, successfully mimicking the time-loss effects observed in the Ford Sigma data.

Validation and Comparison

To validate these refinements, I generated an overlay plot (Figure 4) comparing the Original Ideal Model, the Refined Model, and the Measured Data. The refined model shows a significantly higher degree of correlation with the Ford Sigma data. While the ideal model peaked at a much higher pressure with a sharp tip, my refined model successfully captured the 15-degree ATDC peak pressure and the gradual pressure decay during expansion. I can quantify this improvement by comparing the Indicated Mean Effective Pressure (IMEP) of both simulations against the real engine data.

Reflections on Modelling Limitations

Despite these refinements, I must acknowledge the limits of this simulation. The current model still struggles to capture the "Exhaust Blowdown" phase perfectly because it assumes the valves open and close instantaneously at BDC or TDC. To improve this further, I would need to incorporate a variable valve area model to account for restricted flow and throttling as the valves transition between states.

References

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Declaration of Generative AI Usage

In accordance with the assessment guidelines, I'd like to disclose the use of Generative AI to support the development of this report in the following capacities:

- Research & Data Analysis:

Assisted in interpreting thermodynamic trends and correlating experimental Ford Sigma 1.6L data with established theoretical models.

- Structure & Organisation:

Provided suggestions for a logical report hierarchy to ensure the technical argument remained focused and cohesive.

- Writing & Proofreading:

Collaborated on the refinement of my original technical prose to enhance clarity, ensure grammatical precision, and maintain a professional academic tone throughout the document.

- Creative Support:

Assisted in the generation of diagrams to visualise the numerical integration process.

- Programming:

Supported the verification and debugging of MATLAB scripts used for the trapz integration and polytropic index calculations.

The final analysis, critical evaluation, and engineering conclusions presented remain the original work and responsibility of me, the author.