Abstract

The present work investigates a means of controlling engine hydrocarbon startup and shutdown emissions in a Wankel engine which uses a novel rotor cooling method. Mechanically the engine employs a self-pressurizing air-cooled rotor system (SPARCS) configured to provide improved cooling versus a simple air-cooled rotor arrangement. The novelty of the SPARCS system is that it uses the fact that blowby past the sealing grid is inevitable in a Wankel engine as a means of increasing the density of the medium used for cooling the rotor. Unfortunately, the design also means that when the engine is shutdown, due to the overpressure within the engine core and the fact that fuel vapour and lubricating oil are to be found within it, unburned hydrocarbons can leak into the combustion chambers, and thence to the atmosphere via either or both of the intake and exhaust ports. As well as shutdown it also affects the startup process, where higher hydrocarbon emissions are caused due to the forced transfer of the unburned gases to the intake and exhaust ducts as the core depressurizes across the sealing grid when it is stationary. These emissions then sit in those volumes, possibly then escaping to the outside world; clearly this is also very important with respect to the SHED testing of any vehicle the engine might be fitted to.

The SPARCS concept is discussed with respect to how it functions versus a conventional wet sump arrangement (as employed by oil cooled rotor Wankel engines). Measurements are taken and steady-state emissions and fuel consumption results with and without pressurization of the core are presented; such a comparison has not been made before. In general, power output, brake specific fuel consumption, hydrocarbon emissions, and combustion efficiency are all better with a depressurized core, with only small improvements in cooling (defined by rotor air inlet temperature) being apparent when it is pressurized. A hypothesis for why this should be so is developed, the knowledge of which can help to guide further development.

The reasons for the engine on/off hydrocarbon issue are apparent. Using a solenoid valve as a means of venting the rotor core pressure directly to the engine intake just before shutdown is proposed as a means of alleviating this problem. This approach would feed the hydrocarbon-rich gases from the core through the combustion process and out through the catalytic converter just before the engine is switched off. In automotive applications this engine is to be used as a range extender and hence there is a great degree of control regarding all modes of its operation, including startup and shutdown, which is the approach investigated for mitigation here.

The results show that depressurizing the core in this manner results in a maximum reduction in total hydrocarbon emissions during warm shutdown and restart of 80% and 60%, respectively. However, it must be remembered that with the pressure relieved in the core, the cooling capability there is slightly reduced, and so the approach has to be calibrated correctly to achieve the best result for the whole system.

Further investigation into the optimum level of pressurization is recommended.

Introduction

The Wankel rotary engine is one of the more promising alternatives to the conventional reciprocating types with regards to range extender (REx) applications in plug-in series hybrid vehicles. Such range-extended electric vehicles (REEVs) have the potential to maximize the total amount of passenger-miles that can be driven on electricity for a given value of installed traction battery manufacturing capability, because the battery can be sized to cover the significant majority of journeys undertaken while allowing longer journeys to still be possible. The resulting alleviation of range anxiety means that drivers will be more likely to use more of the total capability of the battery and so further maximize the use of the installed capacity; logically this therefore maximizes the investment of the energy and resources in it. The Wankel suits the REx role because, as discussed by Turner et al. [1] a primary attribute of such engines is that on a vehicle level its installation should be light. The compactness of the Wankel engine, coupled to its exemplary vibration characteristics, provides this attribute. In comparison to low installed mass, the efficiency of the REx is of lower importance, although it is understood that it still needs to be acceptable. These considerations undoubtedly support Mazda’s recent announcement of its adoption of the Wankel engine as a REx [2] for the series plug-in hybrid version of its MX-30 vehicle [3].

In [4,5] more than satisfactory fuel consumption was shown to be possible in Wankel engines provided an appropriate technology level is adopted and that their operating regime can be suitably constrained. Particularly in REx applications the latter is made possible by the fact that control of the REx is independent of vehicle operation by the driver. The equivalence of best fuel consumption in the full operating map was shown to be the case versus similar-technology reciprocating 4-stroke engines in [5] so in actuality, one may expect no disadvantage for the Wankel. Similarly, in [4] it was shown that there is no reason why, with the right technology, Wankel
enables engines cannot easily meet emissions limits, and so be a very viable engine type for REx use.

In their taxonomy, aside from the whether they are air- or water-cooled, mechanically Wankel engines are principally differentiated by how their rotors are cooled. The two types are the oil-cooled rotor (OCR), as exemplified by production and racing engines from Mazda [6,7] and Citroën and concepts from Daimler-Benz [8], and the air-cooled rotor (ACR), adopted by Fichtel and Sachs and further developed by Norton [9]. Advanced Innovative Engineering have developed air cooling further in the form of the Self-Pressurized Air-cooled Rotor System (SPARCS), and this will be described later. In terms of gas exchange Wankel engines again fall into two main types: those using peripheral ports and those using side ports. Chiefly this distinction is driven by the layout of the intake ports, although the exhaust port configuration also plays an important part, since with full peripheral porting overlap between the ports can be completely eliminated. The importance of minimizing this has been discussed by Ohkubo et al. [6] and Turner et al. [4,5]. Unfortunately, it is not easy to combine side ports which ACRs, as will be explained next.

**Rotor Cooling in Wankel Engines**

The AIE 225CS engine used in this work employs a novel rotor cooling system called SPARCS. As mentioned above Wankel engines have chiefy employed either oil or air cooling for their rotors. The OCR is conventionally combined with the use of a high-pressure lubrication system and employs plain bearings for the shaft, with the rotor having oil sprayed into it which then gyrates within it, absorbing heat, before being flung out and into the engine’s internal core (equivalent to the crankcase volume in a reciprocating engine). Conversely, an ACR system uses charge air with or without fuel in it flowing through the rotor (and typically also channels in the eccentric of the shaft) to cool itself internally. This air has historically been drawn through the core using intake depression, but both “exhaust ejector” and fan-driven systems have been used. An ejector is a device which uses the energy in one fluid stream to pull another into it, and was used successfully by Norton in their racing motorcycles [10]. With the last two options the rotor cooling air is effectively jettisoned overboard which, since rolling element bearings with total-loss lubrication are used in this case, means that this oil is unburned when it leaves the engine system; with the intake depression system the oil at least goes through the combustion system and is burnt with the charge. Blowby past the side seals is similarly either jettisoned or it goes through the combustion system depending on the configuration.

There is subset of the ACR in which the eccentric shaft is also cooled internally by water [11]. While this concept could beneficially use some of the elements discussed here it will not be discussed further in the present work.

ACRs have a number of potential advantages compared to OCRs. They do not need oil control seals and indeed there is less need for a second side seal for combustion gases because the blowby joins the rotor cooling stream; in an intake-air-depression system it then goes through the combustion system again. The simplification of the sealing grid, together with the removal of the need for a high pressure pump due to the use of rolling-element bearings, means that their friction is potentially significantly lower than for OCR configurations. Finally, there is no oil being flung around the eccentric shaft space adding windage effects (however, here it is important to note that pumping work in this area of a Wankel engine is significantly lower than in a reciprocating wet-sump engine because there is no equivalent of under-piston compression occurring when it is running).

As a result of the removal of the requirement for a second side seal and translating oil seals, and the associated need to protect for a track for these to cover, the geometry of an ACR can be different to that of an OCR, which can give some additional degrees of freedom to the engine designer. The subject of geometry and tradeoffs in the design of the Wankel engine are dealt with in numerous texts, including [12,13]. However, it should be noted that because of this and the requirement to provide cooling air to the rotor it is harder to arrange side ports, which are useful in reducing overlap, the importance of which has been covered by the authors in other publications [4,5].

**The Self-Pressurized Air-cooled Rotor System (SPARCS)**

The SPARCS system is similar to conventional air cooling but as discussed by Bailey and Lothan [14] it makes a significant change to it by closing the circuit and using an internal fan to circulate gases within the core. This circulation is through the rotor and eccentric shaft and then through a heat exchanger which can be a gas-to-air matrix external to the engine or, in the case of the “Compact SPARCS” system employed in the 225CS engine, a gas-to-coolant matrix integral with the rotor housing. A photograph of the AIE 225CS engine used in this work is shown in Figure 1, with the two different SPARCS systems and other pertinent engine features being shown in Figures 2 and 3. In both SPARCS configurations there are channels in the side housings through which the circuit operates.

Fig. 1: Photograph of the rear, or drive, side of the AIE 225CS engine as used in this work, also showing the intake and exhaust ports, the former being above the latter. Note that this end of the eccentric shaft employs an external balance weight, bolted to the propeller drive flange in this aero engine “pusher propeller” application. Engine rotation is counter-clockwise in this view.
The core gas is circulated by a fan mounted on the end of the eccentric fan and sealed in the core.

As standard, there is no flow path out from the core for either SPARCS configuration. During combustion there is a significant pressure differential from the working fluid side to the core over the side seals driving blowby gas flow. The accumulation of this blowby mass in the core is what causes the pressure there to increase, there being no vent. The gas pressure in the core builds up until there is an equilibrium between it and the processes simultaneously occurring in the rotor chambers. (Note that the corner seals also provide a small flow path from the rotor chamber to core, but henceforth it will be assumed that when any gas flows over the side seals are referred to that it is implicit that this also includes a small amount of corner seal flow.) Figure 4 shows a profile of the core pressure during engine operation at 8 bar and 4 bar brake mean effective pressure (BMEP); observations on the difficulty of measuring chamber pressure on each rotor flank will be returned to later. Note that although the pressures change considerably in the working chambers, the pressure in the core is remarkably constant.

![Fig. 2: Part-section of the conventional SPARCS System showing rotor cooling air feed to and return from an external heat exchanger. The view shown is from the front, or non-drive, side of the engine. Note the fan on the rotor side of the timing wheel, and also the front balance weight, all of which are internal and fixed to the eccentric shaft. Engine rotation is clockwise in this view.](image1)

![Fig. 3: Part-section of the Compact SPARCS as used in the AIE 225CS engine employed in testing in this work, showing the flow of cooling air through the rotor core and the heat exchanger integrated into rotor housing. The view shown is from the front, or non-drive, side of the engine. Internal balance weight, timing gear, and fan positions on the eccentric shaft are all per the conventional SPARCS arrangement shown in Figure 2.](image2)

The advantages of Compact SPARCS include robustness of the system (there being no external joints) and compactness and low mass. Clearly the rotor housing is more complex, and the load on the liquid cooling system is increased. At the same time the total volume of the rotor core is reduced, and this can have advantages in cooling response, and as reported later, reduced time to relieve the core pressure.

![Fig. 4: Measured 225CS SPARCS pressure profiles for each individual rotor flank and the core at 5000 rpm. Top: 8 bar BMEP (indicative of full load); bottom: 4 bar BMEP. Note how the core pressure changes with load. For this test, the sensors used were all high speed ones (operating at 341.3 kHz): importantly there is no perturbation in the core pressure as the pressures in the individual working chambers change.](image3)
This increase in core pressure is theoretically intentional and beneficial. It increases primarily with load but also with speed, the increase in the number of molecules in the core theoretically giving an increased total heat capacity within the volume to the benefit of cooling capacity of the system (although, because of the amount of burned gas increases as a proportion of the whole, the specific heat capacity of the mixture overall does fall). Furthermore, the oil is no longer intentionally jettisoned as it might be in a classical ACR configuration with an exhaust ejector, rather it is recirculated round the core by the eccentric-shaft driven fan, and so the amount of oil consumed can be reduced. There is no longer a positive oil feed into the intake air as per Mazda’s engines either; such oil as the ceramic apex seals employed require is delivered via backflow to the rotor chambers during the intake and overlap phases when their pressure is lower than that in the core.

SPARCS, then, potentially offers a significant improvement over a conventional air-cooled rotor system in terms of cooling capacity and oil consumption. However, there is some lag in pressure decay as the engine speed and load are reduced, as would be expected since the side seals effectively function as an orifice between volumes at different pressures. This means that on a rapid shutdown the core pressure could still be significantly higher than that in the now-stationary rotor chambers, and some of whatever gases are trapped in it could be forced across the side seals and then out through the ports. It is assumed that these gases will contain hydrocarbons (HCs), be they from incomplete combustion or from the lubricating oil that is fed into the core air flow. Hence, these gases become fugitive emissions and whatever do pass through any still-active catalyst could cause subsequent emissions on start up or emissions not unlike those captured during SHED (sealed housing evaporative determination) testing.

Fortunately, in a REx application there is the opportunity to completely control engine switch off and so to avoid such a hard shutdown. Reducing the engine speed and load gradually would be one way to mitigate this, but this would be expected to come with a fuel consumption penalty because while its full load fuel consumption is essentially the equal of a similar-technology 4-stroke reciprocating engine, the Wankel engine suffers significantly as load and speed are reduced due to increasing losses from sealing and heat transfer [4,5]. This work therefore investigated another possible solution, that of opening a vent from the core to the intake runner during shutdown, such that the core gases flow through the combustion chambers and catalyst system and therefore get treated before emission to the atmosphere. When the engine is switched off the pressure in the chambers is equal to that in the core. The efficacy of this approach is described in the following sections; this has not been investigated before, so is important in understanding whether a pressurized core system such as SPARCS is viable in a range extender application of the Wankel engine.

This approach would apply to any engine with the SPARCS system, although the system volume is obviously important with respect to how quickly the engine can be brought down to idle and switched off with all of the over-pressure core gas having been treated. For the Compact SPARCS configuration this will not take as long due to its smaller core volume, and since the engine used for the testing employed that system the remainder of this paper will only be concerned with results using the compact arrangement.

Test Equipment

As stated, the engine used for this work was an AIE 225CS single-rotor Wankel, the specifications of which are shown in Table 1. It is the main engine studied within the ADAPT-IPT project funded by the UK government. The engine was fitted to a dynamometer at the University of Bath and mapped for fuel consumption and emissions with it operating in both the normal, pressurized core state and also in a depressurized form. Such a mapping of the engine performance has not been presented before and will provide important information on how to develop the system further. Details of the test equipment used are shown in Table 2, with further detail of the Horiba MEXA 7000 Gas Analyzer used given in Table 3. Particulate emissions were not recorded, this being a port-fuel injected engine.

Table 1: Engine specifications

<table>
<thead>
<tr>
<th>Engine Type</th>
<th>225CS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Configuration</td>
<td>Single-rotor Wankel</td>
</tr>
<tr>
<td>Cooling system</td>
<td>Water-cooled housings</td>
</tr>
<tr>
<td></td>
<td>Air-cooled rotor using SPARCS with air-to-coolant heat exchanger integrated into trochoid housing</td>
</tr>
<tr>
<td>Generating radius (R) (mm)</td>
<td>69.5</td>
</tr>
<tr>
<td>Generating eccentricity (e) (mm)</td>
<td>11.6</td>
</tr>
<tr>
<td>Housing width (B) (mm)</td>
<td>52.2</td>
</tr>
<tr>
<td>Compression ratio (geometric) (:1)</td>
<td>9.6</td>
</tr>
<tr>
<td>Fuel system</td>
<td>Port-fuel injection</td>
</tr>
<tr>
<td>Aspiration</td>
<td>Naturally aspirated</td>
</tr>
<tr>
<td>Port configuration</td>
<td>Peripheral intake and exhaust ports</td>
</tr>
<tr>
<td>Spark plug configuration</td>
<td>Two spark plugs in parallel, leading position</td>
</tr>
<tr>
<td>Engine management system (EMS)</td>
<td>GEMS</td>
</tr>
</tbody>
</table>

Table 2: Test equipment

<table>
<thead>
<tr>
<th>Dynamometer</th>
<th>ABB VM25 AC motor dynamometer, 48.2 kW at 8500 rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Emissions measurement</td>
<td>Horiba MEXA 7000 Series Gas Analyzer</td>
</tr>
<tr>
<td>Fuel mass flow rate</td>
<td>Emerson Coriolis Micro Motion CMF010M, 93.5 kg/h</td>
</tr>
<tr>
<td>Coolant volume flow rate</td>
<td>Krohne Optiflux 5000 sensor, 36 l/min</td>
</tr>
<tr>
<td>Torque</td>
<td>HBM T40B torque flange, 100 Nm</td>
</tr>
</tbody>
</table>
gathered however, publications instead the subject is dealt constructing the indicator diagram moving working volume. The times, this is not a trivial task in single pressure transducer can see the pressure in the cycle at all.

The information including a spark plug pressure transducer, speed shown in Figure 5. A photograph of the engine on the test bed is shown in Figure 5, with the location of the core pressure tapping shown. This is in the passage in the rear end plate leading from the outlet of the heat exchanger to the rotor cooling section; it therefore records the gas temperature just prior to when it cools the rotor. Note that the engine shown in Figure 5 is heavily instrumented with several other high-speed pressure transducers disposed around the trochoid housing, including a spark plug pressure transducer, these being used to gather the information necessary to construct the indicator diagram for the engine. Compared to a conventional reciprocating engine, in which a single pressure transducer can see the pressure in the cycle at all times, this is not a trivial task in a Wankel engine with its constantly moving working volume. The equipment used and method of constructing the indicator diagram is not discussed further here; instead the subject is dealt comprehensively with in other publications [15,16,17] to which the reader is referred. It was, however, this approach that let the information shown in Figure 4 be gathered, with all of the measurements being taken at 341.3 kHz.

Table 3: Operating principle, measurement range, and sensitivity for each emissions gas for the Horiba MEXA 7000 Series Gas Analyzer used (NDIR: Non-dispersive infrared; FID: Flame ionization detector; HCLD: Heated chemiluminescence detector)

<table>
<thead>
<tr>
<th>Gas</th>
<th>Operating Method</th>
<th>Measurement Range</th>
<th>Resolution</th>
<th>Span Gas</th>
<th>Zero Gas</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO</td>
<td>NDIR (dry)</td>
<td>0-12%</td>
<td>0.01%</td>
<td>CO/N₂</td>
<td>N₂</td>
</tr>
<tr>
<td>CO₂</td>
<td>NDIR (dry)</td>
<td>0-20%</td>
<td>0.5%</td>
<td>CO/N₂</td>
<td>N₂</td>
</tr>
<tr>
<td>THC</td>
<td>FID (hot-wet)</td>
<td>0-50000 ppmC₁</td>
<td>10 ppmC₁</td>
<td>C₃H₈</td>
<td>N₂/air</td>
</tr>
<tr>
<td>NOx</td>
<td>HCLD (dry)</td>
<td>0-10000 ppm</td>
<td>10 ppm</td>
<td>NO/N₂</td>
<td>N₂</td>
</tr>
</tbody>
</table>

Fig. 5: Test engine fitted to test bed. The core pressure tapping point position is as indicated. The drive end is to the right-hand side, with the SPARCS fan to the left within the front end plate housing on the non-drive side of the engine. The externally-mounted rear balance weight is visible pointing towards the camera. Note the twin-spark plug configuration used (necessary for aero engine applications), the fact that these are in a leading position, and that here one of these spark plugs is also a pressure transducer. Further instrumentation is discussed in the text.

The test operating points used are shown in Table 4. Below wide-open throttle (WOT) fixed BMEP operating points were used; at WOT the BMEP that the engine generated for each of the pressurized and depressurized cases are listed separately in the table.

Table 4: Operating points in terms of engine speed and load (BMEP) for pressurized (P) and depressurized (De) core tests with, where appropriate, the wide-open throttle BMEP achieved in brackets

<table>
<thead>
<tr>
<th>Engine Speed (rpm)</th>
<th>BMEP (bar)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3000</td>
<td>3 4 5 6 WOT (P-6.3, De-6.5)</td>
</tr>
<tr>
<td>3500</td>
<td>3 4 5 6 WOT (P-6.6, De-7)</td>
</tr>
<tr>
<td>4000</td>
<td>3 4 5 6 7 WOT (P-7.2, De-7.5)</td>
</tr>
<tr>
<td>4500</td>
<td>3 4 5 6 7 WOT (P-7.5, De-8)</td>
</tr>
<tr>
<td>5000</td>
<td>3 4 5 6 7 8 De-8 WOT (P-7.7, De-8.2)</td>
</tr>
<tr>
<td>5500</td>
<td>3 4 5 6 7 8 WOT (P-8.2, De-8.7)</td>
</tr>
<tr>
<td>6000</td>
<td>3 4 5 6 7 8 9 De-9 WOT (P-8.7, De-9.5)</td>
</tr>
</tbody>
</table>
The fuel used was standard 95 RON pump gasoline, and the oil was the type specified by AIE, being standard Mobil Pegasus 1 (SAE 15W–40). All operating parameters were kept as mapped in the engine management system (EMS) during prior testing, which itself targeted stoichiometric operation everywhere, so that a simple three-way catalyst can be used for exhaust emissions control. The actually-achieved lambda value (or the relative air-fuel ratio (AFR)) will be discussed later.

Results and Discussion

Steady-state results

This section comprises results taken under steady-state conditions with both the conventional pressurized core arrangement and also with it being depressurized by venting it to the engine air intake from the oil return line to the tank.

All the results are given versus BMEP and engine speed, with all emissions results presented as engine-out (or raw). Results are presented in pairs of maps, with the standard pressurized core case first and bearing the suffix (a), and the corresponding results for the case with the depressurized core being given the suffix (b). This has been done to facilitate more rapid comparison between the two cases by the reader.

Note that the 225CS can operate at engine speeds considerably in excess of those shown in these figures, the limit of 6500 rpm being used because that speed matches the generator being used in this REx application.

Core pressure

Results for the steady-state pressure in the core are given in Figures 6(a) and (b). Although the core was vented to the intake in the depressurized system, some pressure was nonetheless generated in the core at higher speed and loads. This is due to the fact that blowby increases with speed and load (as it does with reciprocating engines) and that instead of the vent line being directly connected with the core it is connected to the oil return line from its port. This causes a build up of pressure in the core due to an orifice effect. Further work may wish to investigate using a larger flow route with orifices of different sizes, or some controlled valve system, in order to influence this effect, and possibly control it.

Fig. 6(a): Contour plot of average SPARCS core pressure (in bar (gauge)) under steady-state operation with a pressurized core for the entire engine operating map

Fig. 6(b): Contour plot of average SPARCS core pressure (in bar (gauge)) under steady-state operation with a depressurized core for the entire engine operating map

The maximum core pressure for the conventional SPARCS operation is in the range of 1.4–4.3 bar (gauge) as can be seen in Figure 6(a). Up to a speed of 5500 rpm the core pressure was generally proportional to load and beyond this speed it starts to increase slightly with increasing speed. Higher speed allows less time for the blowby gases to leak past the side seals and consequently increases the pressure for similar load points as the speed increases. The steady-state core pressure map for the depressurized system presented in Figure 6(b) indicates that a much lower pressure up to 1.38 bar (gauge) was generated in the core for the full load conditions over 5000 rpm. For most of the low to medium loads the pressure is close to atmospheric, as might be expected.

Brake specific fuel consumption

Brake specific fuel consumption (BSFC) under the two steady-state conditions studied is shown in Figures 7(a) and (b). Note that the BSFC is in general very similar for the two cases, although the depressurized core appears to be slightly better above about 5 bar BMEP (by approximately 5 g/kWh, or about 1.6%). This will be discussed later.
Fig. 7(a): Contour plot of BSFC (in g/kWh) under steady-state operation with a pressurized core for the entire engine operating map

Fig. 7(b): Contour plot of BSFC (in g/kWh) under steady-state operation with a depressurized core for the entire engine operating map

Fig. 8(a): Contour plot of THC emissions (in terms of ppmC1) under steady-state operation with a pressurized core for the entire engine operating map

Fig. 8(b): Contour plot of THC emissions (in terms of ppmC1) under steady-state operation with a depressurized core for the entire engine operating map

Fig. 9(a): Contour plot of CO emissions (%) under steady-state operation with a pressurized core for the entire engine operating map

Fig. 9(b): Contour plot of CO emissions (%) under steady-state operation with a depressurized core for the entire engine operating map

Criteria emissions

Emissions results are given in Figures 8(a) and (b), 9(a) and (b), and 10(a) and (b), for total hydrocarbons (THCs), carbon monoxide (CO), and oxides of nitrogen (NOx), respectively.

Fig. 10(a): Contour plot of NOx emissions (%) under steady-state operation with a pressurized core for the entire engine operating map

Fig. 10(b): Contour plot of NOx emissions (%) under steady-state operation with a depressurized core for the entire engine operating map
The general shape and magnitude of the CO and NOx maps are the same whether the core is depressurized or not, but the results for THCs are markedly different; indeed, note that a different scale has had to be used on the maps presented here. For the depressurized running the THC emissions were in the range of 6000-11000 ppmC1 compared to 10000-18000 ppmC1 for the pressurized core system. This is presumed to be due to reduced charge and oil leakage across the seals during the low pressure part of the cycle and possibly better combustion when running depressurized (see later).

In a peripherally-ported engine like this one, recently Pisnoy and Tartakovsky have shown how revised ignition arrangements and optimization of the rotor flank cutout can affect combustion [18]. They studied a three spark-plug configuration, with two in parallel at the leading position and one in a trailing position and showed that further improvements in combustion (and presumably therefore concomitant reductions in HC emissions), independent of the rotor cooling arrangement, may still be possible. This could be important for the 225CS engine, which as shown in Figure 5 also uses two spark plugs in parallel at the leading position as standard. Note that the configuration of Pisnoy and Tartakovsky is unlike the three-spark-plug configuration used by Shimizu et al. [7], where the spark plugs were all in a line along the centre line of the housing to enable combustion at high engine speed in order to generate maximum power.

Lambda, oxygen concentration, and combustion efficiency

Maps of relative AFR (lambda) for the two cases are shown in Figures 11(a) and (b). These values are calculated from the emissions gases and show a leaner AFR for low speed regions for the depressurized core. Even at higher speed and loads points, higher lambda values were achieved, in the range of 0.95-0.99 compared to the richer range of 0.94-0.96 for pressurized system.
Generally the oxygen concentration is higher with the core pressurized, and this difference occurs across the operating map, although it is greater at high loads. This will be discussed later.

Combustion efficiency is shown in Figures 13(a) and (b). It was up to 94% for the depressurized core compared to maximum of 90% for the pressurized case, being a relative improvement of 4.4%, with improvements evident across the map, and particularly at high speed and load.

As a consequence of these results one can say that the depressurized operation results in higher exhaust lambda, lower oxygen concentration in the exhaust, and improved combustion efficiency. Linking this to the lower hydrocarbon emissions presented earlier suggests that there is a change in intra-cycle mass transfer across the side seals when the core is pressurized versus depressurized, insofar as when it occurs in the cycle. The unburned air and fuel would be forced into the core firstly during compression, and then during combustion a proportion of what would normally be considered end gas is in fact blown past the side seals. If the core is depressurized, i.e. allowed to communicate with the intake, the charge makes its way to the inlet port. Conversely, when the core is pressurized, due to the markedly increased core pressure and particularly during the low-pressure part of the cycle, the charge mass can be transferred back past the seals into the working chambers. If this happens during the intake phase then the gases are recycled, but if it occurs in the latter part of the expansion and the exhaust phases, unburnt fuel, air, and oil are ejected. This is a key difference between the modes.

This hypothesis would also help to explain the slight improvement in the BSFC over some parts of the map as shown in Figures 7(a) and (b), and why this improvement is greater at higher speed and load, where the rotor core would be more pressurized (when enabled). Perhaps the biggest piece of evidence supporting this hypothesis is that the engine generated 1.6% more power when the core was depressurized, implying that such air that was passing through the intake was being burned more beneficially, as would be the case if it was recycled from the core to the intake and hence being given another chance to contribute to the combustion process rather than just being ejected into the exhaust. The decrease in power when the core was pressurized could also be due to an increased power requirement from the fan as the core pressure and density increase. This would be worthy of further investigation, as would establishing the magnitude of the flow rate from the core to the inlet under depressurized conditions; this could be done using a flow meter in this line.

Finally, during the early days of Wankel engine development at General Motors Eberle and Klomp [19] established that improving sealing improved power and BSFC. They estimated changes in these metrics that could be expected per percent reduction of side seal flow. Side seal loss was regarded as less important when compared to flow over the apex seal, being only 1/4 to 1/3 of the total, but it was significant nonetheless. With regard to the work reported here, they point out that with significant flow rates and recirculation to the intake, pumping work changes at part load due to a change in the throttle angle necessary to achieve a certain load. This is undoubtedly occurring here, with further changes depending on whether unburned change is being expelled from the exhaust port or going back to the intake system.

**Rotor and exhaust temperature measurements**

The rotor cooling air temperature was measured at the inlet to the rotor. In the cooling circuit this point is after it flows through the integrated heat exchanger. The results are shown in Figures 14(a) and (b), and show that the core was only a few degrees hotter when operated depressurized, and even then this only really occurred at high load (say, 5°C in 145°C, or 1.2%). This is surprising given the original rationale behind pressurizing the core.
The exhaust temperatures are shown in Figures 15(a) and (b). Generally when running with the core depressurized they are about 25°C hotter. This is presumed to be due to the better combustion efficiency and the hypothesized recirculation of previously-ejected HC emissions seen before, plus the reduction in cooler gases from the core being ejected to the exhaust during the latter stages of expansion and overlap periods.

Steady-state operation: observations on performance when running with and without the core pressurized

From the foregoing the engine appears to operate more beneficially with the core being allowed to breathe directly to the intake and not being pressurized. In particular brake power output, BSFC, HC emissions, and combustion efficiency are all better. A potential mechanism for this has been put forward, and this ties in with the lower oxygen concentration in the exhaust as well as the higher exhaust temperatures.

In relation to the HC emissions, it is especially interesting to note that since the vent line was shared with part of the oil return to the tank for part of its run, there may have been some oil from the core being carried over to the intake and burnt. Unfortunately the magnitude of this cannot be quantified but what can be said is that nonetheless the HC emissions when running depressurized were still significantly better than when the engine ran pressurized. It is therefore recommended that further work be undertaken with the vent taken from an area where oil carryover is less likely, and some form of separator be included as is normal practice in 4-stroke engines.

It should be stressed here that the integrated fan has not been fully optimized for depressurized operation; it may well benefit further from this. Even when depressurized the system is theoretically better than a typical OCR configuration because the bulk of the air travelling into the combustion chambers has not heated by the rotor, as was the case in the Norton ACR motorcycle engine [9]; instead the integrated heat exchanger still takes heat rejected from the rotor into the coolant. Nonetheless, for the configuration of the engine at the moment, there would appear to be some potential optimization with respect to whether the core is pressurized or not, and to what degree.

Warm Stop-Start tests

In hybrid applications, the engine can go through the stop-start process frequently. As previously suggested, the shutdown and start-up processes might be expected to emit higher levels of HC emissions because of the core depressurizing across the side seals. It was therefore decided to investigate the magnitude of any such emission for the two cases of pressurized core and unpressurized.
core, with the supposition being that the engine could have the core vented to the intake before a shutdown by the EMS.

To do this consecutive stop-start processes from different speed and load points were undertaken to understand the correlation between THC emissions and core pressure. In these tests, the engine was warm throughout, and no attempt was made to bring it to bring it to an idle condition before switching it off; they therefore represent a hard shutdown, this being done to better show the differences between the pressurized and depressurized conditions.

It was shut down from different steady-state conditions and then after a short break of around 10 seconds, was motored back up to the same condition from the static state and the fuel and ignition switched on. The same speed and load range as for the steady-state tests was studied, again with both a pressurized and depressurized core. The different core conditions are compared against each other in the following sections.

For the pressurized condition, core pressure dropped during the shutdown tests but in two steps for higher speeds and loads. In the first step, soon after the ignition was switched off, the core pressure dropped to an intermediate level before dropping further with time. For lower loads or lower speeds core pressure generally dropped to atmospheric value more rapidly.

**Hydrocarbon emissions**

Since CO and NOx showed little response to whether the core was pressurized or not, only HC emissions are reported here, in Figures 16(a) and (b) for shutdown conditions and Figures 17(a) and (b) for restarts. Note that the scales have had to be adjusted in some cases because of the magnitude of the difference between the cases.

As can be seen, with the pressurized core in the high load regions the warm shutdown HC emissions were particularly high, in the range of 30000-60000 ppmC1 compared to 25000-30000 ppmC1 during a warm restart. The core pressure prior to these high-load stop and restart tests were in the range of 3-4.3 bar (gauge) and 1.4-3 bar (gauge) respectively, and are presumed to contribute to these very high HC emissions by the mechanism presented. Plots of the percentage reduction in HC emissions for the depressurized core versus the pressurized one are presented in Figure 18 for the shutdown case and Figure 19 for the restarts.
Generally from these results it can be seen that depressurized operation yields significantly lower emissions, particularly during shutdown (up to 80% at higher loads), in line with the theory that unburned HCs are leaking across the sealing grid and out of the engine for the pressurized case. Whereas HC emission spikes were observed during shutdown with the pressurized core, no such spikes were seen in the depressurized condition. For both cases there are more-beneficial points of operation from which a shutdown could be performed, but generally it can be seen that depressurizing the core would be beneficial just before shutting down. Conversely, depressurizing the core resulted in up to 60% reductions in THC emission during a warm restart; the closer rise in HC emissions observed between the two systems is presumed possibly to be more due to the base calibration and combustion system than the core configuration. Finally with regards to start-stop HC emissions the observation made for the steady-state emissions tests regarding the sharing of part of the vent run with the oil return line is likely to impact these results in the same manner, i.e. with a fully separated and more-optimized vent they could eventually be lower than reported here.

**Conclusions and Recommendations**

Investigations into the implications of depressurizing the core in a Wankel rotary engine employing a novel air-cooled rotor system (SPARCS) were conducted. Engine performance, fuel consumption and emissions were obtained across an operating map limited by the speed limit of the generator that the engine is to be attached to when it functions as a range extender in a REEV. This work has shown that:

- Power, BSFC, HC emissions, and combustion efficiency were all better across the operating map investigated when the core was depressurized, by 1.6% for power and BSFC and 4.4% for combustion efficiency. The hydrocarbon improvements were evident whether the engine was operated at steady-state or under stop-start conditions, when the difference was most marked; depressurizing the core resulted in a maximum reduction in total hydrocarbon emissions during warm shutdown and restart of 80% and 60%, respectively.

- A hypothesis has been developed that links the flow of unburned charge across the side seals into the core during the compression and combustion phases with the subsequent flow of this charge in the opposite direction during the later expansion, induction, and early compression phases. Depending upon which part of the cycle this outward flow from the core happens in the charge can either be recycled or expelled from the exhaust port. This event gives rise to high HC emissions, increased BSFC, and apparent reduction in combustion efficiency.

- In the depressurized case, recycling of the blowby gases to the intake (instead to being expelled to the exhaust system) are believed to result in the improved power, BSFC, and HC emissions seen in that case; the reduction in power when the core is pressurized may also be due to increased power consumption from the fan as the density of the gases there increases.

- The difference in rotor air inlet temperature between the two cases in not significant, amounting to about a 1.2% increase at high load. The measurement point for this was after the heat exchanger and before the rotor, so the increase in temperature after the rotor is unknown.

As a consequence of this work further investigation is recommended:

- The investigations described here should be repeated over the full engine speed range, i.e. not just that limited by the proposed generator. This may show increased efficacy of the cooling system under higher-power conditions.

- In the above situation, optimization of the appropriate level of pressurization for each speed and load condition is recommended, and a depressurization system, mapped in and controlled by the EMS, should be developed.

- In any further testing the take-off point for the depressurization should be moved away from the oil return line, to minimize any chance of cross-talk. This should also be fitted with some form of oil separation system, as is conventionally found in engines with a wet sump.

- The magnitude of the flow from the core to the inlet system when running depressurized should be established to help in understanding the contribution of charge recycling to the change in performance evident under that condition.

- Measurements of the rotor outlet and heat exchanger inlet temperature should also be taken, in order to compute the changes around the system when the core is depressurized.

If the results from the above work lead to a change in operating strategy as far as depressurization of the core is concerned, then insofar as it affects rotor cooling then some durability testing should be conducted as necessary.
References


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### Definitions/Abbreviations

<table>
<thead>
<tr>
<th>Code</th>
<th>Abbreviation</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>ACR</td>
<td>Air-cooled rotor</td>
<td></td>
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<tr>
<td>AFR</td>
<td>Air-fuel ratio</td>
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<tr>
<td>AIE</td>
<td>Advanced Innovative Engineering</td>
<td></td>
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<tr>
<td>B</td>
<td>Housing width</td>
<td></td>
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<tr>
<td>BMEP</td>
<td>Brake mean effective pressure</td>
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<tr>
<td>BSFC</td>
<td>Brake specific fuel consumption</td>
<td></td>
</tr>
<tr>
<td>CO</td>
<td>Carbon monoxide</td>
<td></td>
</tr>
<tr>
<td>e</td>
<td>Generating eccentricity (of the geometry)</td>
<td></td>
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<tr>
<td>EMS</td>
<td>Engine management system</td>
<td></td>
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<tr>
<td>FID</td>
<td>Flame ionization detector</td>
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<td>HC</td>
<td>Hydrocarbon</td>
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<tr>
<td>HCLD</td>
<td>Heated chemiluminescence detector</td>
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<tr>
<td>NDIR</td>
<td>Non-dispersive infrared</td>
<td></td>
</tr>
<tr>
<td>NOx</td>
<td>Oxides of nitrogen</td>
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<tr>
<td>OCR</td>
<td>Oil-cooled rotor</td>
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<tr>
<td>R</td>
<td>Generating radius (of the geometry)</td>
<td></td>
</tr>
<tr>
<td>REX</td>
<td>Range extender (engine)</td>
<td></td>
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<tr>
<td>SHED</td>
<td>Sealed housing evaporative determination</td>
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<tr>
<td>SPARCS</td>
<td>Self-pressurized air-cooled rotor cooling system</td>
<td></td>
</tr>
<tr>
<td>WOT</td>
<td>Wide-open throttle</td>
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