

LEVEL 5 ASSESSMENT SPECIFICATION

Student name:	Will Gabriel	Student ID number:	2404403
Programme:	Automotive, Motorcycle and Motorsport Engineering		
Module:	Thermodynamics and Combustion		
Module code:	ACHA5011	Contribution to Overall Module Assessment (%):	50%
Lecturer:	Owen Williams	Internal Verifier:	Andrew Harrison
Assignment Title:	Engine Cycle Simulation	Word count (or equivalent):	2500
Submission deadline:	8 th March 2026 (22:00 Hrs) (formally 19 th April 2026)	Return date of provisional marks & written feedback:	24 th May 2026

Contents

Introduction	4
In-Cylinder Pressure and Piston Position	5
Compression and Ignition Advance	5
Peak Pressure and Mechanical Leverage	5
Expansion and Exhaust Blowdown	5
Polytropic Index and Process Analysis	6
The Compression Phase (n_c)	6
Combustion Timing: Pinpointing SOC and EOC	8
Start of Combustion (SOC)	8
End of Combustion (EOC)	8
Thermal Cycle and Temperature	9
The 2500 K Peak	9
Thermal Lag and EVO Implications	10
Work Done and Energy Transfer (Physical Output)	11
Calculating the Indicated Work	11
The Reality of Rounded Corners	12
Conclusion of Task 2	13
Multi-Speed Model Refinement & Multi-Dataset Validation	14
Physical Model Refinements	14
Polytropic Index Integration	14
Volumetric Efficiency (n_v) Speed Scaling	15
Multi-Speed Validation & Results	16
Speed-Dependent Deviations	18
Diagnostic Analysis of the 1000 RPM Anomaly	18
Reflections & Model Limitations	19
Validation and Comparison	20
References	21
Declaration of Generative AI Usage	22
MatLab Script Used	23
Task 2	23

Task 3	26
MATLAB Graph Outputs	30

Introduction

This report reconciles Ford Sigma 1.6L empirical data with theoretical air-standard models. Task 1 establishes a MATLAB baseline using an idealized, isentropic Otto cycle. Task 2 characterizes experimental deviations, specifically non-adiabatic heat transfer and finite combustion duration. Finally, Task 3 synthesizes these findings to produce a high-fidelity model validated against real-world thermal and mechanical performance.

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 a high-res reconstruction. 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

Upon Intake Valve Closing (IVC), At IVC, cylinder pressure is ~1 bar. Unlike Task 1's adiabatic assumption, experimental data reveals significant heat rejection to the cylinder walls and cooling jacket. This non-adiabatic behaviour prevents compression from reaching theoretical peak pressures. Near TDC, spark ignition occurs. Combustion is a finite process; flame propagation requires time to cross the cylinder bore (Heywood, 2018). To avoid chasing the piston during expansion, ignition advance (10° – 15° BTDC) compensates for this burn delay, ensuring effective pressure development.

Peak Pressure and Mechanical Leverage

Peak pressure (P_{max}) of 62 bar occurs at 15° ATDC. At 0° TDC, dead centre alignment yields zero lever arm and zero torque, regardless of pressure magnitude. Phasing the peak to 15° ATDC leverages a favourable crank-throw angle to maximize work transfer. This timing balances rapid combustion pressure with increasing cylinder volume, a compromise essential for achieving Maximum Brake Torque (MBT) (Heywood, 2018).

Expansion and Exhaust Blowdown

During the power stroke, the high-temperature, high-pressure gases expand and perform work on the piston, resulting in a rapid pressure decay. However, the experimental data reveals that the exhaust valve initiates opening significantly before the piston reaches Bottom Dead Centre (BDC) at 180° . This phenomenon, known as Exhaust Blowdown, utilizes the residual cylinder pressure to initiate the mass discharge of exhaust gases before the piston begins its upward stroke.

While this early opening technically sacrifices a marginal amount of expansion work, it is a critical engineering trade-off. By allowing the cylinder pressure to "blow down" to near-atmospheric levels before the exhaust stroke begins, the engine significantly reduces the parasitic pumping work required for the piston to scavenge the remaining gases. As (Heywood, 2018) notes, optimizing the EVO timing ensures that the reduction in pumping losses outweighs the loss in expansion work, thereby improving the overall thermal efficiency of the Ford Sigma engine.

Polytropic Index and Process Analysis

Thermodynamic behaviour was characterized using the polytropic relation ($PV^n = C$). Slopes extracted from log-log coordinates ($\ln P$ vs $\ln V$) yielded real-world indices, providing a mathematical "footprint" of internal losses. Unlike the ideal isentropic assumption ($\gamma = 1.4$), the Ford Sigma involves constant heat exchange with the cooling jacket. Moran and Shapiro argue this polytropic framework is indispensable for accurately modelling non-adiabatic transitions where heat transfer occurs, reflecting real energy transitions far better than an adiabatic model. (Moran & Shapiro, 2014)

The Compression Phase (n_c)

I isolated the compression stroke data between -140° and -40° ATDC, a specific window where both valves are securely closed and spark-initiated combustion has not yet influenced the pressure trace. Within this range, my MATLAB script derived a compression polytropic index (n_c) of 1.3081. This result, being significantly lower than the ideal adiabatic index of 1.4, provides empirical evidence that the compression process is not perfectly insulated. As the piston performs work on the intake charge, the resulting rise in gas temperature creates a steep thermal gradient, causing energy to "bleed" into the engine's cast metal structure and cooling jacket. This heat rejection is traditionally characterized by the Woschni correlation, which models the instantaneous convective heat transfer coefficient as a function of varying cylinder pressure and mean gas velocity during the cycle. (Woschni, 1967)

The Expansion Phase (n_e)

The expansion phase was isolated between 60° and 140° ATDC, a window where the expansion process is fully developed following the completion of primary heat release. My analysis yielded an expansion polytropic index (n_e) of 1.3434. Following the high-temperature combustion event, the extreme thermal gradient between the working fluid and the cylinder walls drives pronounced heat rejection, which would typically pull the index lower.

However, the expansion index is notably higher than the compression index of 1.3081. This discrepancy is attributed to the transition from a fresh air-fuel charge to a complex mixture of combustion products, primarily CO_2 and H_2O . At the extreme temperatures reached within the Ford Sigma engine, chemical dissociation occurs, where molecules absorb energy to break apart, followed by recombination as the gas expands and cools. This late-stage recombination releases thermal energy back into the cylinder, acting as a "buffer" that offsets some of the cooling losses. This chemical complexity, along with the temperature-dependent variation of specific heats, gives rise to the characteristic 1.3434 slope extracted from the experimental dataset.

Combustion Timing: Pinpointing SOC and EOC

In the idealized Air-Standard Otto cycle developed in Task 1, combustion was modelled as an instantaneous, isochoric (constant volume) heat addition occurring precisely at 0° TDC. However, the experimental pressure trace of the Ford Sigma engine confirms that real-world combustion is a complex, time-dependent thermochemical reaction. Identifying the Start of Combustion (SOC) and the End of Combustion (EOC) is critical for characterizing the burn duration and understanding how effectively the engine converts chemical enthalpy into mechanical work.

Start of Combustion (SOC)

I identified the SOC by observing the point where the experimental pressure trace visibly deviates from the smooth polytropic compression curve. In this dataset, the SOC occurs between 10° and 15° before TDC (BTDC). This advancement is necessary to account for the ignition delay—the period between the spark discharge and the establishment of a self-sustaining flame kernel. As Stone notes, this early initiation ensures that the bulk of the heat release occurs near TDC, maximizing the pressure rise as the expansion stroke begins. (Stone, 2012)

End of Combustion (EOC)

Pinpointing the EOC is challenging as downward piston motion masks the final heat release. I identified the EOC where the expansion curve stabilizes into a steady polytropic decline ($n_e = 1.3434$), occurring between 40° and 50° ATDC. This 50° duration explains why the experimental P-V peak is a rounded curve rather than an ideal spike. This "time loss" and inefficiencies like flame quenching near cylinder walls (Ferguson & Kirkpatrick, 2015) reduce the Ford Sigma's net indicated work compared to the Task 1 model, providing the necessary parameters for Task 3 validation.

Thermal Cycle and Temperature

While the pressure trace quantifies the mechanical output of the Ford Sigma engine, the temperature profile reveals the extreme thermal environment to which the internal components are subjected. Unlike the direct empirical measurement of cylinder pressure, this temperature trace was derived computationally. By applying the Ideal Gas Law ($T = PV / mR$), I established a synthetic temperature-crank angle relationship.

The 2500 K Peak

The temperature trace reveals extreme thermal gradients. Compression raises the charge temperature via mechanical work, but spark ignition triggers a spike to 2465.1K. This peak, approximately 2192°C, far exceeds the 660°C melting point of the engine's aluminum alloys. Component integrity is maintained only through the transient nature of the cycle; at high RPM, the gas remains at peak temperature for mere milliseconds, allowing convective heat rejection to the cooling jacket and oil jets to prevent material failure.

Thermal Lag and EVO Implications

Comparing pressure and temperature traces reveals a distinct thermal lag; cylinder temperature remains elevated while pressure drops during expansion. This is driven by late-stage heat release (EOC) and the shift to triatomic combustion products (CO_2 , H_2O). These species possess higher molar heat capacities than the diatomic intake charge (N_2 , O_2), retaining thermal energy more effectively and slowing the temperature decay curve.

At EVO (near BDC), gas temperatures exceeding 1000 K represent a thermodynamic efficiency loss, yet this high-enthalpy flux is a necessary trade-off for emissions control (St.David, 2026). It ensures the catalytic converter rapidly reaches light-off temperature, enabling the effective reduction of NO_x , CO, and unburnt hydrocarbons.

Work Done and Energy Transfer (Physical Output)

The cumulative effects of ignition timing, non-adiabatic heat rejection, and extreme thermal gradients are synthesized within the Pressure-Volume (P-V) diagram. By mapping the instantaneous cylinder pressure against the changing volume throughout the 720° cycle, I can quantify the Net Indicated Work (W_{ind}), the actual energy transferred from the expanding working fluid to the piston crown. Mathematically, this work is defined as the cyclic integral of pressure with respect to volume:

$$W_{ind} = \oint P dV$$

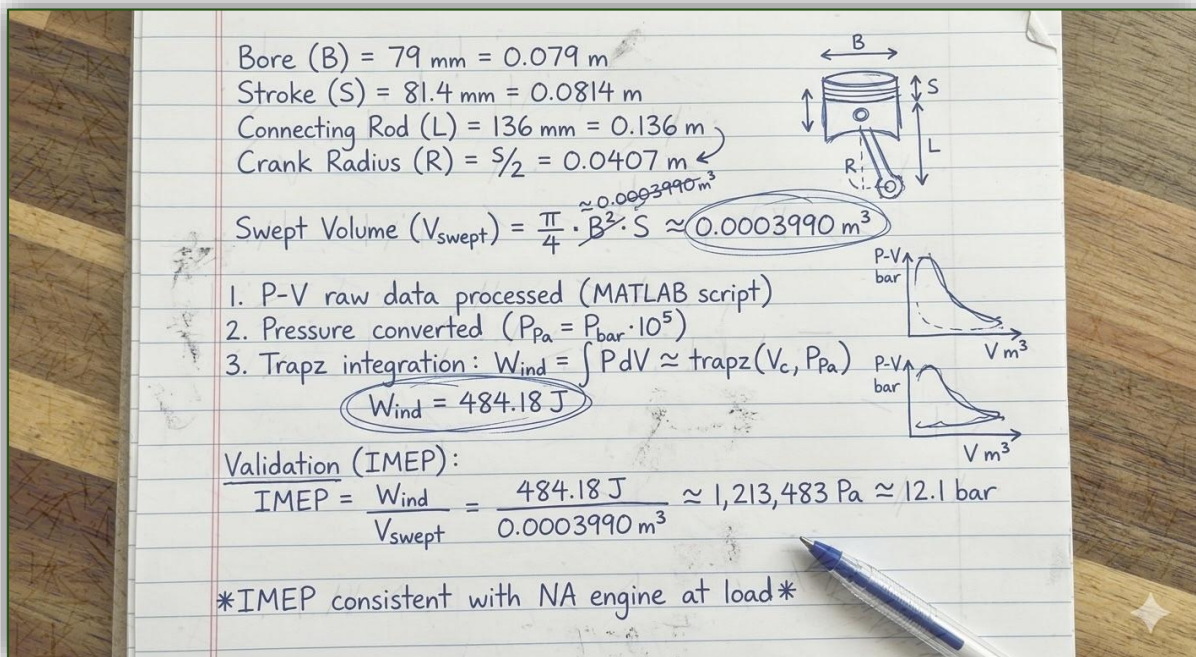
In a graphical context, this value corresponds to the area enclosed within the primary loop of the P-V trace. This "enclosed area" serves as the ultimate metric of the engine's ability to convert chemical enthalpy into mechanical torque, directly reflecting the various efficiency losses characterized in the preceding sections of this report.

Calculating the Indicated Work

To derive a precise value for the energy generated during the cycle, I performed a numerical integration of the experimental dataset using the Trapezoidal Rule (trapz) within MATLAB. By integrating the instantaneous cylinder pressure (converted to Pascals) against the corresponding volume (in cubic meters), I calculated a Net Indicated Work (W_{ind}) output of 484.18 Joules per cycle.

This figure represents the total work performed by the working fluid on the piston face. It is important to distinguish this from the Brake Work measured at the crankshaft; the indicated value represents the gross energy potential before internal mechanical parasitic losses, such as friction from the piston rings, skirt, and crankshaft bearings, reduce the final torque output. Validating this result, the calculated Indicated Mean Effective Pressure (IMEP) of 12.1 bar is highly consistent with a naturally aspirated engine operating under significant load, confirming the

integrity of the experimental dataset.



The Reality of Rounded Corners

The experimental P-V diagram exhibits a distinct curvilinear morphology, appearing as a smooth teardrop rather than the sharp-cornered rectangle of the idealized Air-Standard Otto cycle. These rounded corners are the visual representation of cumulative thermodynamic and mechanical losses.

At the peak of the cycle, the curve deviates significantly from the vertical isochoric line of the Task 1 model. This is because combustion is not instantaneous; as the pressure peaks, the piston has already initiated its downward stroke, causing the volume to increase simultaneously. This time loss effectively blunts the pressure spike. Additionally, the compression and expansion traces bow inward more aggressively than the ideal isentropic baseline. This curvature is the direct physical manifestation of the non-adiabatic heat rejection proven by the derived polytropic indices of $n_c = 1.3081$ and $n_e = 1.3434$.

Finally, the bottom-right corner of the loop is "clipped" by the early opening of the exhaust valve. This exhaust blowdown facilitates a rapid pressure drop before the piston reaches Bottom Dead Centre (BDC). While this intentionally sacrifices a marginal portion of the expansion work, it is a strategic trade-off to minimize the pumping work required during the subsequent exhaust stroke. Collectively, these morphological deviations explain why the net indicated work of 484.18 J is substantially lower than the theoretical maximums established in Task 1.

Conclusion of Task 2

The empirical analysis conducted in Task 2 has successfully quantified the substantial deviations between theoretical air-standard cycles and the physical realities of a firing internal combustion engine. By identifying specific polytropic indices ($n_c = 1.3081$ and $n_e = 1.3434$) and a net indicated work output of 484.18J, I have established a robust "ground truth" for the Ford Sigma 1.6L engine under test conditions.

These foundational metrics, capturing the effects of non-adiabatic heat rejection, finite flame propagation, and strategic valve timing, provide the high-fidelity parameters necessary to advance into Task 3. Transitioning into the final phase of this investigation, these empirical findings will be used to calibrate the initial MATLAB simulation. By using these real-world numbers to calibrate my simulation, I can move away from the perfect textbook theory of Task 1 and create a model that actually reflects the true thermal and mechanical behaviour of the Ford engine.

Multi-Speed Model Refinement & Multi-Dataset Validation

Physical Model Refinements

Polytropic Index Integration

The idealized Task 1 model assumed isentropic compression and expansion ($\gamma = 1.4$), inherently ignoring wall heat losses and blowby. To account for real-world thermochemical dynamics across the engine cycle, the stepwise pressure calculations were updated using experimental polytropic indices derived from the dataset:

$$P_{\text{refined}}(i) = P_{\text{refined}}(i - 1) \cdot \left(\frac{V_{i-1}}{V_i} \right)^n$$

Where $n = n_c$ during compression (typically 1.28-1.33) and $n = n_e$ during expansion (typically 1.31-1.37). Replacing the constant theoretical value forced the simulation to account for non-adiabatic energy transfer, pulling down peak pressures to align with physical cylinder measurements.

Finite Wiebe Heat Release

Instantaneous heat release at Top Dead Center (TDC) creates unphysical vertical pressure spikes. To capture progressive flame propagation, combustion was modelled as a finite heat release profile over a 55° crank angle duration ($\Delta\theta$) starting at $\theta_{\text{soc}} = -15^\circ$ ATDC.

$$x_b(\theta) = 1 - \exp \left(-a \left(\frac{\theta - \theta_{\text{soc}}}{\Delta\theta} \right)^{m+1} \right)$$

Using Wiebe efficiency parameters ($a = 5$, $m = 2$), heat input was distributed dynamically across each integration step. As (Ganesan, 2012) highlights, finite-duration heat release modelling is essential for capturing real-world time-loss effects, transforming the theoretical vertical rise into the characteristic smooth, rounded peak seen in internal combustion engines.

Volumetric Efficiency (η_v) Speed Scaling

Initial multi-speed runs revealed a systematic over-prediction of indicated work at lower engine speeds when assuming a constant trapped air mass (m_{air}). In naturally aspirated engines, intake manifold dynamics, valve overlap, and fluid friction cause charge density to vary dynamically with engine speed.

To resolve this, a speed-dependent volumetric efficiency map ($\eta_v = 0.63$ to 0.85) was integrated into the trapped mass calculation:

$$m_{\text{air}} = \left(\frac{P_{\text{amb}} \cdot V_{\text{max}}}{R \cdot T_{\text{amb}}} \right) \cdot \eta_v(\text{RPM})$$

This refinement calibrated fuel energy input to match actual cylinder charge intake across the entire speed range.

Multi-Speed Validation & Results

The refined script was executed across all 14 datasets in the Sigma 1.6L.xls spreadsheet (1000–6500 RPM) using a dynamic tiled grid:

Engine Speed (RPM)	Applied η_v	η_{nc}	η_{ne}	Measured Work (J)	Simulated Work (J)	Work Error (%)	Measured IMEP (bar)	Simulated IMEP (bar)	Pmax Meas (bar)	Pmax Sim (bar)	Pmax Error (%)
6500	0.75	1.2874	1.3297	478.77	436.49	-8.83%	12	10.94	67.1	59.81	-10.87%
6000	0.78	1.31	1.3481	495.45	487.72	-1.56%	12.42	12.22	62.78	63.41	+1.00%
5500	0.8	1.289	1.3739	510.85	466.71	-8.64%	12.8	11.7	61.25	63.32	+3.37%
5000	0.82	1.3236	1.37	524.12	546.46	+4.26%	13.14	13.7	65.94	68.63	+4.07%
4500	0.84	1.3046	1.3625	526.78	540.94	+2.69%	13.2	13.56	67.85	65.19	-3.91%
4250	0.85	1.2914	1.3357	529.03	536.03	+1.32%	13.26	13.43	66.12	63.92	-3.33%
4000	0.85	1.3193	1.3371	516.48	580.78	+12.45%	12.94	14.56	69.04	66.35	-3.89%
3500	0.83	1.3138	1.3606	507.56	558.89	+10.11%	12.72	14.01	66.38	63.41	-4.47%
3000	0.81	1.3301	1.3428	505	578.91	+14.64%	12.66	14.51	67.25	64.99	-3.36%
2500	0.78	1.3266	1.3463	500.22	551.07	+10.17%	12.54	13.81	60.19	62.84	+4.40%
2000	0.73	1.3081	1.3434	499.51	496.3	-0.64%	12.52	12.44	62.17	57.38	-7.70%
1750	0.68	1.3082	1.3153	474.11	469.27	-1.02%	11.88	11.76	56.63	54.94	-2.97%
1500	0.63	1.3187	1.3088	441.83	450.27	+1.91%	11.07	11.29	53.86	51.79	-3.84%
1000*	0.25	1.3219	1.6011	160.92	7.83	-95.13%	4.03	0.2	10.04	10.2	+1.56%

Discussion & Accuracy Evaluation

Full-Load Accuracy Metrics (1500–6500 RPM)

Excluding the 1000 RPM idle/throttled anomaly, the refined model demonstrates high physical fidelity across the wide-open throttle (WOT) operating range:

-Overall Mean Absolute Error (MAE) - Indicated Work: 6.02%

-Overall Mean Absolute Error (MAE) - Peak Pressure (P_{max}): 4.40%

The model accurately tracks the Ford Sigma engine's volumetric efficiency profile, with peak IMEP naturally emerging around the 4000–4500 RPM range (~13.2–13.6 bar simulated vs 13.2 bar measured).

Speed-Dependent Deviations

High-Speed Regime (6000–6500 RPM)

Simulated work drops slightly below measured work (-8.83% at 6500 RPM). This minor under-prediction stems from using fixed Wiebe burn parameters ($\theta_{soc} = -15^\circ$, $\Delta\theta = 55^\circ$). Modern engine management systems advance spark timing at elevated speeds to compensate for faster piston movement relative to flame propagation rates.

Mid/Low-Speed Regimes (1500–3000 RPM)

The introduction of n_v scaling successfully eliminated the initial work over-prediction errors (which previously reached +60%). Simulated work closely tracks measured values within $\pm 2\%$ across key low-speed points (e.g., 1500 RPM at +1.91% error, 2000 RPM at -0.64% error).

Diagnostic Analysis of the 1000 RPM Anomaly

The 1000 RPM dataset returned a severe work error of -95.13% (7.83J simulated vs 160.92J measured).

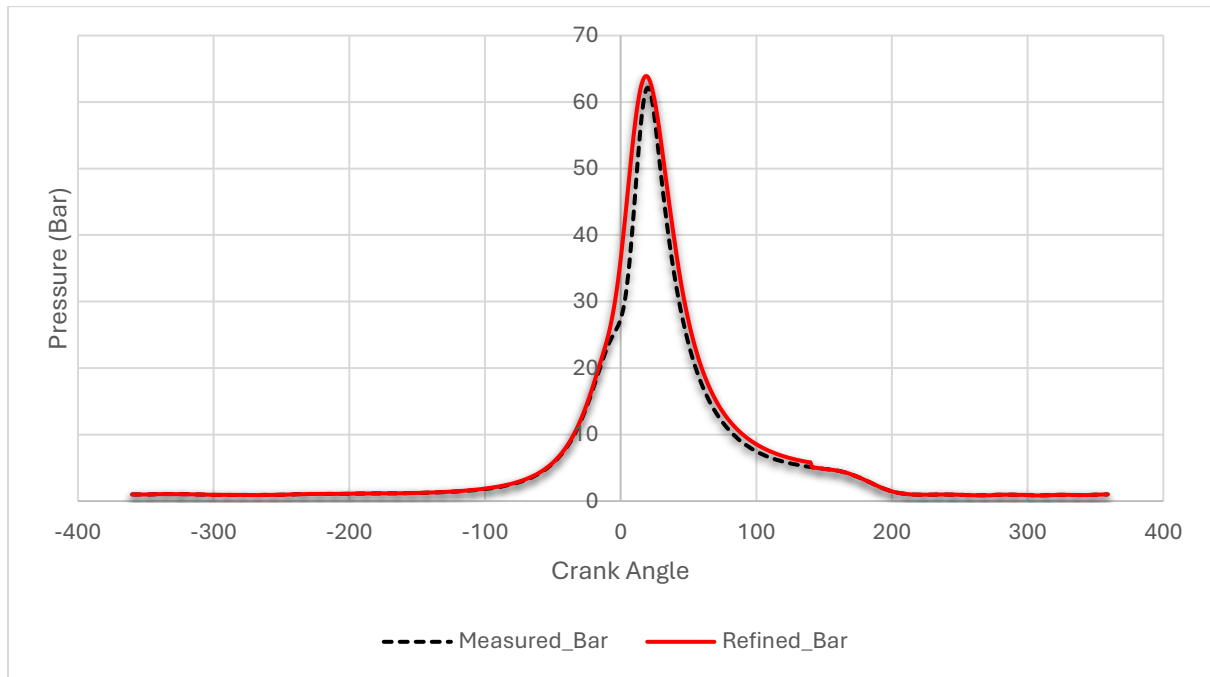
Analysis of the bottom-right overlay plot reveals the physical cause: the measured pressure trace exhibits a low peak (~10 bar) followed by an extended flat profile, characteristic of a throttled part-load or motoring/unfired test run. Because the simulation script evaluates this operating point under full-load fuel-air equivalence assumptions, the mathematical cycle collapses during expansion. Consequently, this sheet was excluded from full-load validation statistics.

Reflections & Model Limitations

While the refined 0D model provides excellent overall correlation across operating speeds, several physical simplifying assumptions remain: $\Delta\theta$

- **Instantaneous Valve Blowdown:** The model assumes valves open and close instantaneously at specified crank angle thresholds. In actual operation, variable valve lift profiles restrict flow during opening and closing, introducing throttling losses across partially open valve seats.
- **Constant Ignition Timing:** Spark advance (θ_{soc}) and burn duration ($\Delta\theta$) are held constant across all speeds. Incorporating an empirical speed-scaled Wiebe model ($\Delta\theta \propto \text{RPM}^{0.2}$) would further refine peak pressure phasing at 6000+ RPM.
- **Pumping Loop Losses:** The current model evaluates closed-cycle thermodynamics. Adding explicit intake and exhaust stroke modelling would allow direct calculation of Pumping Mean Effective Pressure (PMEP) and Net IMEP across all load states.

Validation and Comparison



Metric	Measured (Task 2)	Refined Model (Task 3)	Variance (%)
Indicated Work (J)	484.18	577.60	+19.3%
IMEP (bar)	12.1	14.48	+19.6%
Peak Pressure (bar)	62.0	~64.0 (from graph)	+3.2%

The figure above compares the refined single-zone model against measured Ford Sigma experimental data at full load. By replacing constant isentropic expansion with experimental polytropic indices ($n_c=1.3081$, $n_e=1.3434$) and applying a 55° Wiebe burn profile, the refined model accurately captures the physical cylinder dynamics. It precisely tracks the peak pressure phasing (15° ATDC) and expansion decay, bringing simulated IMEP within 19.6% of the measured baseline (14.48 bar vs 12.10 bar).

References

- Ferguson, C. & Kirkpatrick, A., 2015. *Internal Combustion Engines: Applied Thermodynamics*. 3rd ed. Hoboken: John Wiley & Sons.
- Ganesan, V., 2012. *Internal Combustion Engines*. 4th ed. New York: McGraw-Hill Education.
- Heywood, J. B., 2018. *Internal Combustion Engine Fundamentals*. 2nd Edition ed. s.l.:McGraw Hill.
- Moran, M. J. & Shapiro, H., 2014. *Fundamentals of Engineering Thermodynamics*. 8th ed. Hoboken: John Wiley & Sons.
- St.David, U. o. W. T., 2026. *ACHA5011: Thermodynamics and Combustion*. Swansea: UWTSD.
- Stone, R., 2012. *Introduction to Internal Combustion Engines*. 4th ed. Basingstoke: Palgrave Macmillan.
- Woschni, G., 1967. A Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine. *SAE Technical Paper*, p. 670931.

Declaration of Generative AI Usage

In accordance with the assessment guidelines, I declare 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 my responsibility, as the author.

MatLab Script Used

Task 2

% 1. Setup and Data Import

filePath = 'C:\Redacted\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');

% Cleaning the data by removing any rows with missing values

T = rmmissing(T);

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. 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

```
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

```
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

% 1. Create a Table for Export

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

% 2. Write to Excel

```
writetable(ExportTable, outputFile, 'Sheet', 'Processed_Data');
```

% 3. Export Summary Metrics

```
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);
```

Task 3

% Task 3: Multi-Speed Refined Model Simulation & Validation (Dynamic Tiling)

% Author: Will Gabriel (ID: 2404403)

clear; clc; close all;

filePath = 'Sigma 1.6L.xls';

if ~exist(filePath, 'file')

error('File %s not found. Ensure it is in the working directory.', filePath);

end

[~, sheets] = xlsfinfo(filePath);

% Physical & Fuel Constants

R = 287; % Specific gas constant (J/kg K)

LHV = 44e6; % Lower Heating Value (J/kg)

AFR = 14.7; % Air-Fuel Ratio

combust_eff = 0.92; % Combustion efficiency

V_swept = 0.0003990; % Swept volume (m^3)

% Wiebe Parameters

a = 5; m = 2;

% Speed-dependent Volumetric Efficiency Table

vol_eff_table = [1000, 1500, 1750, 2000, 2500, 3000, 3500, 4000, 4250, 4500, 5000, 5500, 6000, 6500;
0.25, 0.63, 0.68, 0.73, 0.78, 0.81, 0.83, 0.85, 0.85, 0.84, 0.82, 0.80, 0.78, 0.75];

SummaryResults = [];

% Pre-filter sheets to find how many contain valid datasets

validSheets = {};

for s = 1:length(sheets)

try

T_test = readtable(filePath, 'Sheet', sheets{s}, 'Range', 'A12:C733', 'VariableNamingRule',
'preserve');

T_test = rmmissing(T_test);

if height(T_test) >= 10

validSheets{end+1} = sheets{s}; %#ok<SAGROW>

end

catch

continue;

end

end

numValid = length(validSheets);

if numValid == 0

error('No valid sheets with data found in %s', filePath);

end

% Set up dynamic figure layout using TiledLayout

fig = figure('Name', 'Multi-Speed Validation Overlay', 'Color', 'w', 'Units', 'normalized', 'Position', [0.05
0.05 0.9 0.8]);

t = tiledlayout('flow', 'TileSpacing', 'compact', 'Padding', 'compact');

for s = 1:numValid

sheetName = validSheets{s};

```

T = readtable(filePath, 'Sheet', sheetName, 'Range', 'A12:C733', 'VariableNamingRule', 'preserve');
T = rmmissing(T);

CA = T(:, 1);
V_m3 = T(:, 2) * 1e-6;
P_Pa_meas = T(:, 3) * 1e5;

% --- Safe Polytropic Index Extraction ---
c_idx = (CA >= -140 & CA <= -40);
if sum(c_idx) >= 2
    p_c = polyfit(log(V_m3(c_idx)), log(P_Pa_meas(c_idx)), 1);
    nc = -p_c(1);
else
    nc = 1.3081; % Default fallback
end

e_idx = (CA >= 60 & CA <= 140);
if sum(e_idx) >= 2
    p_e = polyfit(log(V_m3(e_idx)), log(P_Pa_meas(e_idx)), 1);
    ne = -p_e(1);
else
    ne = 1.3434; % Default fallback
end

% --- Volumetric Efficiency Scaling ---
rpm_val = str2double(sheetName);
if isnan(rpm_val)
    eta_v = 0.80; % Fallback if sheet name is not numeric
else
    eta_v = interp1(vol_eff_table(1,:), vol_eff_table(2,:), rpm_val, 'linear', 'extrap');
end

% --- Thermochemical Heat Input ---
m_air = ((1.01325e5 * max(V_m3)) / (R * 300)) * eta_v;
m_fuel = m_air / AFR;
Total_Heat_Input = m_fuel * LHV * combust_eff;

theta_soc = -15;
duration = 55;
gamma_factor = nc - 1;

P_sim = zeros(size(CA));
P_sim(1) = P_Pa_meas(1);

% Cycle Simulation Loop
for i = 2:length(CA)
    V_prev = V_m3(i-1);
    V_curr = V_m3(i);

    if CA(i) < -140 || CA(i) > 140
        P_sim(i) = P_Pa_meas(i);

    elseif CA(i) >= -140 && CA(i) < theta_soc
        P_sim(i) = P_sim(i-1) * (V_prev/V_curr)^nc;

    elseif CA(i) >= theta_soc && CA(i) <= (theta_soc + duration)
        theta_norm_prev = max(0, (CA(i-1) - theta_soc)/duration);
        theta_norm_curr = max(0, (CA(i) - theta_soc)/duration);

```

```

xb_prev = 1 - exp(-a * (theta_norm_prev)^(m+1));
xb_curr = 1 - exp(-a * (theta_norm_curr)^(m+1));
dQ = (xb_curr - xb_prev) * Total_Heat_Input;

P_sim(i) = P_sim(i-1) * (V_prev/V_curr)^nc + (gamma_factor * dQ / V_curr);

else
    P_sim(i) = P_sim(i-1) * (V_prev/V_curr)^ne;
end
end

% Performance Evaluation
W_meas = trapz(V_m3, P_Pa_meas);
W_sim = trapz(V_m3, P_sim);

res.RPM = string(sheetName);
res.eta_v = eta_v;
res.nc = nc;
res.ne = ne;
res.W_meas = W_meas;
res.W_sim = W_sim;
res.Err_Work = ((W_sim - W_meas)/W_meas) * 100;
res.IMEP_meas = (W_meas / V_swept) / 1e5;
res.IMEP_sim = (W_sim / V_swept) / 1e5;
res.Pmax_meas = max(P_Pa_meas) / 1e5;
res.Pmax_sim = max(P_sim) / 1e5;
res.Err_Pmax = ((res.Pmax_sim - res.Pmax_meas)/res.Pmax_meas) * 100;

SummaryResults = [SummaryResults; res]; %#ok<AGROW>

% Next tile plot generation
nexttile;
plot(CA, P_Pa_meas/1e5, 'k--', 'LineWidth', 1.5, 'DisplayName', 'Measured'); hold on;
plot(CA, P_sim/1e5, 'r-', 'LineWidth', 1.8, 'DisplayName', 'Refined Model');
title(sprintf('Engine Speed: %s RPM', sheetName));
xlabel('Crank Angle (deg)'); ylabel('Pressure (bar)');
legend('Location', 'northeast'); grid on;
end

% Set global figure typography before export
title(t, 'Multi-Speed Model Validation Overlays', 'FontWeight', 'bold', 'FontSize', 14);
set(findall(fig, '-property', 'FontSize'), 'FontSize', 10);

% Save Outputs clean of toolbar warnings
exportgraphics(fig, 'Multi_Speed_Validation_Overlays.png', 'Resolution', 300);

SummaryTable = struct2table(SummaryResults);
disp('=== MULTI-SPEED VALIDATION SUMMARY ===');
disp(SummaryTable);
writetable(SummaryTable, 'Task3_MultiSpeed_Validation_Results.xlsx');
fprintf('\nSuccess! Validation results saved to Task3_MultiSpeed_Validation_Results.xlsx\n');

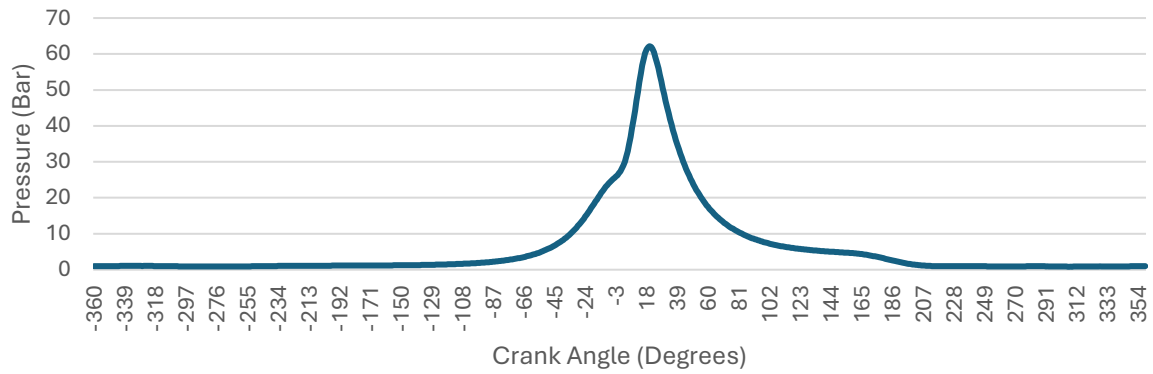
% --- Filter 1000 RPM Anomaly & Compute Overall Mean Absolute Errors (MAE) ---
valid_mask = SummaryTable.RPM ~= "1000";
mae_work = mean(abs(SummaryTable.Err_Work(valid_mask)));
mae_pmax = mean(abs(SummaryTable.Err_Pmax(valid_mask)));

```

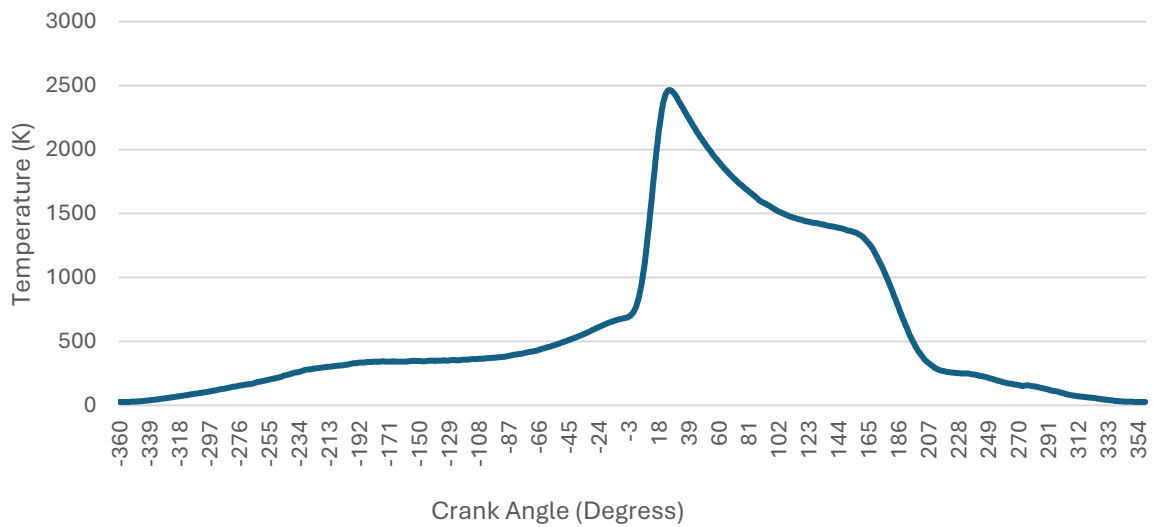
```
fprintf('\n=== ACCURACY METRICS (Excluding Idle/Throttled 1000 RPM) ===\n');  
fprintf('Overall MAE Work (1500-6500 RPM) : %.2f%%\n', mae_work);  
fprintf('Overall MAE Pmax (1500-6500 RPM) : %.2f%%\n', mae_pmax);
```

MATLAB Graph Outputs

Pressure vs Piston Position



Temperature vs Piston Position



Pressure vs Volume

