Testing of a Modern Wankel Rotary Engine - Part III: firing condition analysis

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Abstract
This work represents a further contribution to reporting experimental activities carried out on a modern Wankel rotary engine. Specifically, in this study, the firing performance of the Advanced Innovative Engineering 225CS engine is analysed. Preliminary presentations of the experimental and measurement setup and a motoring analysis were extensively covered in Part I and II of this suite of papers while the current work presents the combustion analysis of the firing indicated pressure cycles collected through the bespoke combustion analyser software developed within the project. With the Wankel rotary engine gaining popularity again due to its potential as a range extender for battery electric vehicles, the aim of this work was mainly to analyse the fuel consumption together with the overall efficiency and the emissions at different engine speeds and loads as per classic steady-state engine testing. The characteristic curves of power and torque thus derived from the experimental measurements are reported while further deductions on combustion phenomena are then drawn from an analysis of the indicated pressure cycles. Specifically, parameters such as the rate of heat release, the net heat release, the IMEP and the indicated instantaneous torque are assessed. Further considerations are drawn on the overall mechanical efficiency relying on the IMEP computed from the indicated pressure cycles and the BMEP inferred from the torqued measured experimentally under steady-state conditions. Furthermore, the effects of the combustion on the internal pressure of the Self-Pressurizing Air-Rotor-Cooling System employed are evaluated in addition to parameters such as the Coefficient of Variation of the IMEP for the evaluation of the cycle-to-cycle combustion quality and engine regularity. Finally, the post-processed data represent an update to the historical literature on Wankel rotary engines. In addition, it can be used for the development and validation of numerical models for such engines hence allowing investigation by means of simulations of the effect on efficiency and performance of the rotary engine when employing alternative fuels such as hydrogen in the future.

Introduction
Currently, there is a renewed interest in internal combustion engines (ICEs), especially when applied as range extenders (RExes) in hybrid electric vehicles (HEVs) [1-4] and/or employing decarbonized fuels such as hydrogen or ammonia. The global COVID-19 pandemic with its subsequent lockdowns and deep traffic reduction caused a beneficial reduction of CO₂ emissions, but the decline in emissions has been reported as “less than what we expected” [5,6] with an average reduction of around -26% for each country in 2020 [7]. This fact partially demonstrated that automotive ICEs and civil aviation are not the main causes of CO₂ production while still surely representing an easy target for lawmakers and governments when it comes to emissions-reduction legislation. Conversely, the increased awareness around the emissions and the raw material supply issues generated by batteries, manufacturing electrical machines and electrical energy production undermined the predominance of battery electric vehicles (BEVs) in being considered the most feasible and only solution for CO₂ emissions reduction from light-duty transportation [8-12]. In addition, the increased demand for electrical energy and raw materials certainly played a role in soaring energy and material prices [13-15]. As a consequence, the application of ICEs in the transportation field has been reconsidered in light of their reliability, well-developed energy infrastructure, low cost and simple maintenance.

The scientific community is moving its interest towards the development of new and improved decarbonised fuels (e-fuels) or the implementation of technologies based on hydrogen (H₂) ICEs and fuel cells) together with innovative engine architectures and powertrains. Innovate UK and the Advanced Propulsion Centre predicted this trend and funded the ADAPT Intelligent Powertrain project, led by Westfield Sports Cars Limited in partnership with Advanced Innovative Engineering (AIE) UK, Saietta, General Engine Management System (GEMS) and the University of Bath. The project aimed to develop an innovative hybrid powertrain for automotive applications. Within the University of Bath, the Institute for Advanced Automotive Propulsion Systems (IAAPS) established a fruitful co-operation with AIE UK where their 225CS rotary engine [16] has been experimentally tested and modelled through kinematic and computational fluid dynamics codes to make this machine Euro 6 compliant. Within the project, much effort was spent on innovative fuel control strategies to reduce the rotary’s well-known appetite for fuel [17,18]. In addition, the several theoretical investigations supported by numerical activities proved that is possible to improve the emissions of Wankel rotary engines and successfully employ them as range extenders [19,20]. Furthermore, the Wankel configuration demonstrated the capability to be coupled with a rotary expander of the same geometric family and so improve the overall system efficiency and fuel consumption while further reducing its emissions [21].

Without any doubt Mazda is considered the world largest and most successful rotary engine manufacturer, having started production of the engine type in 1967. After over 50 years of production and the discontinuation of the Mazda RX-8 model, the aforementioned automaker still firmly believes in the capabilities of this architecture and plans to reintroduce it as a range extender during the first half of 2022 [22, 23]. The design and engineering aspects of this singular
machine are extensively covered in [24, 25], and amongst them, undoubtedly the peculiar and advantageous characteristics of compact design, high power-to-weight ratio, low vibration and high specific power output make the Wankel rotary engine a highly suitable for REx applications.

Advanced Innovative Engineering UK with its 225CS model (shown in Figure 1) further enhanced rotary engine technology by employing a licenced rotor cooling system called Self-Pressurised Air-Rotor-Cooling System (SPARCS) [26]. This is shown in Figure 2. In this system, the blow-by gas from the combustion chambers is used as a cooling medium for the rotor internals to reduce the likelihood of dangerous failures of apex and side seals and prevent the main rotor bearing from over-heating [27].

![Figure 1. AIE UK Ltd. 225CS Wankel Rotary engine [16].](image1)

![Figure 2. Compact SPARCS layout in the 225CS [26].](image2)

For sake of clarity, it is worth mentioning at this point that the engine discussed here is a single-rotor unit as is apparent from the sectional view reported in Figure 2. With a swept volume of 225cm³, the engine has a compression ratio of 9.6:1 and a core mass of 10kg while it can deliver a maximum power of 30kW at over 8000 rpm, hence respecting all the aforementioned positive characteristics inherent in Wankel rotary engines. Other essential geometrical parameters are reported in Table 1 together with some interesting insight regarding the inlet and outlet port timing and port overlap duration.

![Table 1. AIE 225CS engine parameters and port timing.](image3)

The current work continues the presentation of the experimental results collected on the engine equipped with fast-response pressure transducers employed for the combustion analysis and, together with the motoring analysis reported in Part II [2], represents a further insight into the thermodynamics and fluid dynamics phenomena within the rotary engine. The aims of the current study are various and all focused around the performance of the engine under firing conditions: the usual torque and power curves from the experimental measurements will be provided together with other parameters such as the fuel consumption, the indicated mean effective pressure (IMEP) and the engine emissions. Other important parameters will be computed from the recorded experimental measurements: the pressure cycles in the firing condition will be reported and from them, computations of the heat release, the rate of heat release and the rate of pressure rise will all be carried out. The latter parameters will represent the baseline for further considerations on the combustion mechanism, overall efficiency and emissions for the AIE 225CS engine. Finally, fast-response and crank-resolved parameters, e.g. the instantaneous indicated torque, will be averaged and compared with the steady-state measurements taken from the test rig in order to validate the mechanical and thermal measurements and broadly evaluate the energy cascade for this particular machine.

Most of the experimental data in firing conditions will be reported in the next sections together with the methodologies used for the acquisition and post-processing of the data, especially those coming from the pressure transducers employed for the combustion analyser. Subsequently, further considerations will be made on the evaluation of the combustion processes inside the engine by analysing its...
pressure cycles. For sake of conciseness, the text will mainly report the results at three different rotational speeds, 3000, 4500 and 6000 rpm, with same BMEP of 5 bar and then three different BMEP values of 3, 5 and 7 bar at a single rotational speed of 4500rpm. Other data will be reported in the Appendix for sake of completeness.

**Engine performance at steady-state conditions**

The AIE 225CS was installed and tested at the University of Bath within the Institute for Advanced Automotive Propulsion Systems (IAAPS) laboratories. As extensively reported in Part I [1], the engine was connected to a dynamic dynamometer through an HBM torque flange and a rubber-damped propeller shaft to reduce and isolate any torsional vibration between the engine and the electrical machine (see Figure 3).

![Figure 3. AIE 225CS engine installed IAAPS at the University of Bath.](image)

The engine was tested in a range from 3000 to 6500 rpm in steps of 500 rpm. The choice of the rotational speed testing range was made by taking into account the power and operating range of current RExs with a displacement of ~675cm³. Some parameters, such as the torque and the rotational speed were acquired directly as outputs from the HBM torque flange and collected and saved by the Sierra CP Cadet data acquisition system, which was also used for the internal computation of derived parameters such as the engine power. Both the aforementioned engine parameters are reported in Figure 4 for the wide-open throttle (WOT) condition, as usual as a function of the rotational speed. As is possible to appreciate in the aforementioned figure the engine confirms the expected high performance while not reaching the maximum power declared by the manufacturer due to the rotational speed testing range being limited to 6500 rpm. However, the delivered maximum power of nearly 25kW is sufficient and comparable with the power of the range extenders in the same category.

![Figure 4. AIE 225CS torque and power curves.](image)

Subsequently, it was straightforward to compute the engine brake mean effective pressure (BMEP) by post-processing the collected data on the torque and power at the different rotational speeds. It is important to mention that particular attention must be paid when computing the BMEP for a Wankel engine by using the experimental torque: while the general and well-known Equation 1 is still valid, the factor ε used is three, different to 2 and 4-stroke engines where the factor equals one and two respectively:

$$\text{BMEP} = \frac{2\pi \cdot \epsilon \cdot T_t}{V_t} = \frac{2\pi \cdot T_t}{V_s} \quad [\text{Pa}]$$  

where:

- $T_t =$ Total torque measured [Nm]
- $V_s =$ Swept volume for a single flank [m³] (225cm³ or 225e-6 m³ for the AIE 225CS)
- $V_t =$ Total swept volume displaced= 3*$V_s$
- $\epsilon =$ number of crank revolutions for each power stroke, 1 for 2-stroke, 2 for 4-stroke and 3 for Wankel rotary engines

The computed BMEP map is then reported in Figure 5 as a function of the rotational speeds and throttle positions. The map presents a maximum BMEP of around 9.5bar at the maximum tested rotational speed of 6500 rpm and WOT, still highlighting the possibility of the engine going higher in terms of speed.
The brake specific fuel consumption (BSFC) and the related overall efficiency maps are reported in Figure 6 and Figure 7. As expected, the engine does not show high efficiency compared to common 4-stroke engines and presents a minimum BSFC of nearly 290 g/kWh, equivalent to a maximum efficiency of around 29%. Even if these values are not the best possible for internal combustion engines, it is worth mentioning that the engine is running on an ECU setup tuned for the aviation sector (unmanned aerial vehicles – UAVs) and no specific optimization was carried out in terms of fuel injection and ignition timings, with the latter mostly fixed at -18 degrees before top dead centre (BTDC) for all the speeds and loads. In addition, no energy recovery system is present at the exhaust where a large amount of thermal energy is rejected. Finally, no optimization was carried out on the oil flow rate injected for lubrication reasons. This may have affected combustion parameters and as a consequence also the overall efficiency of the engine. It will be more apparent from the subsequent sections that an optimisation of the aforementioned parameters, in addition to an exhaust energy recovery system (e.g. a turbocharger or a rotary device such as the one presented in [21]) will surely provide an enhancement to the performance and the efficiency of the machine.

Undoubtedly, combustion inefficiency played a role in reducing the overall efficiency of the engine. It is also possible to appreciate the effects of that on the emissions of carbon monoxide (CO) and unburned hydrocarbons (HC) as reported in Figure 8 and Figure 9, both in terms of g/kWh for easy comparison with the current range extenders on the market e.g. the Mahle Dedicated Hybrid Internal Combustion Engine (DHICE) [28], the latter being the latest iteration of range extenders produced by the that company. For sake of comparison, the aforementioned 2-cylinder engine can reach maximum efficiency of 40%, which gives a BSFC of 218 g/kWh when using the same lower heating value (LHV) of 41.28 MJ/kg as the fuel used in this project or 209 g/kWh if considering a corrected LHV of 43 MJ/kg as reported in another work from Mahle [29]. Although the fuel performance of the Mahle range extender is far better when compared to the rotary engine investigated in this work, it is important to state that the reciprocating 4-stroke is a turbocharged engine operating on the Miller cycle, employing cooled exhaust gas recirculation and using the latest jet ignition technology. In addition, the Mahle REx operates at higher BMEP compared to the AIE 225CS so possibly with increased mechanical efficiency and hence affecting the overall efficiency of the thermal machine.

Figure 8 and Figure 9 report the carbon monoxide (CO) and unburned hydrocarbons (HC) emissions map for the AIE engine respectively. It is important to clarify at this point that those emissions maps are not from the engine instrumented with the pressure transducers and mainly discussed here although they are from an engine of identical specification and control strategy operated at IAAPS. Furthermore, they were collected by using a MEXA 7000 gas analyser connected to the exhaust pipe at a point close to the exhaust flange and well before any silencer and/or after-treatment system. Hence, they represent the emissions coming from the engine which can be reduced using any after-treatment system, including a three-way catalyst, given the ability of the engine control unit to operate the engine at λ=1 (with λ defined as usual as the relative ratio between the actual air-fuel ratio (AFR) and the stoichiometric one).
Figure 8. AIE 225CS carbon monoxide (CO) emissions map.

Figure 9. AIE 225CS unburned hydrocarbons (HC) map.

The map of the oxides of nitrogen in Figure 10 looks particularly smooth with its peak at high speed and load, where, as expected, the gas temperature is likely to be the highest in the entire operating range. Finally, all the emissions maps are reported with the aim to provide updated data to the reader that may be useful for the validation of numerical models.

Figure 10. AIE 225CS oxides of nitrogen (NOx) map.

Indicated pressure analysis

Pressure signal processing

The details of the combustion analyser employed in this study are extensively covered in Part I and II [1,2] of this suite of papers. It is worth reporting to the interested reader that it relies on several pressure transducers radially installed along the two-lobe peritrochoidal housing as also shown in Figure 11 and Figure 12 and summarised with their measurements uncertainties in Table 2. The correct location of the pressure transducers was computed through an additional routine in order to ensure a pressure measurement overlap between two consecutive (in the direction of the rotor rotation) pressure transducers, hence allowing merging all the pressure readings and the building up of a single pressure cycle for a specific rotor flank. The output of this routine is a timing diagram for each pressure transducer. Also, each rotor flank has its timing diagram (three in total) but, given the geometrical periodicity of the rotor flanks, it is easy to realise that it is the same timing matrix with two phase shifts of 360 and 720 degrees. A graphical example of the timing matrix computed for this study and taking into account all the four pressure transducers employed is reported in Figure 13.

From a fluid dynamic point of view, the aforementioned routine helps in transforming the measurement from an Eulerian description of the fluid dynamic field (pressure measurement in a single point) to a Lagrangian one (pressure measurement of the fluid that is moving together with the rotor within the housing).

Figure 11. View of the housing and the positions of the pressure transducers: P1 inlet pressure transducer, P2 compression pressure transducers, P3 combustion pressure transducer, and P4 outlet pressure transducer. Inlet port at bottom right, exhaust port at bottom left. Rotor rotates counter-clockwise.

Figure 12. Sectional view and in particular of the pressure transducers installed for the compression and combustion phases. The spark plug tube is visible within the water jacket and between the pressure transducers.
The pressure traces collected by means of the dedicated data acquisition system (DAQ) were smoother when compared to the ones collected during the motoring test of the engine, with lower noise and fluctuations embedded, as a consequence of the pressure transducers working within their designed operating ranges. Also, the ensemble average operated over multiple cycles (≥100) further reduced the “turbulent” component of the measurement. Nevertheless, it was chosen to carry out the same post-processing flow as employed in Part II [2] since many of the computed parameters in this study rely on the pressure derivative that usually shows increased noise with respect to the original signal as a consequence of the derivation process. In a manner identical to the procedures reported in [12, 30-33] specific filters have been designed by analysing the Discrete Fourier Transform (DFT) of the pressure signals using a Fast Fourier transform (FFT) algorithm. An example of this is reported in Figure 14 at 5 bar BMEP and the three different rotational speeds of 3000, 4500 and 6000 rpm.

Table 2. Type and positions of the pressure transducers employed.

<table>
<thead>
<tr>
<th>Pressure Transducer</th>
<th>Position</th>
<th>Type</th>
<th>Measurement Type</th>
<th>Measurements Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>P1 – Inlet pressure transducer</td>
<td>128° ATDC</td>
<td>Kistler 4007D</td>
<td>Absolute pressure</td>
<td>Max deviation pressure: ≤±1% FSO</td>
</tr>
<tr>
<td>P2 Compression pressure transducer</td>
<td>132° ABDC (402°)</td>
<td>Kistler 6052C</td>
<td>Relative pressure</td>
<td>Linearity: ≤±0.3% FSO</td>
</tr>
<tr>
<td>P3 - Combustion pressure transducer</td>
<td>137° ATDC (677°)</td>
<td>Kistler 6052C</td>
<td>Relative pressure</td>
<td>Linearity: ≤±0.3% FSO</td>
</tr>
<tr>
<td>P4 – Outlet pressure transducer</td>
<td>142° BTDC (938°)</td>
<td>Kistler 4049B</td>
<td>Absolute pressure</td>
<td>Max deviation pressure: ≤±0.3% FSO</td>
</tr>
</tbody>
</table>

The indicated pressures were collected for all of the experimental points populating the operating map of the engine, i.e. in the speed range from 3000 to 6500 rpm and in the load (BMEP) range from 2 bar to WOT condition. As for the motoring analysis reported in [2], also in this work a minimum of 100 indicated cycles were collected and analysed for each operating point. To give an exhaustive and succinct insight on the engine performance, in this text the analysis of only three rotational speeds and three different loads that are representative of a matrix of experiments (reported in Table 3) covering from low load/low speed to high load/high speed conditions will be reported. Unfortunately, at 3000 rpm the engine was not able to reach a BMEP of 7 bar with the closest point at 6.7 bar. For sake of conciseness that point will be reported as 7 bar in the subsequent tables and plots, while obviously the real value will be considered when computing the engine performance parameters.

Figure 13. Pressure transducer timing diagram for rotor flank 1. The diagram shows the active and overlap time for several sensors. [1,2].

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Table 3. Experimental points investigated in this work.

<table>
<thead>
<tr>
<th>Experimental points investigated</th>
<th>Rotational Speed</th>
<th>BMEP [bar]</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>3000</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>4500</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>6000</td>
<td>3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.7 (maximum possible at this speed)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>7</td>
</tr>
<tr>
<td></td>
<td></td>
<td>7</td>
</tr>
</tbody>
</table>

Table 4. Cut-off frequencies employed for the signal post-processing.

<table>
<thead>
<tr>
<th>Filters cut-off frequencies</th>
<th>BMEP [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>3</td>
</tr>
<tr>
<td></td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>7</td>
</tr>
<tr>
<td>1 kHz</td>
<td>1.5 kHz</td>
</tr>
<tr>
<td>2 kHz</td>
<td>2 kHz</td>
</tr>
</tbody>
</table>
For this study a 6th order Infinite Impulse Response (IIR) Butterworth filter was chosen (as was also used in Part II [2]) because of its flat response and smooth transition to the rejected frequencies. As already stated, the original signals were treated by means of an ensemble average, then the signals were filtered with a zero-phase shift procedure in both the forward and reverse directions [34]. A direct comparison of the unfiltered and filtered pressure traces at 4500 rpm and 5 bar BMEP is reported in Figure 15. As is possible to appreciate the two curves are nearly the same with slight differences at the start of the combustion (SOC) and the cycle peak pressure points. Both the points have been highlighted in the additional magnified frames reported in Figure 15 while the rest of the diagrams show an overall good agreement and overlap.

As expected, the effect of the filters is considerably more evident in the pressure derivative computation, with an example of that reported in Figure 16 where it is possible to appreciate the rejection of the high-frequency noise coming from the pressure transducers and the measurements chain itself. In the same plot, it is also possible to appreciate the difference in noise produced by the different pressure transducers that employ different sensing technology: the P1 inlet and P4 exhaust transducers (initial and final part of the diagram) are piezo-resistive and produce less noise than the P2 and P3 piezo-quartz transducers employed for the compression and combustion phases (the central part of the diagram). Finally, the overall continuity of both the pressure and pressure derivative traces demonstrate the reliability of the combustion analyser and the pressure merging algorithm when operating under firing conditions.

**Heat release analysis**

The filtering process provided as output the pressure traces ready to be represented in the classic p-V diagrams together with their log-log equivalent. The overlapped diagrams for 4500 rpm and different BMEPs and 5 bar BMEP and different rotational speeds are only reported in the text to offer to the reader a succinct and exhaustive insight into the engine performance. Supplementary data and plots at other rotational speeds and loads are reported in the Appendix with the aim of providing the reader with a useful means for comparing experimental results and/or validating the numerical models.

It is important to underline here that the values of BMEP used as a swept parameter are inferred from the torque using Equation 1 presented earlier in the text.

Analysing the data reported in Figure 17 and Figure 18, where different BMEPs are considered at the same rotational speeds, it is possible to draw several conclusions. First of all the diagram at the highest BMEP (7 bar) presents a maximum pressure of around 31 barA (absolute pressure), far below what is experienced in modern 4-stroke engines, even in light of the fact that the AIE unit demonstrated a maximum BMEP of 9.5 bar at 6500 rpm and WOT condition (see Figure 5). Nevertheless, the pressure values reported for the AIE 225CS are much higher than those reported by Danieli et al. in [35] highlighting several technology advancements throughout the last few decades. The current experimental results are instead perfectly in line with those reported by Spreitzer et al. in [36] for a Wankel engine produced by Austro Engine GmbH, with a displacement of 404 cm³, an identical compression ratio of 9.6:1 and a similar power range and rotational speed (35kW at 6000rpm), showing a peak pressure of around 34 bar. In [36] the pressure traces for the Austro Engine unit show similar characteristics (SOC, exhaust discharge phase, peak pressure) to the AIE 225CS and are reported in Figure 19.
Figure 17. Comparison of the pressure diagrams at three different BMEPs of 3, 5 and 7 bar and a rotational speed of 4500 rpm.

Figure 18. Log-log diagrams of the data reported in Figure 17.

As also demonstrated in [35], part of the loss in maximum peak pressure is due to seal leakage, with the real cycle departing from the ideal one proportionally to the amount of the total leakage area. Subsequently, the diagram at 3 bar BMEP shows another peculiarity of Wankel engines: slow combustion speed. It is possible to appreciate this by inspecting the trend of the cycle after combustion TDC with the pressure rise slowly continuing into the expansion phase. This will be also demonstrated later in the section when the diagrams of net heat release and rate of heat release (ROHR) are reported. Together with the seal leakage, the slow combustion phenomena affect the pressure rise after the start of ignition (SOI) and eventually the maximum peak pressure which is significantly different from the common values found on 4-stroke engines.

Nevertheless, it partially represents a drawback when considering the maximum torque obtainable: studies on the minimum spark advances for best torque (MBT) demonstrated that it is mostly influenced by the pressure distribution within the cycle rather than only the cycle peak pressure.

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Apart from that, the AIE rotary unit presents conventional compression and expansion phases, with the exhaust blowdown phase dropping the chamber pressure to the ambient value in approximatively 25 degrees of eccentric shaft rotation (see Figure 15). More detailed information regarding all the cycle phases can be gathered by investigating the log-log diagrams in Figure 18, as also suggested in [37, 38]. In the figure it is possible to recognize essentially linear compression and expansion phases, apart at from the top of the diagram that is affected by the slow combustion as discussed. The linear trend of the compression and expansion demonstrates a fairly constant polytropic coefficient and a nearly-exact phasing of the pressure traces with the instantaneous chamber volume which was one of the aims of the work conducted in Part II [2]. In the log-log diagrams, it is also possible to highlight the substantial difference of the pumping loop that is hidden in the classic p-V diagram: as expected at fixed rotational speed the pumping loop at lower BMEP is fairly more extended than that at higher BMEP due to the losses of the throttle valve in the inlet pipe. That mainly affects the amount of air ingested by the engine, the amount of fuel injected and consequently, all the most important pressure points of the diagram, shifting it downwards in the reference frame.

The heat release analysis confirmed that the combustion is rather slow in Wankel engines. Again, similar results were obtained by [36] while, surprisingly, slightly faster combustion was found by Danieli et al. in [39]. For the AIE engine the rate of heat release was computed with the same classic methodology described in [40, 41] and by means of the usual Equation 2:

\[
\frac{dQ_{hr}}{d\theta} - \frac{dQ_{ht}}{d\theta} = \frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} \frac{dv}{d\theta} + \frac{1}{\gamma - 1} \frac{dp}{d\theta} \quad [J/deg] \quad \text{Equation 2}
\]

where:

- \(dQ_{hr}/d\theta\) = Total heat release [J/deg]
- \(dQ_{ht}/d\theta\) = Heat transfer losses [J/deg]
- \(dQ_n/d\theta\) = Net heat release [J/deg]
- \(\gamma\) = thermodynamic process polytropic coefficient [/]
p = Instantaneous pressure [Pa]
V = Instantaneous volume [m³]
θ = Eccentric shaft position [deg]

The numerical computations of the pressure and volume derivative terms were conducted by using a central scheme to increase the accuracy while a specific routine able to consider two different polytropic coefficients during compression and expansion phases was implemented. Specifically, the routine was also able to detect the absolute maximum pressure and switch between the two polytropic coefficients by smoothly blending the transition in a linear fashion to mitigate any discontinuity in the ROHR diagram. The length of the transition is up to the user and for this work a value of 300 samples (over the total of 12288 samples available for an entire cycle) was used. The polytropic coefficients used to compute the ROHR were determined in the Part II study at three different coolant temperatures of 30, 60 and 90°C [2, 42]. Given the firing operation assessed in this work, and hence that the coolant conditions are more similar to the 90°C presented in the previous work, it was chosen to use an average of the compression and expansion coefficients found for that coolant temperature, equal to 1.27 and 1.38 respectively. The results of the aforementioned procedure are reported in Figure 20 for the same rotational speed of 4500rpm and the three analysed BMEPs.

![Figure 20. Rate of heat release (ROHR) for the operating conditions of 4500 rpm and the three BMEPs of 3, 5 and 7 bar.](image)

As predicted, the combustion speed is higher at the highest load, decreasing in intensity at lower loads. Impressively, the peak values found for the ROHR are far below those found for modern 4-stroke engines where those values can be of a magnitude of 120 J/deg [40]. In addition, the fairly complex shapes of the ROHR curves denote different flame speeds during the combustion: they suggest that the classic Wiebe function may predict the indicated cycle with low accuracy while a double-Wiebe function would suit better when it comes to simulating and predicting the performance of the engine under test [43, 44]. The peculiar trend of the ROHR curves is also reflected in the shape of the cumulative sum for the net heat release curve reported in Figure 21, where the central part of the diagrams presents some oscillating behaviour. Nevertheless, all three curves present a similar trend even if the final values are different and equal to ~260J, ~380J and ~480J for 3, 5 and 7 bar BMEP respectively.

![Figure 21. Net heat release for the operating conditions of 4500 rpm and the three BMEPs of 3, 5 and 7 bar.](image)

Finally, the slow combustion speed events were also highlighted and confirmed by the pressure derivative of the three indicated pressure cycles, which at the highest load presents a peak of around 0.7 bar/deg, still far below what is experienced on modern 4-stroke engines. As a consequence, the combustion duration for the AIE 225CS engine sits in a range from 120 to 140 degrees of eccentric shaft rotation, as is also possible to deduce from Figure 21, which can be considered as an analogue of the mass fraction burned (MFB) curve.

![Figure 22. Pressure derivative for the operating conditions of 4500 rpm and the three BMEPs of 3, 5 and 7 bar.](image)

Different conclusions can be drawn by inspecting the previously-reported parameters and diagrams, this time at the same BMEP and different rotational speeds. Generally, as per the BMEP definition, at constant BMEP values, similar peak pressures and pressure cycles are expected for the same engine at different rotational speeds. That was confirmed both by the standard and log-log p-V diagrams presented in Figure 23 and Figure 24 where the pressure cycles mostly overlap. Slightly lower peak pressure for the highest rotational speed can be the result of counteracting effects due to the instantaneous rotor velocity. Undoubtedly, the loss of peak pressure is recovered by a slightly higher pressure along the expansion phase which gives in the end the same BMEP. It also demonstrates that the rotor speed affects the distribution of the heat released during the combustion event as is also possible to appreciate in Figure 25. Specifically, increasing the engine rotational speed the combustion presents a slower speed with its peak value shifted later about the eccentric shaft position. This
phenomenon may be due to reduced turbulence at higher rotor speed in addition to the gas being mostly compressed in the trailing edge of the rotor itself while the flame is stretched with increased strength in the direction of the rotation. Again and without any doubt all these hypotheses need to be proved ideally by designing a rotary test engine with optical access or, more conveniently, by employing modern CFD methods.

In the log-log diagram, the main difference is related to the pumping loop that is more evident at higher speeds. From the first principle of the thermodynamics and fluid-dynamics principles, considering the pipe and the throttle valve as an equivalent nozzle connected to a closed system with a moving boundary (i.e. the rotor flank) the lower pressure in the pumping loop is because at higher rotational speeds the derivative of the volume is larger hence inducing increased pressure losses.

As already mentioned above, the ROHR curves at the same BMEP and different speeds present a different distribution with the peak values shifted later for higher rotational speeds as is possible to appreciate in Figure 25. Those values are of the same order of magnitude as those reported for the case of the same speed and different BMEPs, and still far lower than is typical for current 4-stroke spark ignition engines.
An interesting analysis presented in [40] relates the 0-10% MFB duration to the 10-80% MFB. This analysis aims to relate the duration of the initial laminar combustion that is represented by the initial 10% MFB to the speed of what is considered the fully turbulent combustion period represented by the subsequent 70% of the mass burned. Usually, in 4-stroke engines when combustion starts slow it continues with a slow speed. Representing the previously analysed data on a similar scatter plot as reported in [40] it is possible to see that the combustion mechanism is fairly different from that in a common 4-stroke spark ignition engine: in Figure 28, the data shows that the laminar combustion duration is much shorter than the turbulent one (generally about half), confirming that the thermo-fluid dynamic field inside Wankel engine chambers do not induce any significant turbulence and so do not favour increased combustion speed, therefore affecting the performance and emissions of the engine itself. For a direct comparison, Figure 29 reports the equivalent data from [40] for a common 4-stroke engine. The present authors suggest that adoption of modern combustion technologies, i.e. pre-chambers, or employing a repeatable flow field induced by mixing-controlled combustion may help in increasing the turbulence produced by the combustion phenomenon.

Finally, a simple analysis of the instantaneous gas temperature has been carried out by using a single zone model when the inlet and exhaust ports are closed and the system can be assumed as closed with no (or negligible) mass transfer. The computation of the temperature was possible due to the computation of the air trapped by taking into account the amount of fuel injected and the fact that the ECU strictly worked at $\lambda=1$. The instantaneous temperature was then computed by applying the ideal gas law using the total mass of air and fuel inside the chamber. Although a simplified model was adopted for this analysis, the results obtained and reported in Figure 30 and Figure 31 are in line with what has been presented in the literature for both 4-stroke and Wankel rotary engines [39-41,45].

Unfortunately, it was not possible to carry out any analysis regarding the minimum advance for best torque (MBT) given the ECU setup with a SOI angle fixed at -18 degrees BTDC and experimental calibration activities were outside of the scope of the project. Adjusting spark advance to MBT would be interesting to investigate in a subsequent project, in order to verify any related combustion speed improvement as well. It is hoped that this can be considered as further experimental work together with the adoption of alternative fuels and there effect on the combustion performance of the rotary engine with its inherent geometrical and kinematics characteristics.
IMEP computation and its COV

The measurement of the indicated cycles gave a deep insight into the complex combustion mechanism evolving in the Wankel engine as exemplified by the AIE 225CS. The analysis of the cycles carried out in the previous section highlighted the low combustion speed, as already broadly described. Additionally, another important parameter that it is possible to infer from the indicated cycle is the Indicated Mean Effective Pressure (IMEP), computed through the commonly used Equation 3:

$$IMEP = \frac{W_i}{V_d} = \frac{\int p_i dV}{V_d} \quad [\text{bar or Pa}] \quad \text{Equation 3}$$

where:

- $W_i$ = Indicated work per cycle [J]
- $V_d$ = Swept volume [m$^3$]
- $p_i$ = Indicated pressure [Pa or bar]

The results of the computation are reported in Table 5 where it is also possible to make a direct comparison between the aforementioned IMEP at the different BMEP points. It is worth recalling here that the computed IMEP is the net one over 1080° of eccentric shaft rotation, hence it already includes the Pumping Mean Effective Pressure (PMEP). The difference between the IMEP values reported in the table and the correspondent BMEP are representative of the sum of the Ancillary (or Accessory) Mean Effective Pressure (AMEP) and the Rubbing Friction Mean Effective Pressure (RFMEP) that were measured and presented in previous work in this suite of papers [2].

Table 5. Indicated Mean Effective Pressure computed from the indicated cycles

<table>
<thead>
<tr>
<th>IMEP</th>
<th>BMEP [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>3</td>
</tr>
<tr>
<td>Rotational Speed</td>
<td>3000</td>
</tr>
<tr>
<td>3000</td>
<td>3.45 bar</td>
</tr>
<tr>
<td>4500</td>
<td>5.5 bar</td>
</tr>
<tr>
<td>6000</td>
<td>7.25 bar</td>
</tr>
</tbody>
</table>

The correct computation of the IMEP heavily relies on the TDC adjustment carried out and largely described in Part II [2], which presented an estimated error on the mechanical TDC of +2.66 degrees. As proof for the methodology adopted, a computation of the IMEP was carried out also considering the incorrect mechanical TDC: the results showed a difference in the range from 6 to 10% between the incorrect and the correct IMEP, confirming what was reported in [46-51].

From an engineering point of view, the coefficient of variation (COV) of the IMEP represents another important and useful parameter to evaluate the regularity and the variability of the combustion of the engine (and hence of its operation). It is defined as in Equation 4:

$$COV_{IMEP} = \frac{StdDev_{IMEP}}{IMEP} \cdot 100 \quad \% \quad \text{Equation 4}$$

with the StdDev_{IMEP} defined as in Equation 5:

$$StdDev_{IMEP} = \sqrt{\frac{n \sum_{i=1}^{n} (IMEP_i - \overline{IMEP})^2}{(n-1)}} \quad [\text{-}] \quad \text{Equation 5}$$

where:

- $n$ = number of computed IMEP samples [\-]
- $IMEP$ = Mean IMEP from the total number of samples [bar or Pa]

Usually, values in the range from 3 to 5% are acceptable for modern single-cylinder engines and are considered the industry standard. Surprisingly, the AIE 225CS demonstrated interesting low values of COV as proof of its regularity of combustion, especially at high speeds and loads: see Table 6. The values reported in Table 6 were computed by considering the IMEP from all three flanks of the rotor. As a consequence, the low values not only demonstrated that in general the combustion in the 225CS is regular with low variability, but also that all three flanks presented similar indicated pressure cycles and combustion characteristics, confirming the optimal design and conditions of the side and apex seals together with the housing and the side plates of the engine. Also, very low variation of COV of IMEP could be a not-often-considered contribution to the very smooth operation of Wankel rotary engines. Finally, given the approach in computing the COV values using an automatic routine able to analyse the indicated pressure cycles, it would be easy to compute the COV for each flank if this parameter is of interest in future studies or as an index of the health status of the engine itself.

Table 6. Coefficient of Variation of IMEP and their Standard Deviations (in brackets) for different speeds and loads

| IMEP Coefficient of Variation (COV) averaged on the three flanks (Std. Dev. in brackets) |
|-----------------------------------------------|-----------------------------------------------|
| Rotational Speed | BMEP [bar] | 3 | 5 | 7 |
|-----------------------------------------------|-----------------------------------------------|
| Rotational Speed | 3000 | 3.88% (0.13) | 3.88% (0.21) | 1.68% (0.12) |
| 4500 | 2.08% (0.08) | 1.64% (0.09) | 1.28% (0.11) |
| 6000 | 2.58% (0.1) | 1.76% (0.1) | 1.35% (0.11) |

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10/19/2016
**SPARCS pressure**

The SPARCS that forms the architecture of the engine core was instrumented with a fast response pressure transducer and the crank-resolved data were logged with the same sampling rate as the main pressure transducers used in the combustion analysis within this work. As reported in Part II [2], pressure measurement in the core showed a nearly steady-state behaviour at fixed operating points. In a manner different from what was expected, the pressure did not present fluctuations related to the pressure cycles evolving in the main chambers. The only tangible mechanical effect (neglecting here the effects on the emissions that are covered in [20]) of the firing conditions was increased core pressure with respect to the values recorded in the previous work [2]. Those pressure values are summarised in Table 7 where they are shown to be dependent only on load rather than the engine rotational speed.

Table 7. SPARCS pressure values at different rotational engine speeds and loads.

<table>
<thead>
<tr>
<th>SPARCS Pressure</th>
<th>BMEP [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotational Speed</td>
<td>3</td>
</tr>
<tr>
<td>3000</td>
<td>3.05 bar</td>
</tr>
<tr>
<td>4500</td>
<td>3.04 bar</td>
</tr>
<tr>
<td>6000</td>
<td>3.27 bar</td>
</tr>
</tbody>
</table>

**Indicated Torque**

The indicated pressure cycles also give the opportunity to compute the instantaneous indicated torque using the trigonometrical relation reported in both [24,25]. To proceed with the computation, the rotor flank area and the engine eccentricity values are needed. For the AIE 225CS, they are 6569.1mm² and 11.6mm respectively. Examples of the computed total torque produced by all three flanks are reported in Figure 32 and Figure 33 for the same conditions as the diagrams reported earlier in this section. The same and obvious considerations regarding the already-analysed pressure cycles are valid here: at a constant speed and different loads the torque produced is proportional to the load while at the same brake load the indicated torque presents similar values at different rotational speeds. For sake of conciseness here Figure 32 and Figure 33 report only the two aforementioned examples while a full set of averaged values of the instantaneous torque is given in Table 8.

Table 8. Average indicated torque at different rotational speeds and loads.

<table>
<thead>
<tr>
<th>Average Indicated Torque</th>
<th>BMEP [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotational Speed</td>
<td>3</td>
</tr>
<tr>
<td>3000</td>
<td>13.19 Nm</td>
</tr>
<tr>
<td>4500</td>
<td>13.63 Nm</td>
</tr>
<tr>
<td>6000</td>
<td>14.38 Nm</td>
</tr>
</tbody>
</table>

Analysing Table 8 the same considerations can be drawn as earlier regarding the IMEP computation: the difference in torque between the indicated values reported in Table 8 and the brake ones is due to the torque absorbed by rubbing friction and the engine ancillaries.

**Conclusions**

The current study assessed the performance of the AIE 225CS Wankel rotary engine under firing conditions with the help of all the specific tools and methodologies developed within the ADAPT-IPT research project. These in turn allowed the analysis of the indicated
pressure cycles through current and advanced digital technologies, decades after the pioneering works of Danieli et al. and Nguyen et al. [35, 52].

During the experimental activities, the general mechanical performance of the engine was recorded at different speeds and loads under steady-state conditions. Several indicated pressure cycles were collected at each investigated operating point in addition to the classic mechanical parameters. With its 10kg mass and nearly 25kW of power at 6500rpm, the AIE 225CS demonstrated the necessary characteristics of light weight and high power density which make it an ideal candidate as a range extender in advanced and innovative powertrains. Also, from an emissions point of view, the engine showed itself to be not very far from the 4-stroke technology that is considered the current state-of-the-art, and in any case, discredited the common misconception of producing a disproportional large amount of harmful gases with which to heavily impact the environment. Nevertheless, many expedients may be adopted to further improve the performance while reducing the emissions, e.g. a specific engine control strategy on oil metering, and injection and ignition timing optimisation in addition to a dedicated after-treatment system which can reduce the emissions to a level acceptable for current Euro 6 regulations.

The analysis of the indicated pressure cycles gave deeper insights into the combustion mechanism evolving within the machine. It is apparent from the analysis conducted that the combustion speed is fairly low even if presenting very good regularity and low variability (at least as load is increased). Without any doubt, this is the consequence of the specific geometry and kinematics of the machine coupled to additional counteracting effects related to heat transfer and internal fluid dynamics, as also extensively reported in the text. As a consequence, the rate of heat released to the working fluid was affected too by the low combustion speed and this resulted in experimental curves of complex shapes, more similar to what it is possible to simulate with a double-Wiebe heat release function. Similarly, the rate of pressure rise was limited too, being an order of magnitude lower than what it is possible to experience in a common 4-stroke engine. Nevertheless, the conjoined contributions of combustion inefficiencies and heat transfer were not so high as to ruin the overall engine efficiency which has its peak at around 29%. Detailed analysis on the heat and work fluxes was out of the scope of the current study work and will be carried out in subsequent work. Once again, many possibilities exist to improve the thermal performance of Wankel engines, but in this case, a specific engine management and control strategy could play an important role in enhancing the overall engine performance. Most importantly, the adoption of innovative fuels such as hydrogen (H₂) and bespoke direct injection systems could significantly increase the combustion speed given the inherent and well-known high flame speed characteristics. In addition, the knock-resistance and the absence of carbon in the chemical composition of hydrogen make are considered to make it an ideal candidate for rotary engine technology when it comes to improving efficiency by increasing the compression ratio and removing the production of CO, CO₂ and HC emissions from fuel combustion. Finally, with its inherently slow combustion as demonstrated here, the engine geometry itself may mitigate the effect of the violent combustion that is usually reported with hydrogen, while the stability of the engine performance when operated in lean conditions (λ>2 for limiting NOx production and emissions) needs to be evaluated first by means of CFD methods and subsequently by testing a newly-designed engine that employs all the aforementioned upgrades.

This work extends the presentation of the experimental activities and results collected on the AIE 225CS equipped with fast-response pressure transducers and within the ADAPT-IPT project. Subsequent work will be carried out mainly through analytic and numerical methods. As already mentioned, modelling activities were carried out in parallel to the experimental ones: a specific 0D/1D Wankel rotary engine model was developed employing the exact analytical relations for that class of engines, hence avoiding the usual trend of modelling the Wankel engine as a three-cylinder 4-stroke engine. Furthermore, the flexibility embedded in the numerical model allows for simulating different Wankel configurations (1:2, 2:3, …, n+1) or even a single operating flank. The experimental data reported so far in both the current and the previous works will be of great help in the validation of the aforementioned model as well as for more advanced 3D CFD models. Concluding, the validated models of the standard engine will be used to preliminarily investigate the benefits of using different fuels such as hydrogen and/or different injection and ignition systems and strategies.

References


Rail Fuel Feed System”, IOP Conf. Ser.: Mater. Sci. Eng. 327 022053, 2018


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Dr Vorraro intends to express his personal gratitude to Dr David Rogers from Kistler for providing useful advice in addition to the technical topics included in his book.

All the illustration of the engine are kindly provided by Advanced Innovative Engineering UK.

Definitions/Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>ABDC</td>
<td>After Bottom Dead Centre</td>
</tr>
<tr>
<td>AFR</td>
<td>Air-Fuel Ratio</td>
</tr>
<tr>
<td>AMEP</td>
<td>Ancillary (or Accessory) Mean Effective Pressure</td>
</tr>
<tr>
<td>ATDC</td>
<td>After Top Dead Centre</td>
</tr>
<tr>
<td>BBDC</td>
<td>Before Bottom Dead Centre</td>
</tr>
<tr>
<td>BEV</td>
<td>Battery electric vehicle</td>
</tr>
<tr>
<td>BMEP</td>
<td>Brake Mean Effective Pressure</td>
</tr>
<tr>
<td>BSFC</td>
<td>Brake Specific Fuel Consumption</td>
</tr>
<tr>
<td>BTDC</td>
<td>Before Top Dead Centre</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamic</td>
</tr>
<tr>
<td>CO</td>
<td>Carbon monoxide</td>
</tr>
<tr>
<td>COV</td>
<td>Coefficient of Variation</td>
</tr>
<tr>
<td>DAQ</td>
<td>Data Acquisition System</td>
</tr>
<tr>
<td>DFT</td>
<td>Discrete Fourier Transform</td>
</tr>
<tr>
<td>DHICE</td>
<td>Dedicate Hybrid Internal Combustion Engine</td>
</tr>
<tr>
<td>ECU</td>
<td>Engine Control Unit</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier Transform</td>
</tr>
<tr>
<td>FMEP</td>
<td>Friction Mean Effective Pressure</td>
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<tr>
<td>FSO</td>
<td>Full Scale Output</td>
</tr>
<tr>
<td>HC</td>
<td>Hydrocarbons</td>
</tr>
<tr>
<td>Abbreviation</td>
<td>Full Form</td>
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<td>--------------</td>
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<tr>
<td>HEV</td>
<td>Hybrid Electric Vehicles</td>
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<tr>
<td>ICE</td>
<td>Internal Combustion Engine</td>
</tr>
<tr>
<td>IIR</td>
<td>Infinite Impulse Response</td>
</tr>
<tr>
<td>IMEP</td>
<td>Indicated Mean Effective Pressure</td>
</tr>
<tr>
<td>LHV</td>
<td>Lower Heat Value</td>
</tr>
<tr>
<td>MBT</td>
<td>Minimum Advance for Best Torque</td>
</tr>
<tr>
<td>MFB</td>
<td>Mass Fraction Burned</td>
</tr>
<tr>
<td>PMEP</td>
<td>Pumping Mean Effective Pressure</td>
</tr>
<tr>
<td>REx</td>
<td>Range Extender</td>
</tr>
<tr>
<td>RFMEP</td>
<td>Rubbing Friction Mean Effective Pressure</td>
</tr>
<tr>
<td>ROHR</td>
<td>Rate of Heat Release</td>
</tr>
<tr>
<td>RPM</td>
<td>Revolution per minute</td>
</tr>
<tr>
<td>SOC</td>
<td>Start of Combustion</td>
</tr>
<tr>
<td>SOI</td>
<td>Start of Ignition</td>
</tr>
<tr>
<td>SPARCS</td>
<td>Self-Pressurised Air-Rotor-Cooling System</td>
</tr>
<tr>
<td>UAV</td>
<td>Unmanned Aerial Vehicle</td>
</tr>
<tr>
<td>WOT</td>
<td>Wide Open Throttle</td>
</tr>
</tbody>
</table>
Appendix

Figure 34. Comparison of the pressure diagrams at three different BMEPs of 3, 5 and 7 bar and rotational speed of 3000 rpm

Figure 35. Rate of heat release (ROHR) for the operating conditions of 3000 rpm and three BMEPs of 3, 5 and 7 bar.

Figure 36. Net heat release for the operating conditions of 3000 rpm and three BMEPs of 3, 5 and 7 bar.
Figure 37. Instantaneous indicated torque for the operating point of 3000 rpm and three BMEPs of 3, 5 and 7 bar.

Figure 38. Comparison of the pressure diagrams at three different BMEPs of 3, 5 and 7 bar and rotational speed of 6000 rpm

Figure 39. Rate of heat release (ROHR) for the operating conditions of 6000 rpm and three BMEPs of 3, 5 and 7 bar.
Figure 40. Net heat release for the operating conditions of 6000 rpm and three BMEPs of 3, 5 and 7 bar.

Figure 41. Instantaneous indicated torque for the operating point of 6000 rpm and three BMEPs of 3, 5 and 7 bar.

Figure 42. Comparison of the pressure diagrams at three different rotational speeds of 3000, 4500 and 6000 rpm and a BMEP of 3 bar.
Figure 43. Rate of heat release at three different rotational speeds of 3000, 4500 and 6000 rpm and a BMEP of 3 bar.

Figure 44. Net heat release at three different rotational speeds of 3000, 4500 and 6000 rpm and a BMEP of 3 bar.

Figure 45. Instantaneous torque at three different rotational speeds of 3000, 4500 and 6000 rpm and a BMEP of 3 bar.
Figure 46. Comparison of the pressure diagrams at three different rotational speeds of 3000, 4500 and 6000 rpm and a BMEP of 7 bar.

Figure 47. Rate of heat release at three different rotational speeds of 3000, 4500 and 6000 rpm and a BMEP of 7 bar.

Figure 48. Net heat release at three different rotational speeds of 3000, 4500 and 6000 rpm and a BMEP of 7 bar.
Figure 49. Instantaneous torque at three different rotational speeds of 3000, 4500 and 6000 rpm and a BMEP of 7 bar.