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ANALYSIS OF DIFFERENT UNIFLOW SCAVENGING OPTIONS FOR A MEDIUM-DUTY 2-STROKE ENGINE FOR A U.S. LIGHT-TRUCK APPLICATION

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ABSTRACT

The work presented here seeks to compare different means of providing uniflow scavenging for a 2-stroke engine suitable to power a US light-duty truck. Through the 'end-to-end' nature of the uniflow scavenging process, it can in theory provide improved gas-exchange characteristics for such an engine operating cycle; furthermore, because the exhaust leaves at one end and the fresh charge enters at the other, the full circumference of the cylinder can be used for the ports for each flow and therefore, for a given gas exchange angle-area demand, expansion can theoretically be maximized over more traditional loop-scavenging approaches. This gives a further thermodynamic advantage.

The three different configurations studied which could utilize uniflow scavenging were the opposed piston, the poppet-valve with piston-controlled intake ports and the sleeve valve. These are described and all are compared in terms of indicated fuel consumption for the same cylinder swept volume, compression ratio and exhaust pressure, for the same target indicated mean effective pressure and indicated specific power.

A new methodology for optimization was developed using a one-dimensional engine simulation package which also took into account charging system work. The charging system was assumed to be a combination of supercharger and turbocharger to permit some waste energy recovery.

As a result of this work it was found that the opposed-piston configuration provides the best attributes since it allows maximum expansion and minimum heat transfer. Its advantage over the other two (whose results were very close) was of the order of 8.3% in terms of NSFC (defined as ISFC net of supercharger power). Part of its advantage also stems from its requirement for minimum air supply system work, included in this NSFC value.

Interestingly, it was found that existing experiential guidelines for port angle-area specification for loop-scavenged, piston-ported engines using crankcase compression could also be applied to all of the other scavenging types. This has not been demonstrated before. The optimization process that was subsequently developed allowed port design to be tailored to specific targets, in this case lowest NSFC. The paper therefore presents a fundamental comparison of scavenging systems using a new approach, providing new insights and information which have not been shown before.

INTRODUCTION

While many consider that the future of ground transportation will be electric-propulsion and hydrogen fuel cell only, industry and academia believe that this will take many years to complete, and furthermore that there is considerable potential left in the internal combustion engine (ICE) with regards to improving its efficiency. Indeed, for surface transportation it is entirely possible that long-range haulage will take even longer to adopt these solutions (if ever), and that aviation and shipping may never be able to adopt them. Thus technologies that can improve the efficiency of the ICE are of crucial importance since this will enable easier compliance with future targets and will also ensure their continued relevance for a longer time period.

In the automotive world, the 2-stroke engine has historically long been completely overshadowed by its 4-stroke counterpart. The reasons for this possibly stem from the fact that at the dawn of the automotive age the ICE was in itself new, and engineering knowledge regarding it was likewise in its infancy. The 4-stroke cycle, utilizing either spark-ignition (SI) or compression-ignition (CI) combustion, was much simpler to understand and optimize compared to the 2-stroke engine in which multiple events occur simultaneously. With the growth of the automotive industry being synergistic with the simultaneous development of the ICE, the 4-stroke engine became dominant because it was more easily developed. However, in areas where either power density or efficiency are the primary motivations, the 2-stroke reigns supreme, and it is intriguing to observe that the largest and smallest reciprocating engines operate on the 2-stroke cycle.

Interestingly, it was for reasons of wanting to circumvent the then-existing 4-stroke engine patents of Nikolaus Otto that the first loop-scavenged 2-stroke engine was created by Joseph Day (together with one of his workmen, Frederick Cook) in Bath in 1889-1891 [1]. As engineering understanding of the thermodynamics of engines developed it became apparent that the 2-stroke cycle yielded significant benefits in the form of the minimization of pumping work, through the elimination of the induction stroke. Instead of being done at the same expansion ratio as the combustion part of the cycle, the 2-stroke is free to adopt scavenge pumps with more-optimized pressure ratios and so mitigate this loss.

The result is that as airflow reduces with load in a 2-stroke engine, the combined throttling and pumping losses reduce, whereas the opposite is true for a 4-stroke. This is coupled to other thermodynamic and mechanical advantages. For the same output torque and swept volume, the brake mean effective pressure required of the 2-stroke engine is half that of the 4-stroke. Coupled to this the peak cylinder pressures are lower, and this leads to reduced emissions of oxides of nitrogen (NOx) and thermal losses. The lower pressures also mean higher mechanical efficiency, which is often (but not always, as is the case for the engines discussed here) compounded by the fact that timing drives for gas exchange mechanisms can often be deleted.

The disadvantages of the cycle stem from the scavenging of the burnt gases and their replacement with fresh charge. Because the exhaust phase overlaps with the intake, there is considerable potential for charge short-circuiting. For a simple loop-scavenged engine with external mixture formation (i.e. one using a carburetter and crankcase scavenging) the unburned fuel loss is significant, leading not only to higher exhaust emissions but also to the fuel consumption disadvantage that the 2-stroke engine has traditionally had versus the 4-stroke, despite the theoretical advantages discussed above. The remainder of the fuel consumption disadvantage is largely due to poor combustion, the magnitude of which increases as the load decreases. This in turn is due to worsening scavenging as load is reduced, because the amount of air flowing into the cylinder and available to displace the burnt gas is reduced. Thus the internal residual gas fraction increases and the engine starts to misfire, again worsening fuel consumption and emissions. This phenomenon becomes severe enough that alternate cycles fail to ignite, meaning that the others have a higher proportion of fresh fuel and air in them, then permitting combustion initiation. The engine is then said to be ‘4-stroking’.

Further challenges exist in minimizing oil consumption when ports are used, since this has a detrimental effect on long-term exhaust after treatment (EAT) performance. However, through the use of high-conformability oil control rings the technology exists to reduce this to the level of 4-stroke engines, as reported by Lotus in their research engines [2]. Undoubtedly, further work needs to be done in this area though.

Historically, then, the 2-stroke engine has not had the same level of research expended on it by the automotive industry and as a consequence there are still several fundamental types of scavenging system which could deliver excellent results. The scavenging system effectively defines the major architecture of a 2-stroke engine, and together with the combustion system dictates its performance and fuel consumption[1]. The work presented here seeks to compare three different means of ‘uniflow’ scavenging for a 2-stroke engine suitable to power a US light-duty truck. All of these concepts were compared in terms of indicated fuel consumption for the same cylinder swept volume, and a new methodology for optimization was developed using the GT-Power one-dimensional (1-D) engine simulation package which also took into account charging system work. The charging system was assumed to be a combination of supercharger and turbocharger to permit some waste heat recovery; under some conditions it was found that the supercharger could possibly be deactivated completely.

2-STROKE SCAVENGING SYSTEMS STUDIED

In automotive terms, 2-stroke engines are typically imagined as Day-style piston-ported ones [1]. Because such engines employ either cross- or Schnürle loop-scavenging (the latter being the normal case for more modern designs), which give mechanical simplicity, such engines are usually light and powerful but for the reasons outlined above they are not very fuel efficient or clean. A degree of complication can improve things, and the expansion chamber – where exhaust gas pulsation is used to push short-circuited charge back into the engine just before exhaust port closure (EPC) – does indeed do this. Variable exhaust port height, simultaneously changing both exhaust port opening (EPO) and EPC, can also tailor the gas exchange event, and more complex mechanisms giving asymmetric timing have been shown to help significantly, especially if they are variable as in the Lotus Charge Trapping Valve System (CTVS) [2].

However, apart from the normally-symmetrical timing issue, the juxtaposition of the transfer and exhaust ports on roughly the same plane in the cylinder necessarily also means that their dimensions are limited circumferentially. To increase

[1] Note that historically the 4-stroke engine had several different potential architectures depending on the scavenging mechanisms used (e.g. sleeve valves, rotary valves), but now effectively only has one due to the hegemony of the poppet valve.

the available flow area, the port height has to be increased, resulting in concomitantly earlier timing and therefore meaning that the port duration is long as well. This then exacerbates the port overlap and with it the charge short-circuiting issue, as well as reducing the effective expansion ratio available compared to uniflow scavenging concepts. Finally, the ports are so wide (in terms of the angles subtended by them) that the compression rings have to be pegged, which may or may not be an issue in terms of manufacture and durability because of bore wear.

Logically this means that uniflow scavenging would be expected to be a better option. Through the 'end-to-end' nature of the uniflow scavenging process, it can in theory provide improved gas-exchange characteristics since it helps to utilize the full circumference of the cylinder for the ports for the individual flows. Also, for a given gas exchange angle-area demand, the required angle area for each can be provided over fewer crankshaft degrees and overall expansion can theoretically be maximized over more traditional loop-scavenging approaches. This gives a further thermodynamic advantage.

Mechanically, since the transfer (cold) and exhaust (hot) functions are at opposite ends of the engine, the cooling arrangements provided can be more closely tailored to the requirements, and any thermal distortion of the bore reduced. This has obvious and important ramifications for durability. Finally, any necessary bridges to support the cylinder can be arranged so that the piston rings do not necessarily have to be pegged to stop their rotation as is the case for cross- and loop-scavenged engines. As mentioned, this is potentially important from a bore wear standpoint, too.

In uniflow-scavenged engines, it is conventionally the intake ports that are positioned in the base of the bore wall and which the piston uncovers. However, the exhaust function in a uniflow-scavenged engine can and has been provided by a variety of mechanisms, with three different concepts having been built and operated on a multi-cylinder basis. These are the concepts investigated here. They are:

1. The opposed-piston engine, which has been used for aircraft propulsion as well as power generation and rail traction and is exemplified by the Junkers 205 series [3,4] (refer to Figure 1), the Napier Deltic [5] and latterly by research engines from Achates Power [6,7]
2. The poppet-valve uniflow engine with varying numbers of exhaust valves, typified in production by the Detroit Diesel Series 71 [8], numerous ship engines such as the MAN B&W MC [9] and ME [10], and latterly by research engines proposed by Wang and co-workers [11]
3. The sleeve-valve 2-stroke engine, the unusual arrangement of which was used in the Rolls-Royce Crecy engine, intended for high-speed interceptor aircraft application [12] (refer to Figure 2).

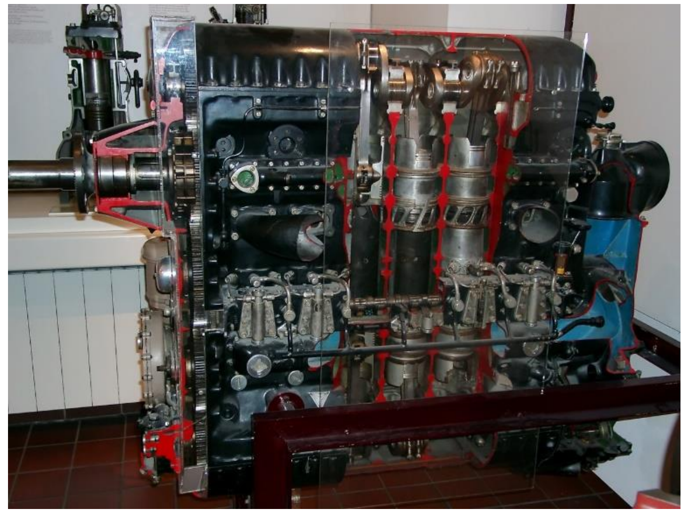


Fig.1: Junkers Jumo 205 opposed-piston engine (illustration taken from [4]. Exhaust pistons and crankshaft are at the top.

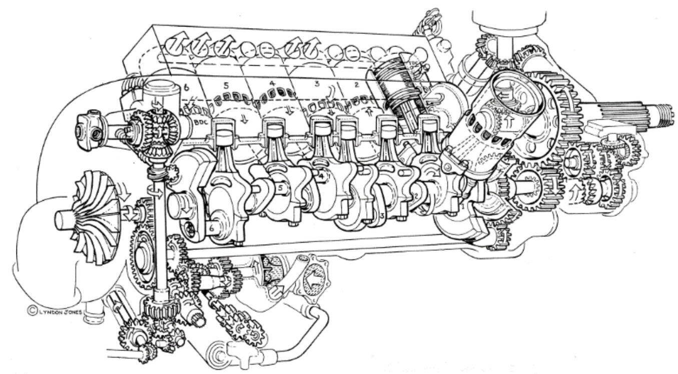
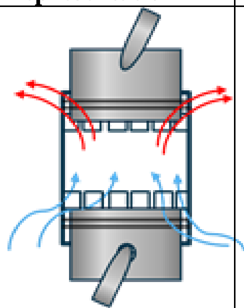
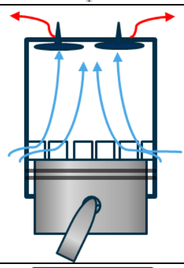
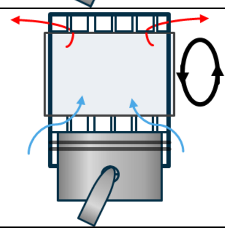


Fig.2: Rolls-Royce Crecy II mechanical schematic. Note the sleeve valves and their drive mechanism on the crankshaft. Reproduced by permission of Rolls-Royce Heritage Trust from [13], copyright the estate of Lyndon Jones.

These scavenging concepts are sketched in Table 1. Note that the Crecy used a Burt-McCollum sleeve valve, which is more usually associated with 4-stroke applications. In a 2-stroke engine it permits the gas exchange to be at opposite ends of the cylinder, whereas this is not the case for 4-stroke applications [14].

Although it may appear unusual to modern eyes, the use of sleeve valves in the mechanical arrangement of 2-stroke was strongly championed by Sir Harry Ricardo before World War II [15], and was subject to extensive research work by D. Napier & Son in addition to Rolls-Royce [15]. While not discussed further here, the Crecy was also a spray-guided spark-ignition engine capable of load control by mixture quality, and together with the fact that versions of it were turbocompounded it represents an interesting engine case study in its own right.

Table 1: Summary of engine concepts investigated in this work.

Name	Description	Visual Representation	Example Engine(s)
OP2S	Opposed piston, piston-controlled intake and exhaust ports		Junkers 205 series etc. [4], Napier Deltic [5], and Achates Power [6]
Port-Poppet	Piston-controlled port intake, cam-driven poppet valve exhaust, uniflow scavenge		Detroit Diesel Type 71 [8] and BUSDIG [11]
Sleeve	Burt-McCollum sleeve valve-controlled intake and exhaust ports		Rolls-Royce Crecy [12]

Some of these mechanisms afford the potential to realize variable timing of the exhaust versus the intake, with the ability to control combustion through homogeneous charge compression ignition-type combustion systems, e.g. the port-poppet engine which in production form has already used electrohydraulic continuously-variable exhaust valve control [10,16] and for which exhaust cam profile switching has been proposed for automotive use [17]. Also, partly because of the improved bore distortion performance mentioned above, all of those engines used as examples employed or employ a wet sump lubrication system, which in theory could permit oil consumption levels approaching those of 4-strokes [2], and, through their proven applications in ships, the durability to eclipse automotive engines.

It should be mentioned that in order to investigate the Crecy-type sleeve, layout drawings had to be created and analysis of these had to be undertaken, using design principles gleaned from the few remaining documents pertaining to this engine [12].

Finally it should be noted that due to the complexity of the undertaking, no detailed modelling of an equivalent 4-stroke engine was conducted. However, other researchers have done this recently, with Warey *et al*. [18] showing that a 2-stroke opposed-piston diesel engine could be expected to have 13-15% lower fuel consumption than its modern poppet-valve 4-stroke counterpart. The fact that the work conducted here also shows the opposed-piston engine to be the best of the options modelled is considered to be some validation of this previous work.

SUMMARY OF THE SIMULATION METHOD

The reported study was an initial one in order to assess the concept of the sleeve valve versus the other, better known systems. Because of its exploratory nature it did not justify a full CFD study at this early stage, and hence GT-Power was used with a common combustion model and boundary conditions for each system. In order to remove conflicting assumptions regarding friction, all of the simulation results quoted here are on an indicated basis, the construction of full engine models and estimation of friction losses being outside of the scope of the current project. In this paper we focus on the simulation results for lowest fuel consumption.

Table 1 summarizes the three engine concepts investigated here. All are configured in a similar way to simplify the comparison, using a common swept volume and assuming 4-stroke wet crankcase designs. All of the engines investigated were configured with the same notional swept volume of 750 cc in order to correspond to a cylinder size suitable for a medium-to-heavy duty engine (these sectors being where it is expected such high efficiency 2-stroke engines will be introduced first). However, their bore and stroke dimensions were chosen to match the scavenging system, i.e. with appropriate levels of under-square geometry suitable for the uniflow scavenging systems; specifically, the OP2S uses the same total stroke:bore ratio of 2.2 as in the Achates engine [6]. Note that once these stroke:bore ratios were chosen, no further individual optimization of this variable was conducted. The resulting specification for each engine is given in Table 2.

Table 2: Summary of basic engine specifications.

Engine Type	OP2S	Port-Poppet	Sleeve
Bore [mm]	75.75	86	86
Stroke [mm]	166.65 (combined)	129.29	129.29
S/B ratio [-]	2.2	1.5	1.5
Swept Volume [cc]	751	751	751
Connecting rod Length [mm]	166.65	258.58	258.58
Compression ratio	15:1	15:1	15:1
Cylinder surface area difference [%]	+4.56	0	0

The models were created using the GT-Power 1-D engine simulation software package. Rather than model specific engines, generic single-cylinder versions of each concept were created. The models consist of a cylinder with plenums either side to represent manifold volumes. Aramco have conducted studies into gasoline compression ignition (GCI) [19] and a common imposed combustion profile model was used in all the variants, using data taken from prior Aramco research work which had been conducted at high load. Is in that prior work fuel preparation is by direct fuel injection, with the quantity being calculated from the desired air-fuel ratio, which is scheduled with engine speed and load. Since the purpose of this investigation was to compare the performance of the three scavenging systems, the adoption of a common combustion model for all cases was considered both justifiable and desirable. At operating point 1 (see later) the AFR was held at 43.7 for all of the configurations, while at operating points 2 and 3 the AFR was increased to 16.2 for all. It was assumed that due to the low engine-out emissions provided by GCI that emissions compliance could be achieved with a suitable EAT system. The intake plenum pressure is closed-loop controlled to achieve the target IMEP, whilst the exhaust plenum pressure is user imposed.

Scavenging air supply was assumed to be by a combined turbocharger and supercharger system, so that there is some waste heat recovery and the friction associated with driving a supercharger is minimized. However, in each case this scavenging system is not explicitly modelled, and instead the conditions at cylinder inlet and exhaust are imposed. Friction is not modelled in this study and therefore the results shown are on an indicated basis rather than brake. Therefore, in order to evaluate fuel consumption and simultaneously account for the energy required to drive the mechanical supercharger, a net specific fuel consumption (NSFC) was calculated as follows:

$$NSFC\ [g/kWh] = \frac{Fuel\ Flow\ [g/h]}{(Indcated\ Power\ [kW] + Supercharger\ Drive\ Power\ [kW])}$$

Supercharger drive power was estimated from an energy balance of the exhaust and intake conditions, with the difference to what could be provided by the turbocharger assumed be made up by the supercharger. Obviously this brings in further assumptions and unknowns, but in order to militate the effect of these, all turbomachinery efficiency was assumed to be fixed at 70% throughout this work.

The intention of this work was to assess the different scavenging systems in as equitable a way as possible and so detailed in-cylinder flow is also excluded. Instead, the scavenging behavior of each design is dictated by a profile which relates in-cylinder to exhaust manifold burned gas fractions, and this was determined from an extensive survey of the literature. From this survey scavenging profiles which were believed to be representative of the scavenging systems chosen were taken and used. These profiles are shown in Figure 3, with that of the opposed piston scavenge taken from work by Mattarelli *et al*. [20] and the port-poppet and sleeve valve arrangements from a study described by Laget *et al* [21].

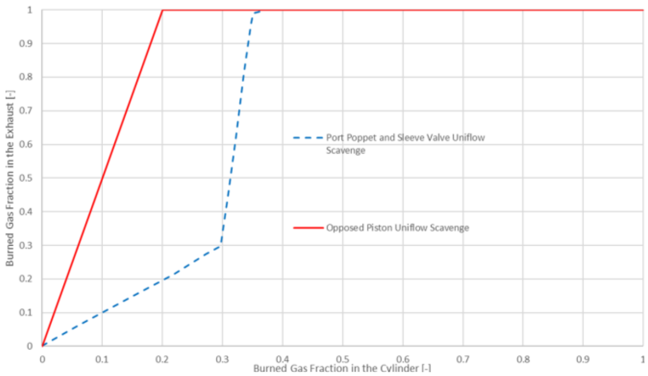


Fig. 3: Scavenge profiles used in the models, taken from literature [20, 21].

Since the study focused on single-cylinder modelling an exhaust system was not explicitly modelled; as a consequence the intake and exhaust pulsations will not be representative of a multi-cylinder engine with a manifold. Instead, simple intake and exhaust manifold (pre-turbine) pressures were imposed, although exhaust pressure sweeps were also performed to verify the trends that were seen at the individual operating points.

Specific time-area calculations were used extensively when analyzing the port and valve optimization results. A significant body of historical work exists where this metric is used to guide the design of 2-stroke engine ports and this was referenced here as a guide. The specific time area is calculated as the integral of port open area with time divided by the total cylinder swept volume, and it provides a measure of port availability for gas flow during the cycle. There are therefore different values which affect the engine performance:

1. Intake specific time area – the intake open area calculated over the time interval from intake port opening (IPO) to intake port closing (IPC)
2. Blowdown specific time area – the exhaust open area calculated over the time interval from exhaust port opening (EPO) to IPO, sometimes referred to as the free exhaust period

An additional time-area calculation was created for insight into the scavenge process:

3. Scavenge specific time area – the minimum of the exhaust and intake port open areas over the interval EPO to exhaust port closing (EPC)

Note that in this paper the term 'port' is used in reference to timing events even when valves (and not pistons) are used for this purpose. Also note that for the opposed-piston engine, the terms top dead centre (TDC) and bottom dead centre (BDC) are referenced from the exhaust piston angular position. The injection timing for this model has been compensated to allow for the fact that as the phase between the pistons changes, the angles of maximum and minimum volume also change, effectively varying the position of TDC and BDC as far as the engine cycle is concerned (i.e. minimum and maximum volume in the cylinder, respectively).

At the start of the study, the guidelines recommended by Naitoh and Nomura [22] at Yamaha were used as a starting point for establishing target angle-areas to lead the design of the port geometry. Although these guidelines were originally set with regards to high-performance loop-scavenged racing motorcycle engines (actually using crankcase compression), it was believed by the authors that they should be just as applicable to any 2-stroke scavenging configuration, and that they could be used to get sufficiently close to the eventual configuration that a numerical optimizer could then be used (see later).

Three operating points were used to evaluate the performance of the different scavenging system designs. These took into account the medium-duty nature of the study and were:

1. 1500 rpm, 3 bar IMEP, 1.2 bar exhaust manifold pressure
2. 1500 rpm, 14 bar IMEP, 2.0 bar exhaust manifold pressure
3. 3000 rpm, 12 bar IMEP, 2.5 bar exhaust manifold pressure

Operating point 1 was intended to be a representative part-load operating point, while points 2 and 3 were notionally peak torque and peak power respectively, in turn representing nominal specific outputs of 225 Nm/l and 60 kW/l. Together with the cylinder capacity, these were considered representative of reasonable performance targets for a medium-duty truck, which is arguably the likeliest place where the 2-stroke cycle might find an application again.

Port timings were determined by numerical optimization of the models at operating point 2 (1500 rpm 14 bar IMEP). NSFC was minimized at this point within the constraints imposed by the geometry and design of each concept. The resultant port timings were then applied at the other operating points. The general constraints on the engine operating envelope for optimization were set to be that:

1. The exhaust port should open before the intake
2. The exhaust port should close before the intake

As discussed above it was assumed that some form of turbocharger/supercharger system would be necessary to supply air to the engine, and the necessary work to drive these systems was calculated and applied so that this requirement was included in the results. To reflect this a further restraint was imposed, to ensure that there would be sufficient exhaust pressure available to drive a turbocharger in such as a system:

3. Exhaust manifold pressures were set to 1.2 bar (Point 1), 2.0 bar (Point 2), and 2.5 bar (Point 3)

ENGINE PORTING ARRANGEMENTS

The general porting arrangements for the opposed-piston engine are well known and are both controlled by their respective pistons. A preliminary study was conducted to investigate the optimum amount of exhaust piston lead over the intake; this is what gives this engine its asymmetric port timing and is also what gives rise to the minimum and maximum chamber volumes not coinciding with the piston dead centre positions as mentioned above. The results of this study, which will be reported in later work, were that the crankshaft phasing should employ 7.5° of exhaust crankshaft lead as a fixed value. While it is theoretically possible to vary the timing between the two crankshafts, this was not assumed to be available in the present study, i.e. once the exhaust lead had been set it was left fixed for the operating points investigated.

In terms of port width, 75% port open area was adopted here for a total of 12 ports, giving 22.5° for the subtended angle of each port. This sizing was itself based on the recommendation by Blair [23] that a maximum of 66% could be used to avoid the use of pegged rings in his book of 1996, it being expected that improvements in materials might permit an increase, but equally that if necessary the rings could be pegged anyway. The topic of port sizes has also recently been covered comprehensively for intake ports on a port-poppet engine by Ma *et al*. [11]. In order to make fair comparisons, this 75% value was adopted for all of configurations, including the sleeve ports for the sleeve-valve engine, while noting in that case that the ports could be larger in the cylinder itself since there the rings run only against the internal diameter of the sleeve.

Once these general parameters had been set for the cylinder ports the main variable left was port height. This simultaneously affects angle-area and the timing of the engine, and with it the compression ratio, expansion ratio and the ratios between the two (see below). Logically one targets maximum expansion and then the lowest value of the ratio of compression to expansion ratios; theoretically if this can reach a value of less than unity then one can create a degree of Miller cycle operation, an operating regime that is not normally associated with the 2-stroke cycle engine.

For the port-poppet engine, the intake port geometry approach is the same as that in the opposed-piston engine. The exhaust process, however, is controlled by cam-driven valves. In this study it was assumed that four valves would be used for this process: since the intention here was to maximize expansion then logically the greatest amount of valve curtain area would be needed.

The cam profiles were calculated for the exhaust valve reciprocating masses and the spring rates were selected from an existing 1-D engine simulation model. However, they had to be modified for use in the 2-stroke engine. Scaling rules were used to ensure that valve accelerations and velocities were not exceeded, i.e. that valve control would be assured for the engine operating range specified. The port angle-area limitations imposed by the valve kinematics are the factor which limits the performance of this type of engine, since a minimum valve event length is then set and this has to be timed to have the minimum impact on the trapped compression and expansion ratios, in turn limiting work extraction as discussed above.

Finally, for the Crecy-style sleeve-valve engine, for which there is only limited literature available, a general engine cylinder scheme had to be drawn using the dimensions of that engine, and then scaled appropriately for the engine being modelled. In Table 1 it can be seen that the exhaust exits at the top of the cylinder like the port-poppet engine. However, in the Crecy itself the exhaust port was of a 360° dimension: the rings did not have to traverse this end of the sleeve and so it was made to drop fully clear of the junk head (i.e. what amounts to a stationary piston) at the top of the cylinder to provide the minimum angular duration for the required time-area [12]. For this application this approach was deemed impractical because of a desire to control crevice volumes and minimize pressure loss, both of which were not considered serious issues in the Crecy. Hence the sleeve was modelled with lands and angles similar to the approach used for the opposed-piston engine, and a set of junk rings assumed to be included to seal the top of the combustion chamber [14]. This would possibly result in a taller engine than the original Crecy's approach, but this was not considered here.

At the other end of the cylinder, the sleeve is also used to control intake angle-area and timing, and it can be timed with a lead or lag angle relative to the piston. These represented a further set of variables that needed to be optimized. Despite the insertion of the sleeve between the cylinder bore and the piston, changes in heat transfer were not considered for this configuration for two reasons: firstly this was primarily a study on gas exchange and secondly there are conflicting stories regarding heat transfer being better (because while it represents an additional barrier, the elliptical nature of the sleeve motion moves the heat around the cylinder) and it was not considered that sufficient knowledge was available to influence a choice in this area one way or another. Further work would be useful to assess this; however, here we consider the heat transfer to be similar to that of the port-poppet arrangement for ease of comparison.

The optimization process used to determine the port timing and geometry took a two-stage approach. Firstly, the engine models were run at the peak torque operating point and sweeps of the intake and exhaust port timings were performed. The data generated by this process was then used to create response surface models of the variables of interest (e.g. ISFC, NSFC) as functions of the intake and exhaust timing events. An offline optimizer was then applied to these surface response models to find the minimum NSFC whilst adhering to the constraints based on geometry and desired operating conditions (e.g. EPO before IPO). The resulting optimal timings were then used to calculate the port/valve geometries required to achieve them.

SIMULATION RESULTS

The results of the optimization are summarized in Table 3 in terms of NSFC and estimated supercharger power requirements, with the associated port profiles shown in Figure 5. The purpose of the investigation was to compare the performance of the three scavenging systems under the same conditions and with the same combustion assumptions. At each operating point and for each system Table 4 shows the peak cylinder pressure and residual gas fraction, Table 5 presents a numerical summary of the compression and expansion ratios, Table 6 presents the port timings, and finally Table 7 the specific time-area values. It should be noted that the data shown here are for comparison relatively and should not be interpreted as absolute values for these engines, since the specific scavenging systems have not been modelled in detail. However, for the purposes of comparison between the concepts this is considered acceptable, since it still permits a general ranking of the different approaches, which was the original intention of the investigation.

Optimizing at the lower speed full load point (peak torque, Point 2) will result in a penalty at higher speeds (peak power, Point 3) due to reduced port time-area, but this is considered acceptable as the focus of the research was on part-load fuel economy, and it was assumed that the charging system could be made to work harder at maximum power.

The results for each concept are discussed in further detail below. It is accepted that the results are based on constrained assumptions, but all systems were subject to the same ones. Future work could included sensitivity studies and in-depth CFD analysis to verify that utilizing the scavenging profiles of Mattarelli *et al.* and Laget *et al.* is valid for the engine geometries investigated [20,21]. Nevertheless, it is believed that the comparisons made are valid.

Table 3: Comparison of results for the three uniflow scavenging concepts studied at each operating point: NSFC and estimated supercharger power requirement.

Engine Type	**NSFC [g/kWh]**			**Estimated Supercharger Power Requirement [kW]**		
Point Number	1	2	3	1	2	3
OP2S	183	189	192	0.244	0.158	1.96
Port-Poppet	194	211	207	0.210	1.27	3.04
Sleeve	197	214	208	0.224	1.27	3.01

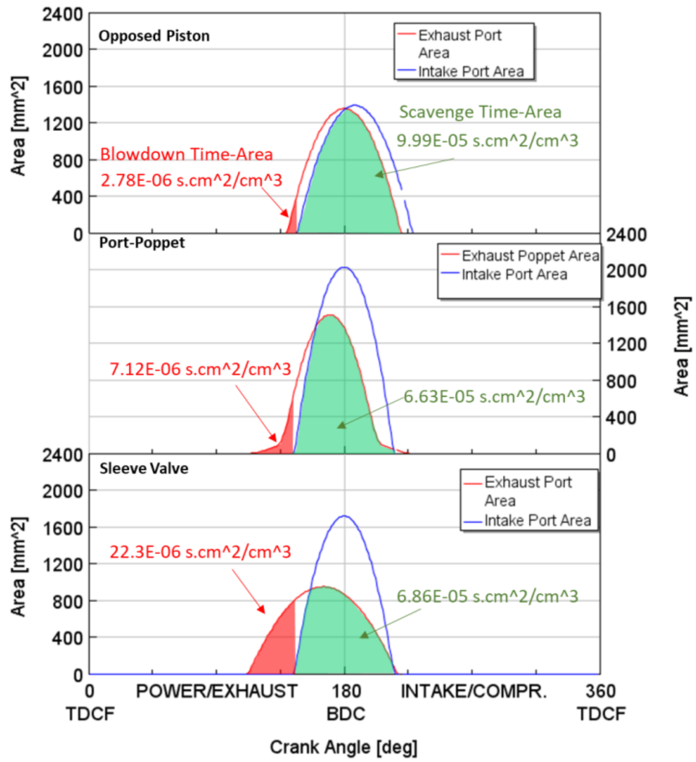


Fig. 4: Port area profiles optimized for lowest NSFC at operating point 2 (peak torque). Top: opposed-piston engine; middle: port-poppet engine; bottom: sleeve valve.

Table 4: Comparison of results for the three uniflow scavenging concepts studied at each operating point: peak cylinder pressure and total burned mass fraction.

Engine Type	**Peak Cylinder Pressure [bar]**			**Total Burned Mass at Combustion Start (EGR+Residual) [%]**		
Point Number	1	2	3	1	2	3
OP2S	52.2	126	131	40.9	24.3	40.5
Port-Poppet	52.2	129	129	39	24.6	40.3
Sleeve	52.1	128	128	38	22.9	39.3

Table 5: Numerical summary of the compression and expansion ratios of the different concepts.

Engine Type	**Effective Compression Ratio**	**Effective Expansion Ratio**	**Ratio of Compression to Expansion Ratios**
	Volume at start of compression / clearance volume	Volume at end of expansion / clearance volume	Compression ratio / expansion ratio
OP2S	13.68	13.67	1.00
Port-Poppet	13.79	11.18	1.23
Sleeve	13.85	11.49	1.21

Table 6: Numerical summary of port timings of the different concepts.

Engine Type	**Optimized Valve / Port Timings [°ATDC]**			
	EPO	EPC	IPO	IPC
OP2S	140	220	147	228
Port-Poppet	115	225	145	215
Sleeve	113	218	145	216

Table 7: Numerical summary of specific time-areas of the different concepts.

Engine Type	**Specific Time Areas (all at 1500 rpm) [s.cm^2/cm^3]**		
	Intake	Blowdown	Scavenge
OP2S	11.8E-05	2.78E-06	9.99E-05
Port-Poppet	12.6E-05	7.12E-06	6.63E-05
Sleeve	11.9E-05	22.3E-06	6.86E-05

Figure 5 presents a breakdown of the power flow from the cylinder at the peak torque operating condition (note that this is not an efficiency breakdown). This is useful for visual comparison of pumping work and heat losses.

OPPOSED-PISTON (OP2S)

The OP2S engine delivers the lowest NSFC over the three selected operating points. This is due to a combination of the lowest ISFC coupled to the lowest supercharger work. The low ISFC is achieved through two principal routes:

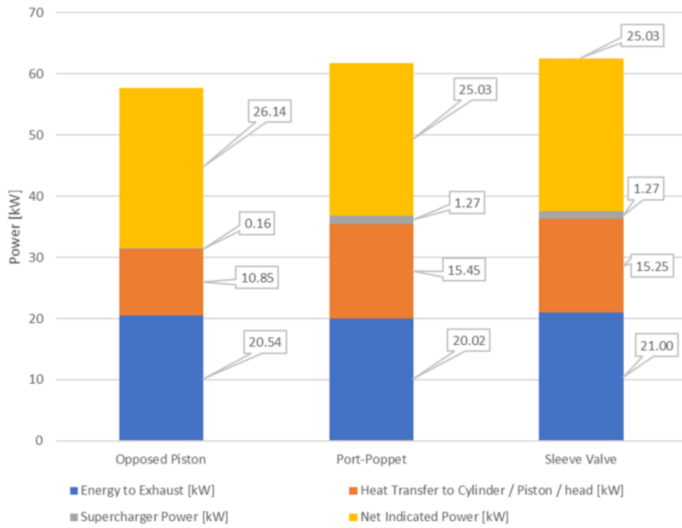


Fig. 5: Breakdown of power flow at the peak torque condition of 14 bar IMEP at 1500 rpm with 2.0 bar Absolute exhaust pressure (note that while the conditions are optimized for lowest NSFC, this is *not* an efficiency breakdown).

1. <u>Maximized expansion work.</u> The OP2S's mechanical arrangement of separate pistons with asymmetric timing each controlling a band of ports permits later exhaust port opening (EPO) which maximizes expansion work. This is due to the fact that having the pistons controlling one aspect of scavenging (exhaust) permits the necessary angle-area to achieve scavenging to be provided in the shortest angular duration, maximizing piston work. Table 6 clearly shows that very late opening of the exhaust ports is possible for the OP2S in comparison to the other two arrangements. This is also shown in Figure 4 and Table 7 by the fact that the OP2S has the lowest blowdown time area of all of the studied systems. Similarly, Figure 4 and Table 7 also show that the OP2S has the lowest intake specific time area. With the intake piston timing established by this, the compression to expansion ratio is approximately 1, compared to values of greater than 1 for the other two concepts (i.e. they operate under expanded). This design therefore has a thermodynamic advantage over the other two.
2. <u>Reduced heat transfer.</u> Although the increased stroke and dual pistons result in a *larger* total surface area compared to the port-poppet and sleeve design (approximately 4.6% greater: see Table 2), the average temperature of the surface is higher (because pistons run hotter than a cylinder head), and thus heat transfer is reduced. The advantage that the OP2S has in this area is clearly shown in Figure 4, where the magnitude of the power loss to heat transfer is the lowest for all three of the operating points investigated.

The above observation on heat transfer is in line with what other researchers have pointed out [7] but attention is drawn to the fact that rather than a simple observation on not having a cylinder head like the other engines, the reason is a summation of surface areas, heat transfer coefficients and surface temperatures. Together these outweigh the increased area that the design carries in this analysis. Further optimization of the stroke:bore ratio may yield further advantages in this area, of course.

The reduced power rejected to coolant might be expected to give an additional benefit at a vehicle level due to a concomitant reduction in radiator area which can benefit aerodynamic drag.

The numerical optimization of the OP2S chose an IPO that follows very shortly after EPO, resulting in the smallest exhaust blowdown period of all the concepts, though the subsequent scavenge period (calculated as scavenge time-area) is the highest. It is thought that this, combined with a slightly more favourable scavenge profile, results in similar levels of trapped residual gases to the other concepts.

The small blowdown period results in a flow of residuals into the intake system at IPO. This is observed to a greater or lesser extent in all the designs due to optimization of port timings for minimum NSFC, however it is most pronounced in the OP2S results due to the very small blowdown period (and least apparent for the sleeve valve timings).

Further retardation in timing of both intake and exhaust events does reduce ISFC but simultaneously incurs higher supercharger work as the incoming air will have to be compressed to a greater pressure, and thus the NSFC increases.

Of all the configurations, the OP2S uses the least amount of supercharger work, as shown in Table 3. Given the simplified nature of the turbocharger-supercharger implementation which relies heavily on estimated efficiencies, it is not possible to say whether or not the supercharger would definitely be de-clutched at any of the engine operating points; however, as shown in Table 3 at peak torque (Point 2) the estimated load is very small and allowing for optimized turbocharger match it is likely that the supercharger could be disengaged.

As mentioned above, the phase offset between the pistons was also investigated. The optimum was found to be with the exhaust leading the intake by 7.5°, and while it is thought this value is specific to the geometry and should not be considered a generic optimum for this type of engine, it is not very different to the 8° value settled upon for one version of the Achates Power engine[2]. This study showed that if the option of variable piston phasing were available, there is a small NSFC benefit to be had from varying the piston phase for the part-load and full power operating points. This will be reported in a later publication. A further and possibly more significant advantage of such a mechanical complication would also be the potential to vary the compression ratio and so control the GCI combustion event more directly. The significant potential of variable compression ratio (VCR) in this context has been demonstrated by Turner *et al*. [2, 26].

[2] Note that these values are significantly different to the value adopted by the Napier Deltic opposed piston engine, which had to utilize 20° of exhaust lead in order solely for its unusual geometry of three crankshafts and three banks of cylinders to work from a mechanical point of view [5]. Also note that this limitation was not shared with the four crankshaft/four bank Junkers 223 and 224 engines [3, 24, 25].

PORT-POPPET

For the port-poppet engine Table 3 shows that the average NSFC is higher than the OP2S. The greatest limitation of this concept comes from the need to keep the poppet valve acceleration forces within mechanical constraints. The poppet valves modelled here are taken from a model of a modern 2.0 litre turbocharged 4-stroke gasoline direct injection engine, with a maximum engine speed of 6500 rpm. Scaling the profile to suit the 2-stroke cycle whilst retaining the acceleration limit results in a minimum exhaust event duration of 110 degrees crank angle for 4 mm of lift. As a result, delaying exhaust valve opening (EVO) for increased expansion work results in EVC occurring later into the compression stroke, and a loss of charge occurs which must then be compensated for with the supercharger system. Hence this concept cannot match the late EVO of the OP2S design and thus has a higher ISFC. If there were a way to improve the poppet valve performance (i.e. to shorten the duration whilst maintaining lift), then it may be possible increase the expansion work and improve ISFC. Such mechanisms may include the use of desmodromic valve operation, as is used in production by Ducati, or an air-valve-spring system. The latter is essentially a motor sport-only system and so is not considered viable here. Other valving systems may offer benefit, but except for the sleeve valve, these are outside the scope of this investigation.

Clearly, when using what amounts to a conventional valve system, there is the scope to employ variable exhaust valve timing afforded by camshaft phasing devices. When investigating this the results show that with nominal timing optimized for peak torque, further retardation of EVO causes a small reduction in NSFC for the part load and full power operating points. At the peak torque condition, retarding the timing reduces residuals, most likely due to increasing the scavenge time-area, although this comes at the cost of increased NSFC due to higher supercharger work.

Table 3 also shows that the power consumption for the charging system is similar for both the port-poppet and the sleeve valve configuration, and that this is generally significantly higher than that for the OP2S.

Finally, from the results it is thought possible that using the port-poppet design in a reverse-uniflow configuration may have some potential benefit, because the ability to phase the intake timing rather than the exhaust might facilitate the application of Miller-cycle operation at certain operating points. However, this was not studied here.

SLEEVE VALVE

As discussed above, the sleeve valve allows the kinematics issues of the port-poppet arrangement to be bypassed. However, in comparison to the other concepts, the interaction of sleeve ports, piston, and cylinder ports makes optimization of valve timing more difficult due to geometric considerations. Varying the phase of the sleeve motion relative to the piston changes the exhaust timing at the top of the cylinder, whereas intake timing is essentially piston-controlled via its interaction with the sleeve ports at the bottom. Indeed, the limiting factor was found to be this interaction of piston motion with sleeve motion near to piston BDC; nevertheless Table 7 shows that the sleeve can afford similar levels of intake time area to the OP2S.

However, despite this complication, Table 6 shows that the optimization process converged on a set of timings very similar to the port-poppet design, and consequently the resulting simulated performance is also very similar. The optimum sleeve phase for the peak torque point was found to be a 15° lead; however, for the part load and peak power conditions a 5° to 10° lag was found to give a slight improvement.

Of all the designs, this concept has the largest blowdown timing, but Table 4 shows that the trapped residual levels are very similar to the other two.

While being aware of the prior discussions, there may be a small benefit in NSFC from reduced heat transfer to the liner due to the sleeve and its movement, but at present this is unknown.

COMPARISON OF THE DIFFERENT SYSTEMS

In order to draw broad conclusions from this work, a simple average of the three operating point fuel economy values (given in Table 3) is used. This gives the ranking shown in Table 8. The supercharger power requirement is also given in this table, but it must be noted that this value reflects only the 'make-up' power (per cylinder) that a supercharger would have to supply as part of a compound charging system within the modelling assumptions made on the charging system discussed earlier.

Table 8: Ranking of the three uniflow scavenging systems investigated.

Ranking	**Engine Type**	**NSFC [g/kWh]**	**Estimated Supercharger Power Requirement [kW, per cylinder]**
1	OP2S	188.0	0.787
2	Port-Poppet	204.0	1.507
2	Sleeve	206.3	1.501

From this it can be seen that the OP2S has a significant advantage over the other two concepts in terms of NSFC – approximately 8.3%. This advantage stems from several things, as mentioned above: reduced heat transfer, increased expansion work, and reduced supercharger power requirement. The latter two points are linked and relate to the ability to use approaching the whole cylinder bore circumference for ports, giving the optimizer the opportunity to use large angle-areas with short duration and yielding the related maximum expansion work. Due to the increased possibility to trade off cylinder pressure and piston work with port timing this also suggests that this type of engine might be well suited to turbocompounding which, while not investigated here, has successfully been applied to 2-stroke engines in the past [5, 12, 27], and proposed for at least one other [28]. Application of this technology should give further-improved fuel economy and is therefore considered worthy of further investigation in connection with the opposed-piston engine.

The other two concepts are very closely matched. Where the port-poppet engine is better for fuel consumption it is worse for supercharger power requirement, and as a consequence of the assumptions made in order to conduct this study it is tempting to rank them equally. However, the sleeve valve may present some further opportunities in its architecture, and these concepts are in the process of development and are the subject of an ongoing patent application. Whether this new technology could be made as efficient as the opposed-piston engine is the subject of some further engine modelling work which will be reported at a later date.

Historically the sleeve-valve 2-stroke engine was further investigated at Rolls-Royce as a major component in a highly-integrated aircraft propulsion system conceived by S.S. Tresilian. This was his so-called 'X-engine' [12] which sought to maximize the architectural possibilities of the sleeve valve in a 16-cylinder 4-row in-line radial engine with the compressor at the front and a compounding turbine at the back. A cross-section of this engine is shown in Figure 6.

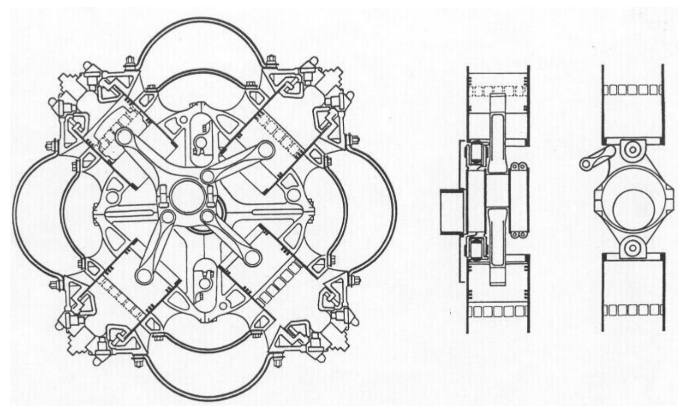


Fig. 6: Cross section of Tresilian's proposed X-engine together with mechanism for driving the sleeves. Sheet metal ducting for intake and exhaust gases is also shown. Reproduced by permission of Rolls-Royce Heritage Trust from [12], copyright Rolls-Royce plc.

As ultimately developed the X-engine was to have had approximately 9.0 litres swept volume; the use of sheet steel to provide the ducting necessary in its highly-compact package would have made the engine light [12]. It never progressed beyond a paper concept, unsuccessfully competing with the then rapidly-improving gas turbine for consideration by engine and airframe manufacturers alike, a fate which it shared with the Napier Nomad [27]. Nevertheless it is interesting to note that the last reciprocating engines realistically to be considered as long-range aircraft powerplants were turbocompounded 2-stroke engines. (Recently Eilts and Friedrichs have again proposed a turbocompound engine to replace turbofans in passenger aircraft, but their proposal is for a 4-stroke engine [29].)

Finally, an important academic finding from this work is that the standard guidelines for deciding the angle-area requirements of conventional crankcase loop-scavenged 2-stroke engines, as given by Yamaha and applied to their racing motorcycle engines [22], have been found to be applicable to the other possible 2-stroke scavenging systems studied here. During the course of the study, these guidelines were used as a starting point for the design of the port geometry, however it also became clear that they were not optimal. For this reason, the approach using numerical optimization of the port/valve timings was adopted. This approach allowed the explicit targeting of minimum NSFC in the design, but the resultant timings showed significant differences in some cases with the guideline values, particularly with respect to the duration of the blowdown process. Further research specifically into the angle-area requirements for modern engines would be useful for the ongoing study of 2-stroke engines.

CONCLUSIONS

Three different uniflow scavenging arrangements – opposed-piston, poppet-valve and sleeve-valve – were studied on a single-cylinder basis using a 1-D engine modelling code. These configurations had previously been applied to multi-cylinder engines in varying production numbers, and so were known to be practical to some degree.

In order to investigate the sleeve valve, layout drawings and analysis of the Crecy-type sleeve had to be undertaken, using design principles gleaned from the few remaining documents pertaining to this engine.

In the work, care was taken to match parameters and specifications where possible. Engine displacement, compression ratio and exhaust back pressure were the primary control variables that were matched, but all were also subject to the same indicated power and torque targets.

The conclusions drawn from this work were:

1. The opposed-piston configuration provides the best attributes since it allows maximum expansion and minimum heat transfer.
2. The poppet-valve uniflow approach was limited by the kinematics of the valve train system.
3. The sleeve-valve uniflow was considered interesting, having the best potential for breathing at higher engine speeds due to the absence of kinematic limitations, although its limiting factor was found to be the interaction of the piston motion with the sleeve motion near to piston bottom dead centre.
4. It was found that existing experiential guidelines for port angle-area specification for loop-scavenged, piston-ported engines using crankcase compression could also be applied to all of the other scavenging types. This has not been demonstrated before. The optimizer also allowed further improvements in NSFC to be made.

The paper therefore presents a fundamental comparison of scavenging systems using a new approach, providing information which has not been shown before.

Furthermore, the work has given rise to a new concept for scavenging 2-stroke engines, which is the subject of further study and a patent application and will be reported on in more detail in later work.

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NOMENCLATURE

1-D	One-dimensional
ATDC	After top dead centre
BDC	Bottom dead centre
CI	Compression ignition
CR	Compression ratio
CTVS	Charge trapping valve system
EAT	Exhaust after treatment
EPC	Exhaust port closing
EPO	Exhaust port closing
EVO	Exhaust valve opening
GCI	Gasoline compression ignition
ICE	Internal combustion engine
IPC	Intake port closing
IPO	Intake port opening
ISFC	Indicted specific fuel consumption
NSFC	Net specific fuel consumption
OP2S	Opposed-piston 2-stroke (engine)
SI	Spark ignition
TDC	Top dead centre
VCR	Variable compression ratio

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