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Funktionsuntersuchung zur Tauglichkeit von 2-Takt Brennverfahren für alternative Kraftstoffe im PKW Hybridverbund

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2 Einleitung

2.1 Aufgabenstellung

Um die Belastung der Umwelt durch den CO_2 Ausstoß zu reduzieren und Mobilität sicherzustellen, ist die Umstellung von rein verbrennungsmotorisch betriebenen Personenkraftwagen auf andere Antriebstechnologien erforderlich. Eine dieser Technologien ist der Elektroantrieb, wobei hier Problemstellen in der Reichweite, den hohen Kosten und hohem Gewicht liegen. Eine Alternative zu rein elektrisch angetriebenen Personenkraftwagen ist die Hybridtechnologie, welche als Brückentechnologie betrachtet wird und aufgrund der steigenden Verkaufszahlen mehr und mehr an Bedeutung gewinnt. Die Hybridtechnologie ist in der Lage, die Vorteile des Elektrofahrzeuges (Minimierung der Schadstoffe in Ballungszentren) mit den Vorteilen der Verbrennungskraftmaschine (hohe Reichweite) zu vereinen. Der Einsatz einer kleineren, leichteren und kostengünstigeren Verbrennungskraftmaschine erhöht die Wettbewerbsfähigkeit des Gesamtprodukts. Mit einer elektrischen Reichweite der Hybridfahrzeuge von 50 km lassen sich 90% der Fahrten bewältigen. Da somit die Verbrennungskraftmaschine im Gesamtantriebsstrang an Gewichtung verliert, ist der Wunsch nach günstigeren, leichteren und kleineren Einheiten naheliegend, um hinsichtlich Kosten, Gewicht und Bauraum konkurrenzfähig zu sein. Als Energieträger für die Verbrennungskraftmaschine haben Biokraftstoffe der 2. Generation und synthetisch hergestellte Kraftstoffe das Potenzial die CO_2 Emissionen der Fahrzeuge weiter zu senken.

Im Forschungsprojekt BEVx-2T wird der Einsatz einer Verbrennungskraftmaschine mit neuartigem Ladungswechsel- und Verbrennungskonzept hinsichtlich Eignung für Hybridanwendungen untersucht. Insbesondere die Kombination mit alternativen Kraftstoffen und die daraus resultierenden Fragestellungen bezüglich Emissionen, Kraftstoffverbrauch sowie Verschleiß steht dabei im Vordergrund. Der Verbrennungsmotor ist ein längsgespülter 2-Taktmotor mit Hochdruckdirekteinspritzung, welcher die Vorteile des 2-Taktmotors hinsichtlich Gewichts, Kosten und Platzbedarf und das Emissions- und Ölverbrauchsniveau eines 4-Takt-Ottomotors nach Stand der Technik in sich vereint. Es wird erwartet, dass sich das 2-Takt Brennverfahren mit Hochdruckdirekteinspritzung günstig mit der Verwendung von Biokraftstoffen, wie Ethanol und 2-Butanol, kombinieren lässt und sich so erhebliche Reduktionspotenziale von Partikel- und NOx Emissionen bieten.

Neben den Untersuchungen zum Brennverfahren mit Bio-Kraftstoffen sind Untersuchungen zum Ölverbrauch und dem Verschleiß von Zylinder/Kolben in diesem Projekt geplant. Die Untersuchungen werden an einem 1-Zylinder Forschungsmotor durchgeführt, wobei bezüglich Verschleißes nur die Zylinder Kolbenpaarung untersucht wird. Die Forschungsarbeiten befinden sich auf TRL 3.

2.2 Schwerpunkte des Projektes

Der Schwerpunkt des Projektes ist nach theoretischen Voruntersuchungen die Erstellung eines 1-Zylinder Forschungsmotors und damit der Funktionsnachweis für das Zweitaktbrennverfahren mit Bio-Kraftstoffen. Damit soll bewertet werden ob durch die Ergebnisse eine Weiterverfolgung des Projektes gerechtfertigt ist und im nächsten Schritt ein Funktionsprototyp – Vollmotor entwickelt wird.

Ziel- und damit zu untersuchende Parameter sind Abgasemissionen entsprechend den gesetzlichen Richtlinien für PKW, CO_2 Emissionen (und damit Kraftstoffverbrauch) sowie eine qualitative Beurteilung des Verschleißverhaltens der Zylinder- Kolben Paarung mit den Kolbenringen. Ein weiterer Schwerpunkt ist die Beurteilung der Eignung einer schlitzgesteuerten 2-Takt Zylinder-Kolben Paarung für üblichen PKW-Fertigungsanlagen und deren Anforderungen.

2.3 Einordnung in das Programm

Das Forschungsprojekt ist ein kooperatives F&E Projekt in der Sparte 4. Verkehrs- und Mobilitätssysteme / 4.1 Wechselseitige Optimierung der Verbrennungskraftmaschine (VKM) unter Verwendung alternativer Kraftstoffe.

2.4 Verwendete Methoden

Für die theoretischen Voruntersuchungen und Konzeptauslegungen werden computergestützte Konstruktions- und Simulationsmethoden wie CATIA, 3-D Strömungssimulationen und Motorprozessberechnungen eingesetzt. Für die experimentellen Untersuchungen wird ein 1-Zylinder Forschungsmotor aufgebaut und am gefeuerten Motorprüfstand betrieben. Dazu werden verschiedenen Varianten des Aggregates samt Vergleichskonzept im Prototypenbau gefertigt und untersucht.

2.5 Aufbau der Arbeit

Als Grundlage für die Arbeit dienten Voruntersuchungen zu Packagingkonzepten für Hybridanwendungen sowie Abschätzungen der Eckdaten der 2-Takt Verbrennungskraftmaschine hinsichtlich Leistung, Emission sowie Herstellkosten im Vergleich mit herkömmlichen PKW-Motorkonzepten. Diese Voruntersuchungen wurden im Vorfeld des geförderten Projektes abschlossen.
Zu Beginn des Forschungsprojektes wurden verschiedene Spülkonzepte der 2-Takt Verbrennungskraftmaschine mit 3-D Strömungssimulation untersucht und anschließend 1-Zylindermotoren als Simulationsmodelle erstellt. Mit diesen Modellen wurden in Motorprozessrechnungen erste Auslegungen sowie Konzeptvergleiche durchgeführt. Darauf aufbauend wurde ein optimales Konzept ausgewählt und als Funktionsmodell in 3D-CAD erstellt; sowohl thermodynamisch als auch mechanisch ausgelegt und die Fertigungsunterlagen erstellt. Die Versuchsaggregate umfassten einerseits Varianten des Vorzugskonzeptes und als Vergleich ein konventionelles 2-Takt Motorkonzept.
Die Fertigung der Versuchsaggregate beinhaltete bereits die Einschätzung auf Fertigungstauglichkeit im PKW-Standard Fertigungsprozess sowie Grundsatzuntersuchungen zu Beschichtungen, insbesondere im Zylinder-Kolben-Kolbenringpaket Verbund.
Mit den experimentellen Versuchsaggregaten wurden Grundsatzuntersuchungen zu den einzelnen Baugruppen Grundmotor, Aufladung, Ladungswechselsystem und Gemischbildungssystem sowie anschließend Untersuchungen im gesamten Betriebsbereich durchgeführt. Diese Untersuchungen hatten

zum Ziel limitierte gasförmige Emissionen sowie Partikelemissionen und den CO_2 Ausstoß zu minimieren, den Betriebsbereich hinsichtlich Drehmoments und Drehzahlgrenzen zu maximieren sowie Verschleißerscheinungen an Zylinder – Kolben – Kolbenringen zu erforschen. Die experimentellen Untersuchungen wurden sowohl mit konventionellen, als auch mit alternativen Kraftstoffen durchgeführt.

3 Inhaltliche Darstellung

Die Darstellung der publizierbaren Projektergebnisse wurde aus den wissenschaftlichen Publikationen entnommen und ist daher in englischer Sprache abgefasst. Der Inhalt gliedert sich in drei Kapitel, die jeweils einer Publikation entsprechen, wobei das dritte Kapitel eine noch nicht veröffentlichte (aber eingereichte) Publikation enthält.

3.1.1 Theoretische Untersuchungen zum Spülkonzept „Overview of Different Gas Exchange Concepts for Two-Stroke Engines“ [50]

The concept of a loop scavenged two-stroke engine, controlling the intake and exhaust port by the moving piston, is a proven way to realize a simple and cheap combustion engine. But without any additional control elements for the gas exchange this concept quickly reaches its limits for current emission regulations. In order to fulfil more stringent emission and fuel consumption limits with a two-stroke engine, one of the most important measures is to avoid scavenging losses of fuel and oil. Additionally, it is necessary to follow a lambda = 1 concept for a 3-way exhaust gas after-treatment.
Therefore, using internal mixture preparation systems in combination with different concepts to control the gas exchange process, the two-stroke engine could become a choice for automotive applications, especially as a Range Extender in a Plugin Hybrid Electric Vehicle (PHEV). As various scavenging concepts and several mixture preparation systems can be used for 2-stroke hybrid propulsion, the pros and cons of different concept combinations have to be compared and weighted for an optimum solution.
To get an overview of possible designs for two-stroke engines for automotive applications, this paper presents different concepts for loop- and uniflow-scavenged two-stroke engines with different gas exchange control elements in detail and compares them. Assessment categories for the comparison are, amongst others, the scavenging process, especially the possibility of adjusting the control timing of the intake and exhaust port, the emission and fuel consumption potential, packaging size and so on. Additionally, influencing parameters on the combustion process like charge movement are illustrated. In a final chapter, the system complexity and the thermal and mechanical durability of the different designs are discussed.

Introduction

The two-stroke engine, in spite of strict emission regulations, still comes into use. The attractive advantage offers the double cycle frequency in comparison to a four-stroke engine, allowing the designer to develop a more compact and lighter unit for the same power target. Due to the compactness and the lower weight, the two-stroke engine could gain a foothold in the automobile industry again as a range extender in a PHEV [1, 2, 3].

For the aim of low emissions, it is necessary to combine the two-stroke concept with technologies like a direct injection as reported by other authors [4, 5]. With additional adaption of a conventional 3-way catalyst, the raw pollutant emissions of CO and HC can be dramatically reduced. But for this concept of an exhaust gas after-treatment system, it's necessary to operate with a stoichiometric exhaust mass flow mixture and guarantee a high trapping efficiency to minimize high peaks of oxygen concentrations in the exhaust flow occurring at the end of the scavenging process. Fortunately, the formation of NOx in the combustion process of a two-stroke engine is partly limited due to the presence of exhaust gas in the cylinder (internal EGR) but the higher the scavenging losses, the worse the efficiency of the catalytic reduction of NOx in the after-treatment system.

Different types of the scavenging process are possible. As shown in Figure 1, the best type is the displacement scavenging with no mixture of fresh air and combustion gas but it's only possible in theory. Most often explained variations are the loop- and the uniflow scavenging designs as illustrated in Figure 2.

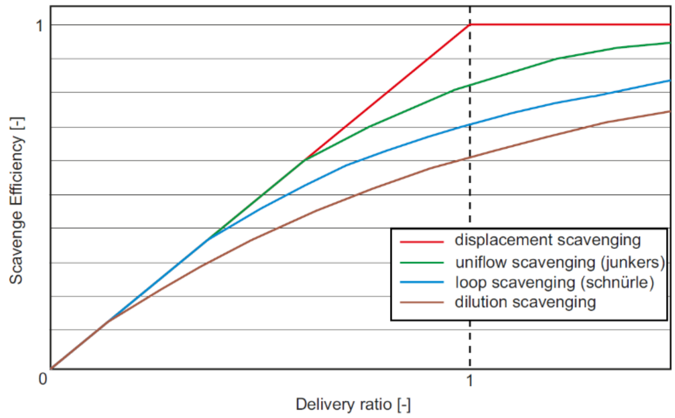

Figure 1: scavenging diagram [6]

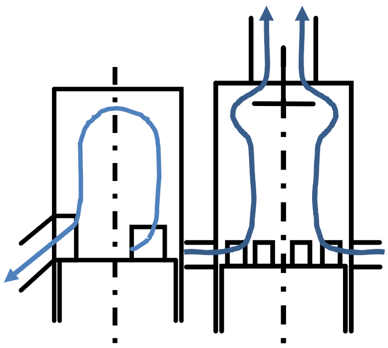

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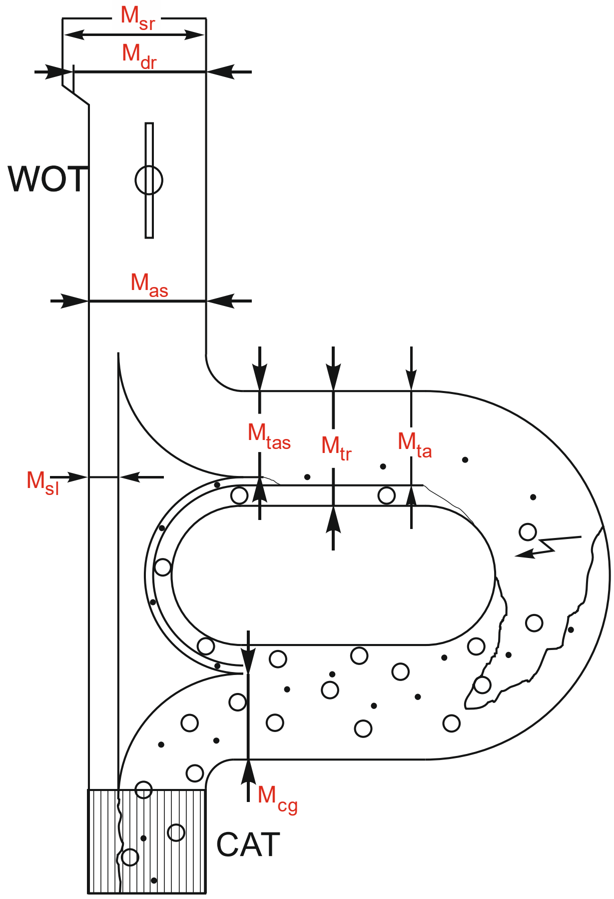
Figure 2: loop scavenged (left), uniflow scavenged (right) [7]

Figure 3 illustrates the mass flow during the scavenging process. Some parts of the combustion products cannot be rinsed during the scavenging process and remain in the cylinder.



$$SR = \frac{M_{as}}{M_{sr}} \quad (1) \qquad DR = \frac{M_{as}}{M_{dr}} \quad (2)$$

$$TE = \frac{M_{tas}}{M_{as}} \quad (3) \qquad SE = \frac{M_{tas}}{M_{tr}} \quad (4)$$

$$SL = \frac{M_{as}}{M_{dr}} \quad (5)$$

Figure 3: mass flow balance of a two-stroke engine [8]

For the quality determination of the scavenging process, there are some ratios introduced (see equation 1-5). For example, the trapping efficiency (TE), which describes how much of the delivered fresh air in the cylinder, will be trapped. It is defined as the ratio between the trapped fresh-charged mass (M_{tas}) and the mass of induced fresh charge (M_{as}). Furthermore, another important ratio is the scavenging efficiency (SE), which describes the purity of the cylinder charge. It is defined as the ratio between the trapped fresh-charged mass (M_{tas}) and the total mass of charge retained at exhaust closure (M_{tr}).

The amount of oxygen in the exhaust gas is descended from excess air from the combustion process and the mass of short-circuiting scavenging (M_{sl}). They are also called scavenging losses. As mentioned before, it is imperative to minimize the scavenging losses to guarantee a high efficiency of the 3-way catalyst. For that reason and because of the open gas exchange process in a two-stroke engine, it's necessary to control the scavenging process.

Therefore, a lot of different two stroke gas exchange concepts were studied. Chiba University is studying on a reverse-uniflow GDI 2-stroke engine (Figure 4) with poppet valve scavenging ports and piston-controlled exhaust ports [9-11].

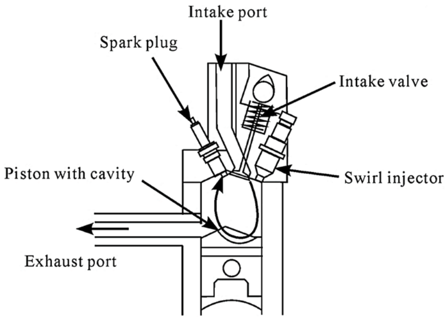


Figure 4: schematic reverse-uniflow engine [9]

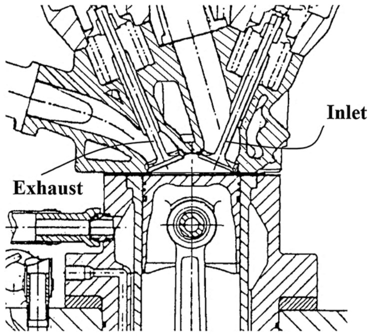
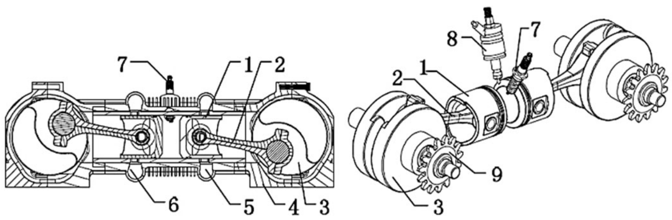


Figure 5: flagship poppet valved two stroke engine [13]

Another proposition of a two-stroke engine concept with 4 poppet valves in the cylinder head for the scavenging process was developed years ago (see Figure 5) [12-16]. Brunel University and Tianjin University is also studying on this concept with different combustion technologies like CAI or external exhaust gas recirculation and water injection [17][18].

A further concept to control the scavenging process is called the opposed piston two-stroke engine, described in [19]. This type features a single cylinder with two pistons to control the ports and two crankshafts (see in Figure 6). The combustion chamber is formed by the pistons head and the cylinder liner. In the middle of the cylinder there is the spark plug and the injector. As can be seen in [20], that this design has got a remarkable thermal efficiency and it is able to generate an efficient uniflow scavenging without poppet valves [21].



1-Piston, 2-Connecting Rod, 3-Crankshaft, 4-Engine-block, 5-Intake Port, 6-Exhaust Port, 7-Spark Plug, 8-Injector, 9-Chain Wheel

Figure 6: OP2S-GDI engine [19]

The focus of this paper is to compare different control elements for the gas exchange process of a loop and uniflow scavenging two-stroke engine to get an overview of the influence and opportunities. As boundary condition, it is assumed that every concept has an external scavenging blower (for example a roots-type blower) and a lubrication system like a conventional oil sump similar to a four-stroke engine. Each concept also uses a high-pressure direct injection fuel system. These constraints are necessary to guarantee an oil- and fuel-free air for the scavenging process.

Loop Scavenging

Design 1.1

The first concept is the simplest design for a loop scavenged two-stroke engine. The exhaust and intake port are controlled only by the moving piston as shown in Figure 7. Usually, the top edge of the exhaust port is above of the intake port to guarantee a pressure reduction in the cylinder to thereby prevent a too high backflow of burnt gas into the intake port. This design represents the reference for the following designs, with regard to the required space (width $W_{1.1}$ and height $H_{1.1}$).

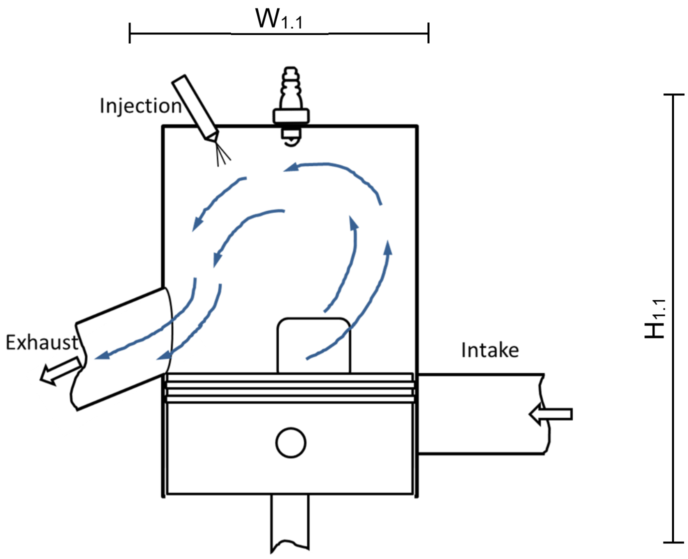


Figure 7: loop scavenged, piston-port controlled concept

For the illustration of the port timing, the exhaust port (red) and the intake port (green) are shown in Figure 8 for a complete cycle. As mentioned before, the exhaust port opens before the intake port to reduce the cylinder pressure. Subsequently, the intake port opens and the scavenging process begins. Therefore, the intake pressure has to be higher than the cylinder pressure and the exhaust pressure (see equation 6).

$$p_{intake} > p_{cylinder} > p_{exhaust} \qquad (6)$$

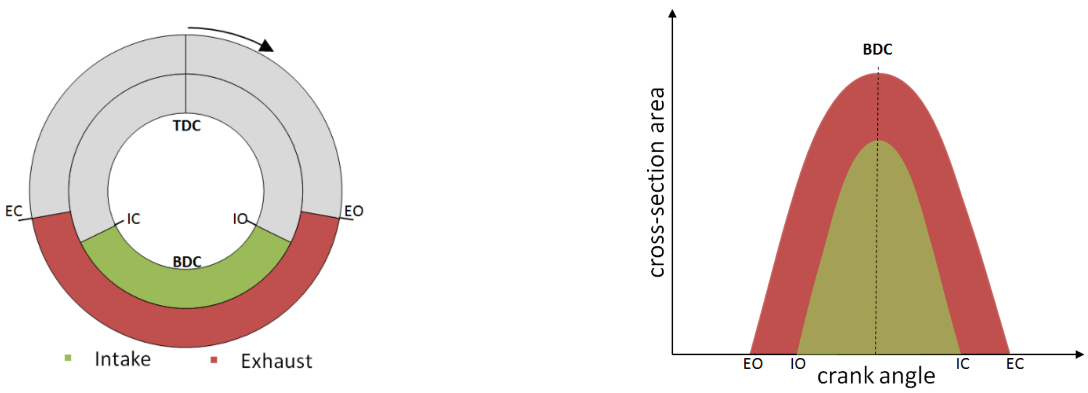


Figure 8: port-timing diagram design 1.1

When the piston is moving back upwards after the BDC, the intake port closes before the exhaust port because of its symmetric port timing. Therefore, a change of the port timing is not possible.
A big advantage of this and the following loop scavenged designs are the completely free configuration of the combustion chamber and the high generation of turbulence. Due to missing control elements for the scavenging process at this design concept, higher scavenging losses are thereby to be expected. With a tuned exhaust system, it is possible to reduce the scavenging losses and generate a higher cylinder charge, but this is not working over the whole speed range. It also should be mentioned, that with an after-treatment system it could be difficult to generate sufficient gas dynamic.

This concept is, from the mechanical aspect, the simplest and cheapest in comparison with the following ones. Due to the position of the exhaust port, there is a one-sided thermal stress for the piston and cylinder. On account of the relatively big cross section of the ports, especially of the exhaust port, it is also to be calculated on higher wear of the piston rings. This problem could be solved by cooled lands in the ports. Therefore, the ports have to be widened to ensure the same port area for the scavenging process. Nonetheless, this solution is limited because of the physical closeness of the intake and exhaust port.

For a compact overview of this concept, a benchmark for the charge cycle, the feasibility for a lambda = 1 concept and the influences for the mechanical system is shown in Table 1. In detail for the charge cycle, there is a subdivision for the possibility of an unsymmetrical port-timing, which means also the potential of supercharging, and the variability of the scavenging process for different operating points. In detail for the next category of the potential for a lambda = 1 concept, the freedom in the creation of the combustion chamber, the viability of charge movement and the scavenging losses, which are important for the after-treatment system, are rated. At least for the mechanical system, the oil tightness and feasibility of the scavenging control element are rated (in particular for this design there is a high grade due to non-existing control elements) and the friction in form of the thermal stress and wear of the piston rings is evaluated. The assessment reaches from very good (+ +) to moderate (o) and very bad (- -).

Table 1: overview of design 1.1

Charge cycle		lambda = 1 concept / after-treatment			Mechanical system			
Unsymmetrical port-timing	Variability	Combustion chamber	Charge motion	Scavenging loss	Oil tightness (control element)	Feasibility (control element)	Thermal stress of piston and cylinder	Wear of rings (oil consumption)
- -	- -	+	+	- -	+ +	+ +	-	-

Design 1.2

The intake and exhaust port at this design is controlled by the moving piston, but to provide an asymmetrical port-timing of the exhaust port, this concept uses a cylindrical valve which rotates in synch to the crankshaft angle, visible in Figure 9. Due to the additional control element, the engine width increases a little, in compare to design 1.1. Some of the effects of this design have already been demonstrated on a small two-stroke engine (50 cm³) with crankcase scavenging and an intake manifold fuel injection developed by the Institute of Internal Combustion Engines and Thermodynamics at Graz University of Technology [22].

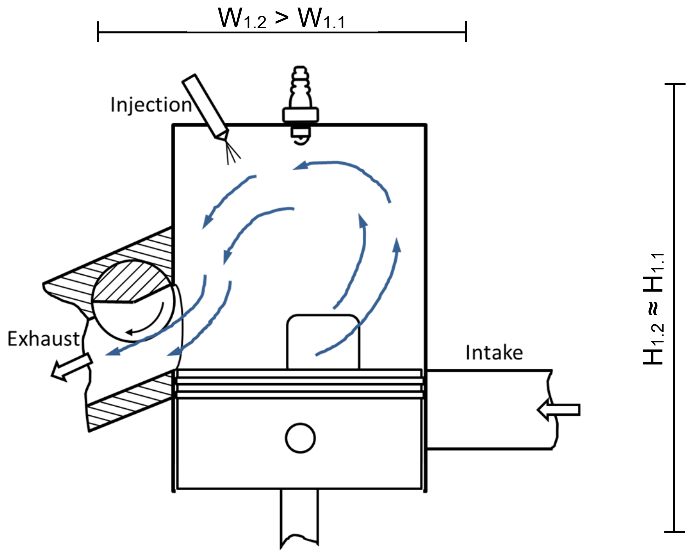


Figure 9: loop scavenged, cylindrical exhaust valve concept

An example of the asymmetrical port-timing implementation with the exhaust valve is shown in Figure 10. The position of IO, EO and IC is unaffected through the cylinder. After BDC, when the piston is moving upward, the position of EC is varied by the rotating valve. Thereby the bright green area between EC and IC illustrates the phase for supercharging. On the investigated prototype in [22], the exhaust pipe closes only a few degrees earlier than before but still after the intake port. A significant reduction of the scavenging losses was reached with this minor modification. The same effect has been demonstrated on other prototypes with this design concept [23, 24]. For a better trapping efficiency in several operating points, it would be necessary to implement a phase adjustment. But because of geometrical reasons an adjustment is only possible in a restricted extension.

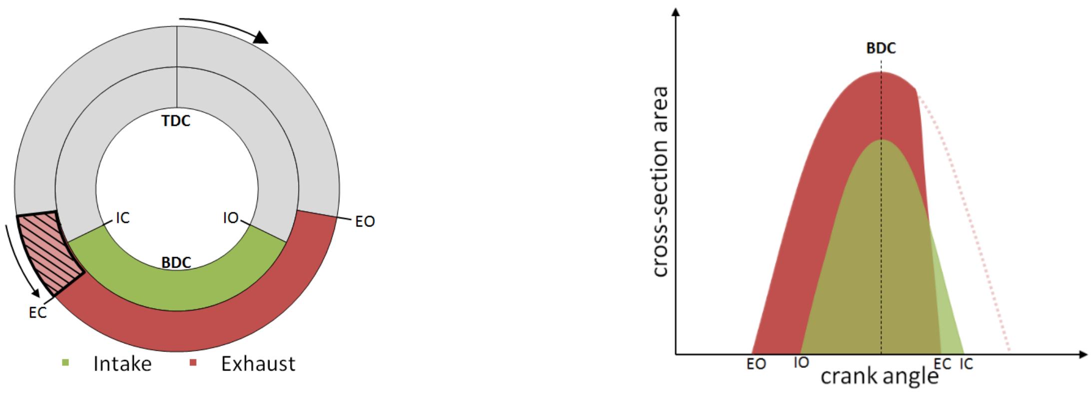


Figure 10: port-timing diagram design 1.2

With closing of the exhaust port earlier than the intake port, an additional potential of supercharging would be possible. But as a complete gas tightness of the exhaust valve could not be guaranteed, the trapping efficiency and potential of supercharging has its limit. The advantage of a free configuration of the combustion chamber and the high generation of turbulence are the same as Design 1.1.
From mechanical aspects, the additional exhaust valve needs a control drive and it has to be supplied with lubrication oil. A separation of the oil cycle could be difficult to realize in this area of hot waste gas. By using additional cooled lands in the exhaust port to reduce the wear of rings, the design of the exhaust valve has to be modified with grooves. An issue could be the mechanical stiffness of a grooved cylindrical exhaust valve in combination with high thermal stress. Also, a thermal expansion of the lands has to be considered, which could have negative impact of the gas tightness and the friction.
As can be seen in Table 2, this design reaches a slightly better rating for the category of the charge cycle and for the scavenging losses. But because of the use of an exhaust valve to control the scavenging process, the oil tightness and feasibility could be a main issue.

Table 2: overview of design 1.2

Charge cycle		**lambda = 1 concept / after-treatment**			**Mechanical system**			
Unsymmetrical port-timing	Variability	Combustion chamber	Charge motion	Scavenging loss	Oil tightness (control element)	Feasibility (control element)	Thermal stress of piston and cylinder	Wear of rings (oil consumption)
-	o	+	+	-	-	-	-	-

Design 1.3

This design uses, as the afore mentioned one, a rotating cylindrical valve to provide an asymmetric port timing, but this time on the cooler side, the intake port. Additionally, at the exhaust port, this concept also uses a slider to optimize the exhaust port-timing for different operating points. Similar to this concept in Figure 11, some of the effects of an intake valve has already been demonstrated on a two-stroke GDI engine prototype (499 cm³) with a piston pump for the scavenging process [2, 25, 26]. As a consequence of a control element in the intake and exhaust port, an increase in the construction width is to be expected.

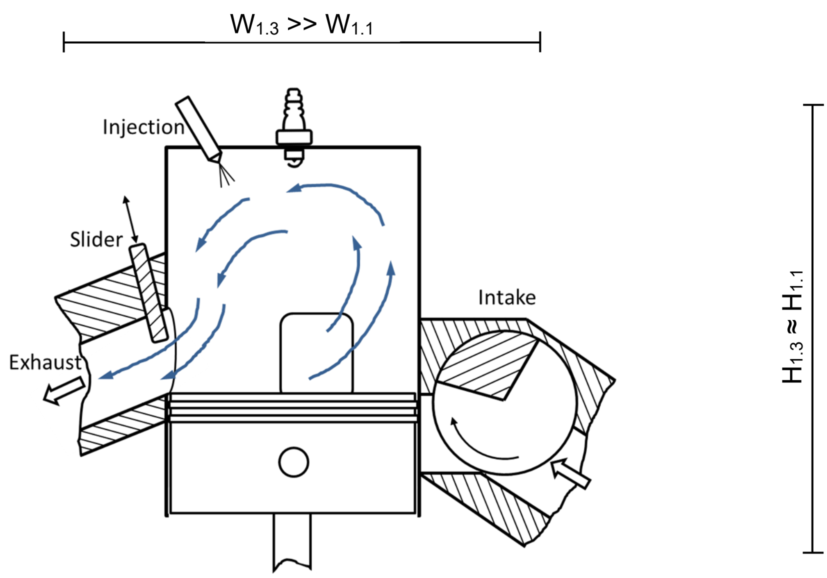


Figure 11: loop scavenged, cylindrical intake valve and exhaust slider concept

An example of the asymmetrical port-timing implementation of the crank angle synch rotary intake valve and the non-cyclical exhaust slider is shown in Figure 12. As can be seen, the slider minimizes the opening duration of the exhaust port by changing symmetrically the position of EO and EC. The intake port-timing is asymmetrically modified by the rotating valve and therefore it is possible to increase in general the height of the intake port. When the piston moves downwards to BDC, the intake port is closed by the intake valve for a bit longer to guarantee a delay of IO. When the piston goes back upwards to TDC, the position of IC is changed. Therefore, the intake port closes after the exhaust port to provide the potential of supercharging (bright green area between EC and IC).

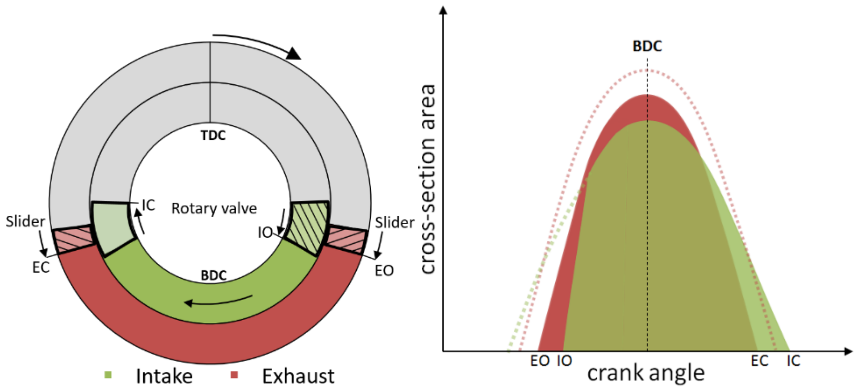


Figure 12: port-timing diagram design 1.3

The adjustment of the slider for different operating points can be realized simply by an actuator. The restrictions of possible adjustments of the cylindrical intake valve are the same as the exhaust valve in design 1.2.

Regarding the trapping efficiency for this concept, no particular improvements in comparison to the design 1.2 are expected. A simulation study [26] on the base of a prototype by the University of Modena with an electric supercharger instead of the piston pump has shown, depending on different operating points, just moderate effects for the trapping efficiency (80-90%).

For this concept, there are two control drives necessary but the tightness of the lubrication cycle is easier to realize because of the simpler control element on the hot exhaust port. By using additional cooled lands in the exhaust port to reduce the wear of rings, the design of the slider has to be modified with grooves. An issue could be the mechanical stiffness of those small fins in combination with high thermal stress in that area. As can be seen in Table 3, the unsymmetrical port-timing and the mechanical aspects of the control element reach a slightly better evaluation.

Table 3: overview of design 1.3

Charge cycle		lambda = 1 concept / after-treatment			Mechanical system			
Unsymmetrical port-timing	Variability	Combustion chamber	Charge motion	Scavenging loss	Oil tightness (control element)	Feasibility (control element)	Thermal stress of piston and cylinder	Wear of rings (oil consumption)
o	o	+	+	-	o	o	-	-

Design 1.4

To provide an asymmetrical port-timing for the intake and exhaust port, this design uses two cylindrical valves and an added slider for an additional variability of the exhaust port, as can be seen in Figure 13. The adjustments of the valves are in synch to the crank angle speed and the slider operates non-cyclically. With regard to the engine space, it should be mentioned that this concept has the widest construction width.

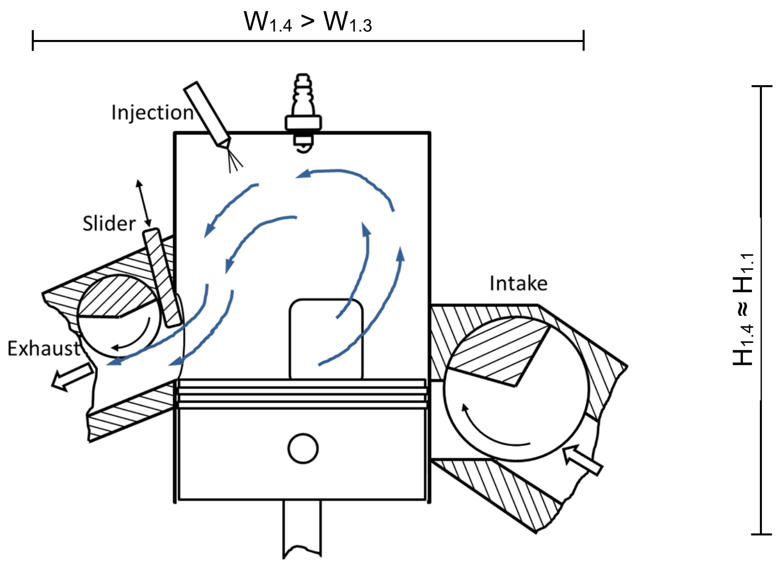


Figure 13: loop scavenged, cylindrical intake/exhaust valves and exhaust slider concept

For illustration, an example of the port timing is shown in Figure 14. The intake port-timing is asymmetrically modified by the rotating valves and the slider minimizes symmetrically the opening duration of the exhaust port as in design 1.3. By the additional rotating valve an asymmetrical port-timing at the exhaust port is implemented to change the EC position.

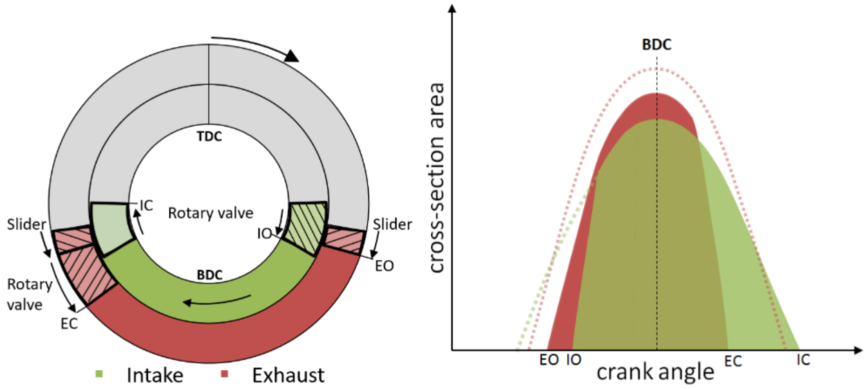


Figure 14: port-timing diagram design 1.4

This design unites the advantages for the scavenging process of the before mentioned designs with a maximum of influence on the port-timing to minimize the scavenging losses and provide a high potential of supercharging. But with regard to a loop scavenged two-stroke concept, the trapping efficiency has its limits and because of the complexity of a slider combined with a valve in the exhaust port and an additional intake valve, it could be hard to ensure a gastight and oil-proof system.
However, Table 4 shows an overview of this design. As mentioned before for the category of the charge cycle, this design reaches the best valuation and a moderate rating for the scavenging losses of all loop scavenged concepts. But this concept shows some weak points with regard to the mechanical system.

Table 4: overview of design 1.4

Charge cycle		**lambda = 1 concept / after-treatment**			**Mechanical system**			
Unsymmetrical port-timing	Variability	Combustion chamber	Charge motion	Scavenging loss	Oil tightness (control element)	Feasibility (control element)	Thermal stress of piston and cylinder	Wear of rings (oil consumption)
+ +	+	+	+	o	- -	-	-	-

Uniflow Scavenging

Design 2.1

A further technique for the scavenging process is the uniflow type. The intake port is at the bottom of the cylinder and controlled by the moving piston. The exhaust port is located at the top of the cylinder and usually controlled by one or more valves. Design 2.1 is a side valve concept, as illustrated in Figure 15, which means that the valves are fitted to the side of the combustion chamber.

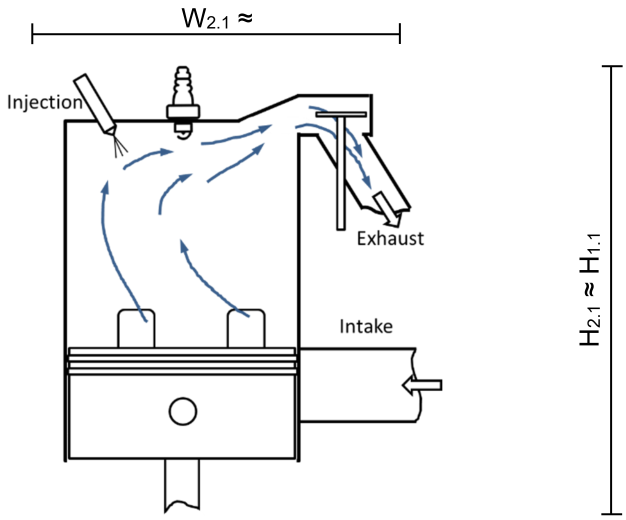


Figure 15: uniflow scavenged, side valve concept

The intake is only controlled by the moving piston. Because of this fact, only a symmetrical port timing of the intake is possible. By the use of a camshaft to control the exhaust port, an asymmetrical exhaust port-timing can be easily realized. As can be seen in Figure 1, a uniflow concept obtains in comparison to a loop concept a higher scavenging efficiency. Therefore, it could be easier to realize a lambda = 1 strategy to use a 3-way catalyst for the after-treatment system.

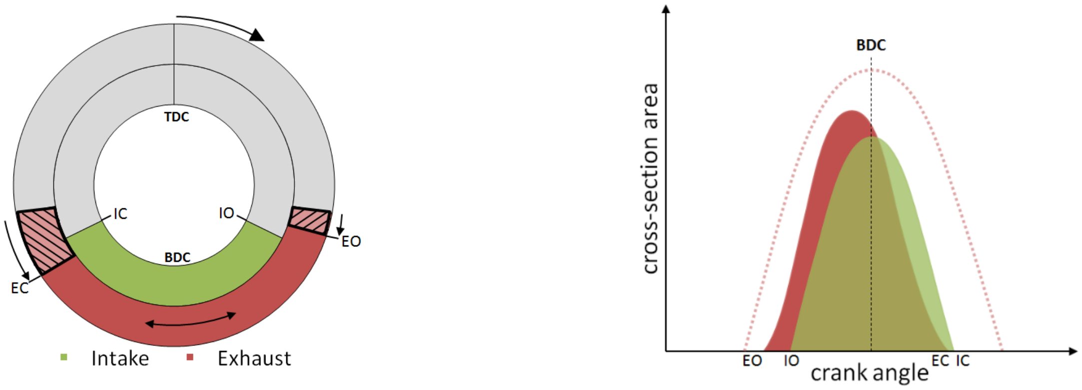


Figure 16: port-timing diagram design 2.1

A concept study similar to design 2.1 by the Institute of Internal Combustion Engines and Thermodynamics at Graz University of Technology [27] has shown the potential of the high trapping efficiency of over 97% at the nominal operation point. Additionally, with a variable valve actuation, which is state of the art, it's also possible to implement a phase adjustment to achieve a high trapping efficiency in different operating points.

A big advantage, with respect to the engine height, originates from the side valve concept because of the valve train, which is located beside the cylinder. This enables the design of a small cylinder height, similar to the loop scavenged concepts, and the width is comparable to design 1.2. Through this shapeless combustion chamber clear disadvantages emerge for the combustion quality and as a consequence thereof higher HC-emissions. Furthermore, it could be problematic to generate a suitable charge motion for a lambda = 1 concept.

As displayed in Table 5, one of the benefits of this design in comparison to the loop scavenged concepts is the mechanical system. Because of the technological state of art of the valve train, there should be low concerns about the feasibility and oil tightness of this gas control system. Only the intake ports could negatively affect the oil consumption because they have to be separated from the lubrication system, nonetheless, a sufficient lubrication of the piston rings must be guaranteed.
The geometrical separation of the intake and exhaust port has its benefits with regards the lower thermal stress of the piston and cylinder. Additionally, more space for the design of the intake ports is available to optimize them for lower wear of the piston rings.

Table 5: overview of design 2.1

Charge cycle		**lambda = 1 concept / after-treatment**		
Unsymmetrical port-timing	Variability	Combustion chamber	Charge motion	Scavenging loss
+	**+**	**- -**	**-**	**+**

Mechanical system			
Oil tightness (control element)	Feasibility (control element)	Thermal stress of piston and cylinder	Wear of rings (oil consumption)
+ +	**+**	o	o

Design 2.2

Another possibility for a uniflow scavenging design is shown in Figure 17 with an overhead valve concept. Thereby, the orientation of the exhaust valve train is similar to conventional 4-stroke engines. Due to this valve orientation, one apparent disadvantage is the engine height, as can be seen in Figure 17. Similar to this design many research studies have been publicized [28-30].

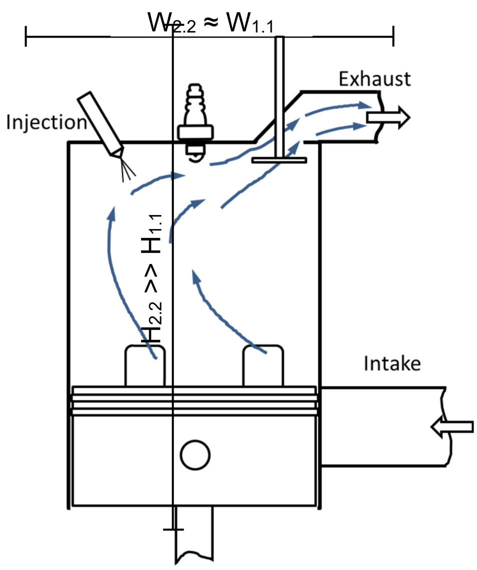


Figure 17: uniflow scavenged, overhead valve concept

The possibilities of the port-timing are the same as in design 2.1 (Figure 16), but with the overhead position of the valve train, it is easier to implement the actuation and a compensation of the valve clearance.
As can be seen in Table 6, the advantage of the overhead valve concept, in comparison to design 2.1, is the higher freedom in design of the combustion chamber to increase the combustion quality and the feasibility of the valve train.

Table 6: overview of design 2.2

Charge cycle		lambda = 1 concept / after-treatment			Mechanical system			
Unsymmetrical port-timing	Variability	Combustion chamber	Charge motion	Scavenging loss	Oil tightness (control element)	Feasibility (control element)	Thermal stress of piston and cylinder	Wear of rings (oil consumption)
+	+	o	-	+	+ +	+ +	o	o

Summary/Conclusions

The paper reviews various designs for loop and uniflow scavenged 2-stroke GDI engines to get an overview of possible concepts for automotive applications such as a REx.

For a loop scavenged type, design 1.3 with a cylindrical intake valve and an exhaust slider would be the best compound between the influence on the charge cycle for a lambda = 1 concept and manageable mechanical control elements. But because of the loop gas exchange process the scavenging and trapping efficiency has its limit. That means, that the use of only a 3-way catalyst for the after-treatment system could be too ineffective for existing emission regulations and an additional device to reduce the NOx-emissions would be necessary. Furthermore, the wear of the piston rings could be an issue for the durability of this concept. A possibility to decrease the friction could be cooled lands in the ports to increase the guiding area of the cylinder. Therefore, the ports have to be widened to ensure the same port area for the scavenging process. Due to the physical closeness of the intake and exhaust port, there is a risk of short-circuit and therewith a decrease of the trapping efficiency may be the result. The challenge would be the design of the mechanical system to guarantee on the one hand the durability of the cylinder, piston and rotating valve and on the other hand low scavenging losses and residual gas for a lambda = 1 concept.

In the case of a uniflow scavenged 2-stroke engine, design 2.2 would be a reasonable solution. The uniflow gas exchange process reaches high scavenging efficiency for a good performance of the 3-way catalyst after-treatment system and therewith probably lower emissions. The valve train provides a sufficient variability for an unsymmetrical port-timing and ensures as well the separation of the lubrication system and also high gas tightness. But it should be mentioned that the camshaft rotates at the same speed as the crankshaft (in a 4-stroke engine it is half of the crankshaft rotational speed), therewith resulting in a higher friction. Due to the physical separation of the intake and exhaust port, lower thermal stress for the piston and cylinder is expected. There is also more space for the design of the intake ports to optimize them for lower wear of the piston rings. An issue could be the charge motion. Generally, uniflow scavenging is used in high efficiency large low speed diesel engines with a swirl concept which can be easily accomplished. The challenge would be a suitable design of the intake ports which generates enough charge motion for a GDI engine with a lambda = 1 concept while retaining a good scavenging efficiency.

3.1.2 Simulation des Ladungswechsels "Simulation Analysis of the Scavenging Process of a Uniflow and Loop Scavenging Concept" [51]

To compare these different two-stroke scavenging concepts, this study focuses on the scavenging and compression phase using 3D-CFD simulation in order to evaluate the scavenging characteristics and the in-cylinder charge motion. The goal of this study is to prepare a basis for discussion of the best configuration, which will be designed, built and tested on the engine test-bench.

Introduction

While being quite common for the propulsion of both two- and four-wheeled vehicles some decades ago, the two-stroke engine lost more and more importance in these fields in the last years. One of the main reasons is that emission regulations are becoming more stringent. In this context, the two-stroke engine suffers from its working principle of processing the exhaust-stroke and the intake-stroke at the same time. Conventional two-stroke engines therefore lose significant amounts of unburnt air-fuel mixture through the exhaust during the scavenging process. Apart from this drawback, two-stroke engines show some quite advantageous characteristics, especially for the use in Range Extender Electric Vehicle and Plugin Hybrid Electric Vehicle applications: The power density is potentially much higher than the one of four-stroke engines, as for one full working cycle only one, instead of two, revolutions is needed; this also reduces torque irregularities. Additionally, the gas exchange needs less components, which also reduces the weight and packaging space, and keeps the mechanical complexity low. In order to reduce the mentioned drawbacks, some novel scavenging concepts have been conceived. A comprehensive study of several strategies [31], based on stoichiometric combustion (lambda = 1), resulted in the selection of the two most promising approaches. The first type of concepts is a loop scavenging type with a cylindrical rotational element in the intake port and an exhaust slider to get a positive influence on the charge cycle (Figure 18). The second type of concepts is a uniflow scavenged two-stroke engine with an overhead exhaust valve train (Figure 27). For both concepts, the aim is to get rid of the in-cylinder residual gas and reach a high trapping efficiency for the aftertreatment system. For verification of the applicability of these concepts, detailed investigations are needed. As prototype manufacturing and the conduction of testbench investigations require a high effort in resources and are very costly, it was decided to carry out 3D-CFD simulations of the concepts first. These calculations are treated and discussed in this paper.

Engine Specification

Both engine concepts compared in this work feature an external scavenging blower, a high-pressure direct injection system and a crankcase like a conventional four-stroke engine. Therefore, it could be guaranteed that in both concepts an oil- and fuel-free air is available for the scavenging process. Furthermore, both engines have the same cylinder displacement and compression ratio.

Table 7: engine data

displaced volume	520 cm³	
stroke	94 mm	
bore	84 mm	
connecting rod	255 mm	
compression ratio	12:1	
cylinder head	Loop	hemispherical combustion chamber, a DI injector and a spark plug
	Uniflow	pent roof with 2/4 exhaust valves, a DI injector and a spark plug

The main difference between these concepts are the control elements for the gas exchange. With the rotary cylindrical element at the intake of the loop type it is possible to realize an asymmetrical modification of the intake port timing (see Figure 18 left). For low load and low speed operation points, a slider is applied in the exhaust port. The uniflow type engine has two/four exhaust valves placed at the cylinder head enabling an asymmetric exhaust port timing by adjusting the valve timing via the camshaft. A variable valve timing enables different exhaust timings according to engine load and speed.

Figure 18: port timing Loop (left) / port timing Uniflow(right)

3D CFD Simulation

A dedicated 3D-CFD model of a loop and uniflow single cylinder engine was built up; the boundary conditions are shown in Table 7. The geometric setup, depending on the model, includes an intake/exhaust plenum, cylinder, piston and exhaust port/valves. At the intake, an external scavenging blower provides a pressure of 1,5 bar at a temperature of 70 °C. Assuming that an aftertreatment system has a significant influence on the exhaust conditions, a counter pressure of 1,2 bar was set for the simulation.

Table 8: simulation boundary condition

Boundary conditions	
intake temperature	70 °C
intake pressure	1,5 bar
exhaust pressure	1,2 bar
intake plenum temperature	75 °C
exhaust plenum	450 °C
piston top temperature	250 °C
cylinder head temperature	250 °C
cylinder liner temperature	150 °C

$$SR = \frac{delivered\ fresh\ charge\ mass}{mass\ of\ trapped\ cylinder\ charge} \quad (1)$$

$$DR = \frac{delivered\ fresh\ charge\ mass}{reference\ mass} \quad (2)$$

$$TE = \frac{mass\ of\ fresh\ charge\ retained\ in\ the\ cylinder}{delivered\ fresh\ charge} \quad (3)$$

$$SE = \frac{mass\ of\ fresh\ charge\ retained\ in\ the\ cylinder}{mass\ of\ trapped\ cylinder\ charge} \quad (4)$$

$$CE = \frac{mass\ of\ fresh\ charge\ retained\ in\ the\ cylinder}{reference\ mass} \quad (5)$$

To characterize the scavenging performance between these two different engine concepts, the five scavenging parameters scavenging ratio (SR), delivery ratio (DR), trapping efficiency (TE), scavenging efficiency (SE) and the charging efficiency (CE) are used in this study.

The reference mass in the above equations is calculated via the displaced volume multiplied by the ambient air density. Another important characteristic is the charge motion, especially the tumble ratio TR and the turbulent kinetic energy TKE, which are important for the combustion process. The following simulations results were calculated for the maximum speed of 4750 rpm at wide open throttle (WOT).

Loop scavenging concept

This design uses a rotating cylindrical element ("intake valve") in three of the eight intake ports (valve-controlled ports) to provide an asymmetric port timing on the intake port (Figure 19 and Figure 20). Six intake ports are without additional control (main intake ports). Additionally, an exhaust slider is used to optimize the exhaust port timing at low speed and load operation points. Similar to this concept, some of the effects have already been demonstrated on a two-stroke GDI engine prototype (499 ccm³) with a piston pump for the scavenging process [32,33,34].

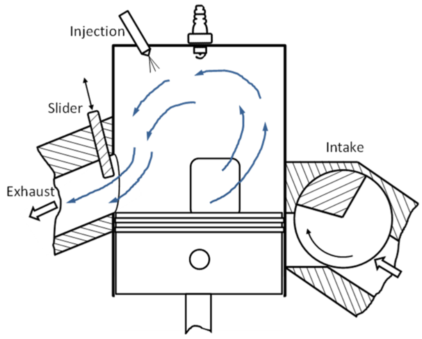


Figure 19: loop scavenged, cylindrical intake valve and exhaust slider concept

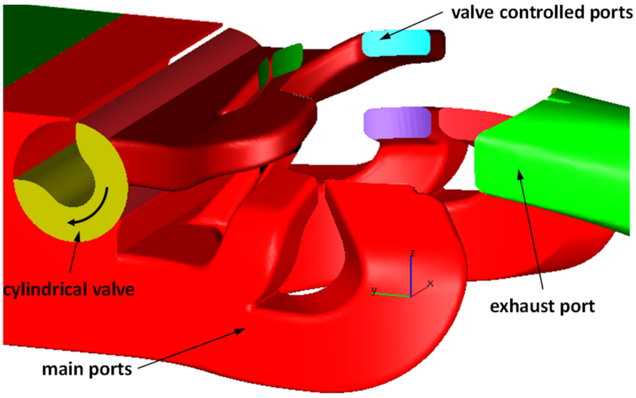


Figure 20: CFD-model of the loop scavenged design

For the illustration of the ports and the cylindrical valve, the CFD-model of this concept is shown in Figure 20 and the port timing is shown in Table 9 and Figure 21. It can be seen that the valve-controlled ports are slightly higher positioned than the exhaust port to get more charge mass into the cylinder. The defined port timing is chosen to accomplish a high air trapping efficiency considering the boundary conditions in Table 8.

Table 9: port timing of the loop scavenged concept

exhaust port	open	108° a. TDC
	close	252° a. TDC
intake port	open	133 ° a. TDC
	close	227° a. TDC
valve controlled port	open	133 ° a. TDC
	close	259 ° a. TDC

Figure 21: port timing diagram

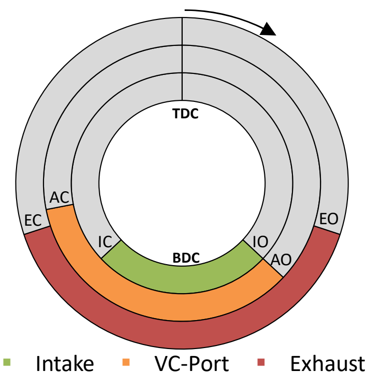


Figure 22 illustrates the rotating intake valve in combination with the moving piston. As can be seen, the valve-controlled ports in the upper right area are closed by the cylindrical valve until the piston approaches the main ports. At BDC the main ports and the auxiliary ports are entirely opened. When the piston is moving back upwards to TDC the exhaust port is closed before the auxiliary port for better charging. To characterize the effects of the design with additional valve-controlled ports in comparison to the conventional loop design, a simulation of both concepts was executed.

intake port opening 133 ° a. TDC | BDC | exhaust port closing 252° a. TDC

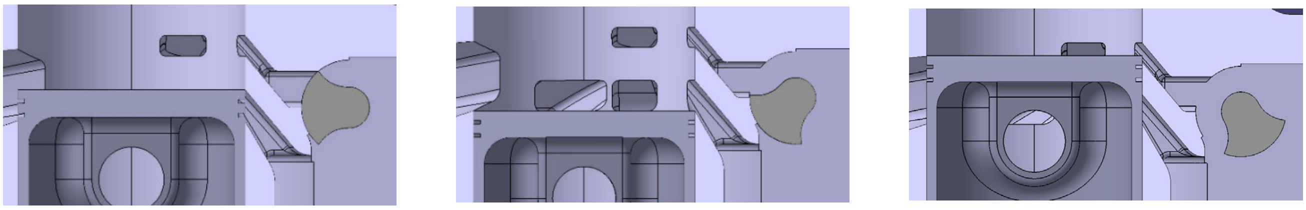


Figure 22: visualization of the valve timing

In the following plot (Figure 23) the intake mass flow of one cycle is demonstrated. It can be seen that at the opening of the auxiliary ports by the moving piston, a fluctuation of the mass flow is generated by the pressure in the cylinder because these ports are closed by the cylindrical valve. When the piston opens the main port, the valve also releases the auxiliary intake ports and a higher amount of fresh air is getting into the cylinder. At 227° a. TDC, when the main port is closed, the impact of the asymmetric port timing can be seen in the mass flow. Fresh air flows through the auxiliary port into the cylinder, having the effect of a better charging.

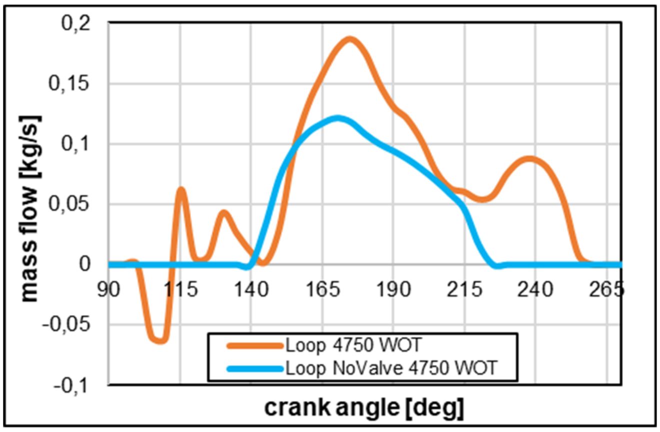


Figure 23: mass flow/crank angle for the loop scavenging concept

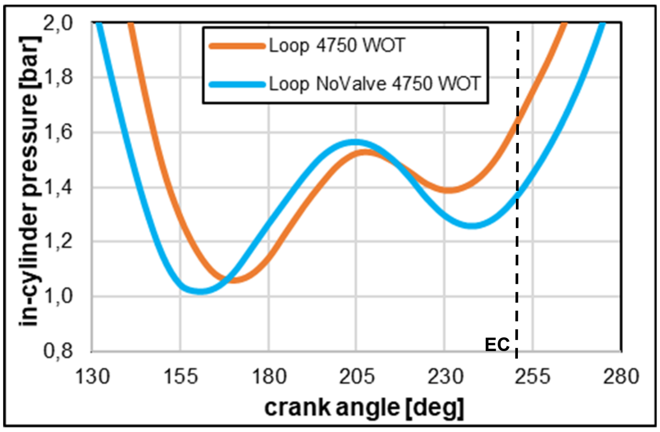


Figure 24: in-cylinder pressure of the loop scavenging concept

The effect of supercharging of the auxiliary valve-controlled ports can also be seen in the in-cylinder pressure in Figure 24. After closing the exhaust port, the in-cylinder pressure of the design with the valve controlled auxiliary ports is about 0,25 bar higher than the design without these ports.

As can be seen in Figure 25, the auxiliary valve-controlled ports also have a positive effect on the charge motion. The tumble ratio and also the turbulent kinetic energy (TKE) are higher at the moment before TDC. That fact could have a positive effect on the combustion process.

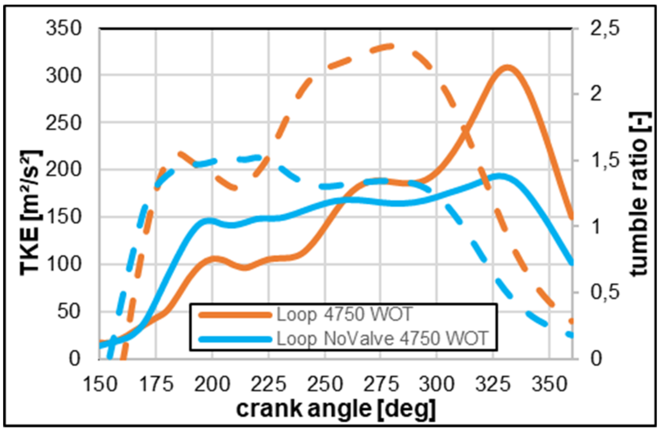


Figure 25: TKE (solid line) / tumble ratio (dashed line) loop scavenging concept

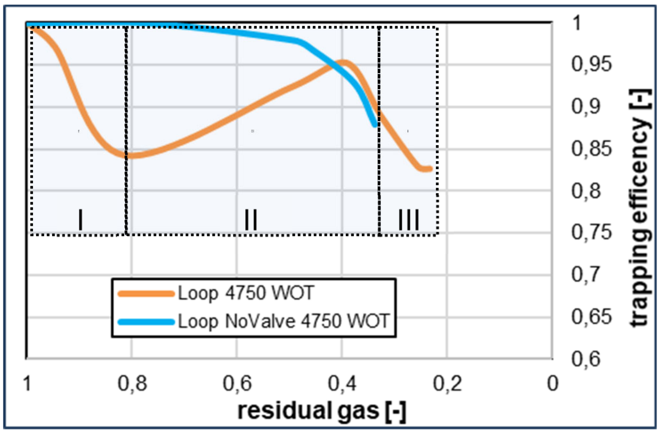


Figure 26: trapping efficiency/residual gas loop scavenging concept

Nevertheless, also some negative effects of the auxiliary ports can be detected. Figure 26 shows the air trapping efficiency over one scavenging cycle, with the in-cylinder residual gas chosen as the control variable. The loop scavenging design without the auxiliary port reaches a high trapping efficiency of ~88%, but the amount of residual gas is about 33%. Because of the pressure fluctuation in the auxiliary port between 99° - 133° a. TDC, a short circuit of the air mass occurs and the trapping efficiency decreases while still showing high amounts of residual gas (I). After 133° a. TDC the main scavenging begins and both types reach nearly the same trapping efficiency (II). At this moment, fresh air streams through the auxiliary port into the cylinder and minimizes the residual gas, but also some air gets lost through the exhaust port and decreases the trapping efficiency (III). With the valve-controlled port configuration, a reduction of ~10% residual gas, but at the cost of a decrease of ~5% trapping efficiency, is possible.

Uniflow scavenging concept

For this concept, the orientation of the exhaust valve train is similar to conventional 4-stroke engines, but the intake is piston controlled at the bottom of the cylinder (Figure 27). Comparable to this design, many research studies have been published [35, 36, 37], but mostly used as a swirl concept. One of the advantages of this concept is the geometrical separation of the intake and exhaust port, additionally, an asymmetrical exhaust port-timing can be easily realized. In this study a 4-valve and a 2-valve concept (Figure 28) each featuring a swirl and tumble intake port design were simulated.

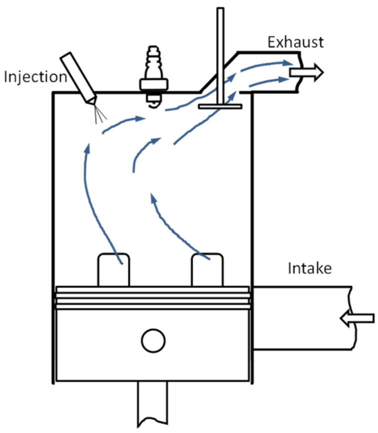


Figure 27: uniflow scavenged, overhead valve concept

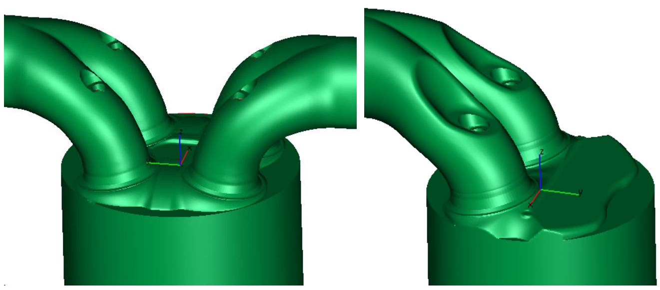


Figure 28: CFD-model 4-valve/ 2-valve concept

The charge motion strategies for the two intake versions have been realized by tangentially oriented intake ports for the swirl type and back-wall oriented intake ports for the tumble type. The back-wall orientation leads to a reflection of the intake streamline forming a reverse tumble. Table 10 and Figure 29 shows the intake port and exhaust valve timings. The intake and exhaust timings for the tumble and swirl concepts are the same, but there were two different valve diameters simulated, to see if there are any influences between the intake port design (swirl / tumble) and the quantity of exhaust valves onto the scavenging quality and characteristic. In Table 11 the different simulation variants are shown.

Table 10: port/valve timing uniflow scavenging concept

exhaust valve diameter		33 mm (2 valve)
		27,5 mm (4 valve)
valve lift		9 mm
exhaust port (0,5 mm valve lift)	open	87° a. TDC
	close	228° a. TDC
intake port	open	136 ° a. TDC
	close	224° a. TDC

Figure 29: port timing diagram

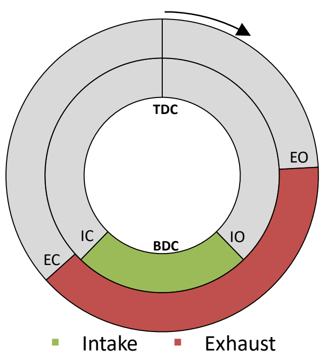


Table 11: simulations variants

variant	intake type	exhaust type	exhaust valve diameter
1	swirl	4-valve	33 mm
2	tumble		
3	tumble	2-valve	27,5 mm
4	swirl		

Figure 30 compares the characteristic parameters of the scavenging process (TE, DR, SE, CE) of the 2- and 4-valve tumble/swirl simulations. The 2-valve tumble concept reaches the best balance between a high trapping efficiency and a low in-cylinder residual gas content of all configurations. Especially the delivery ratio is significantly higher at ~97%, and a scavenging efficiency of ~87% is reached. The residual gas content of the 4-valve cylinder head is about ~18%, and the 2-valve configuration achieves ~14% although the cross-section area of the 4-valve concept is higher than that of the 2-valve configuration.

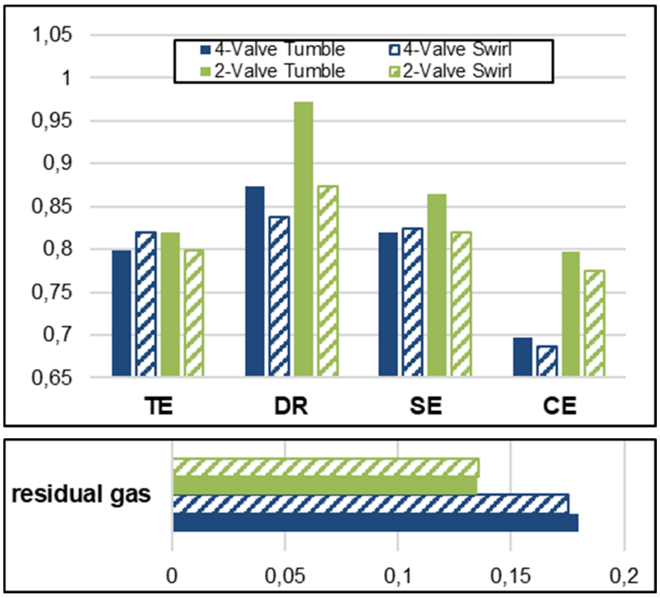


Figure 30: scavenging characteristic 4-, 2-valve configurations

A significant characteristic of the charge motion can be seen in Figure 31. With the swirl intake port configuration, a remarkably high in-cylinder air-motion is generated with the rotation axis being parallel to the cylinder axis. This motion decreases slightly when the piston is moving to TDC, but it persists. In contrary, in the tumble concept the air-motion rotation axis is orthogonal to the cylinder axis, and it is decreasing significantly when the piston is moving towards TDC.

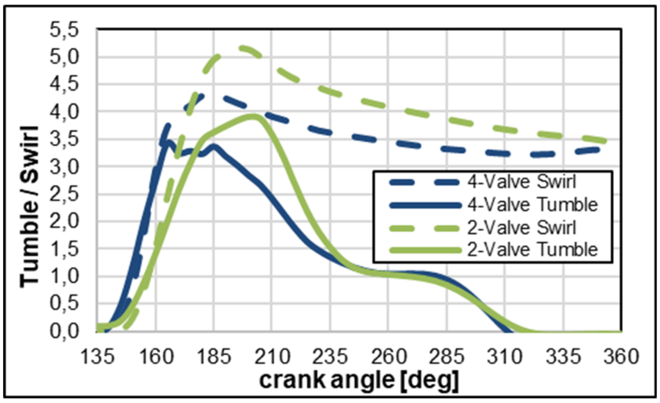


Figure 31: tumble / swirl 4-, 2-valve configurations

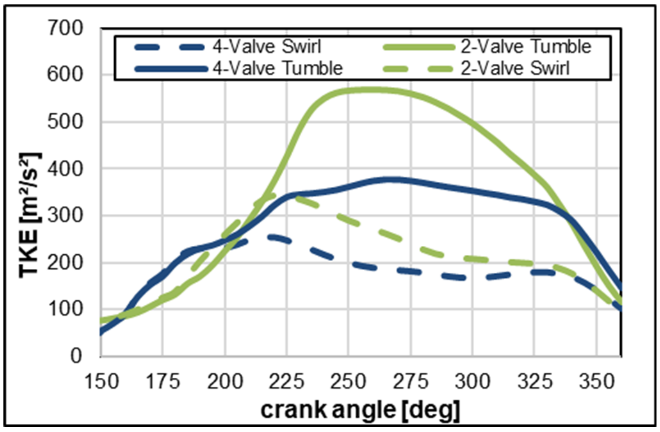


Figure 32: TKE 4-, 2-valve configurations

When the charge motion is breaking up, the energy is transformed into TKE which enables a fast flame front. As the swirl motion stays constant while the tumble is decreasing, a higher turbulent kinetic energy is available for the tumble concept in the region of the ignition timing and combustion, as shown in and Figure 32. During the mixture-formation process, also a higher TKE is generated for the 2-valve swirl and tumble concept in comparison to the 4-valve concept.
All in all, the tumble concept with 2 exhaust valves achieves the best scavenging characteristics and also provides a good charge motion with high TKE during the mixture-formation process and at the time of ignition. Furthermore, considering costs the 4-valve concept has disadvantages, and joining the opposite exhaust ports to one main exhaust port for the connection of the aftertreatment system could be a challenge for the design and thermal management.

Comparison loop / uniflow

The following results show the difference between the loop scavenging concept with the auxiliary cylindrical valve-controlled port and the uniflow scavenging concept with two exhaust valves and a tumble intake design.

Figure 33 shows the characteristic scavenging diagram of a two-stroke engine. The red dotted curve represents a perfect displacement scavenging, which would be the best scavenging type. As can be seen, the curve of the uniflow design is very close to the perfect displacement for a long time, and at the end of the scavenging the efficiency decreases distinctly, which means that there is a loss of fresh charge in the cylinder.

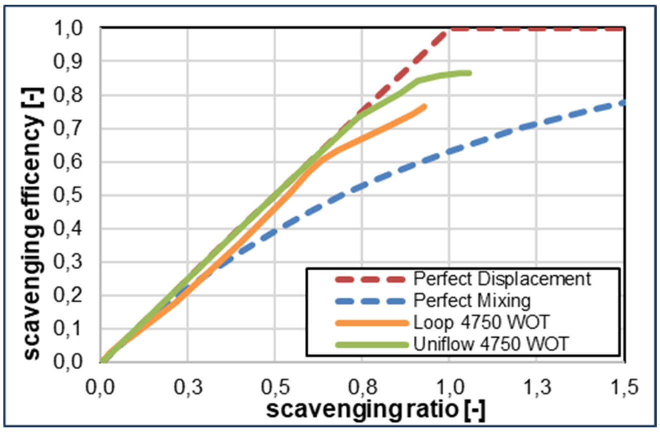


Figure 33: scavenging diagram for loop concept and for 2-valve uniflow tumble concept

The loop design follows, at the beginning, the perfect mixing curve as well, only showing some air loss at the beginning of the scavenging. This is owed to the auxiliary ports during the piston is moving to BDC during the cylindrical vale is closed. In the further progress the curve approximates to the perfect displacement. At the end of the scavenging process, the scavenging efficiency decreases as the result of some significant fresh air losses. All in all, the uniflow design reaches a higher scavenging efficiency, which is an advantage in comparison to the loop design.

A comparison of the charge motion is shown in Figure 34. The uniflow design shows a tumble peak after BDC which is then reduced immediately and leads to a rising TKE during the mixture formation process. The loop design reaches the tumble peak after the closing of the exhaust port. Overall, both designs reach nearly the same TKE at ignition timing.

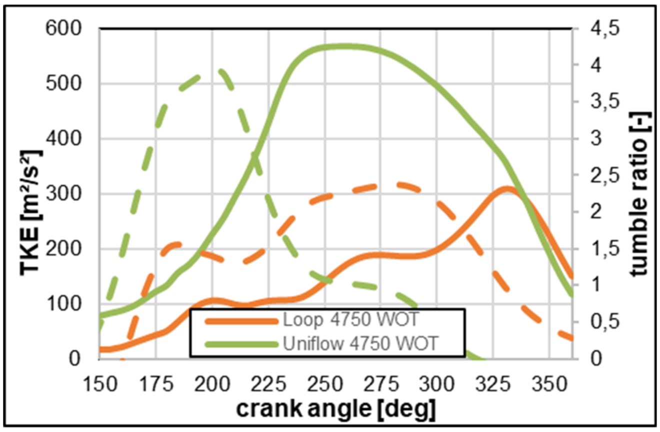


Figure 34: comparison TKE / tumble ratio (dashed) for loop concept and for 2-valve uniflow tumble concept

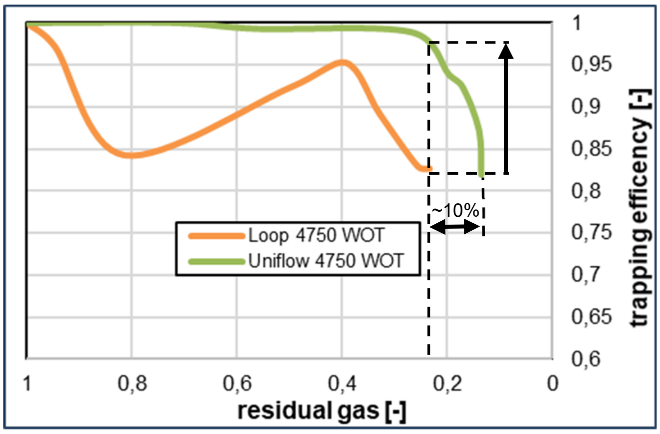


Figure 35: comparison trapping efficiency/residual gas

With the simulated port timings both designs reach nearly the same trapping efficiency, but with the uniflow design a lower residual gas content of ~10 percentage points is reached after the scavenging process. Comparing those two concepts Figure 35 clearly shows, that an optimization towards a higher trapping efficiency could be easier with the uniflow design by an adjustment of the exhaust port timing. Theoretically, a trapping efficiency of ~98% would be achievable for the same in-cylinder residual gas concentration.

Figure 36 to Figure 40 illustrate the scavenging process for the loop and uniflow design concepts, expressed by different colours for the residual and fresh gas. Red areas represent 100% residual gas, whereas blue areas signify pure air.
At the first timestep (100° a. TDC) in Figure 36, the intake ports of the loop design are entirely filled with pure air (blue) and the exhaust port is filled with a mixture of residual gas and air of the previous simulation cycle (green). When the exhaust port opens, the in-cylinder pressure decreases by letting the residual gas go through the exhaust port. Between 100°-130° a. TDC, the high in-cylinder pressure pushes the residual gas into the auxiliary ports and a mixing with pure air takes place.

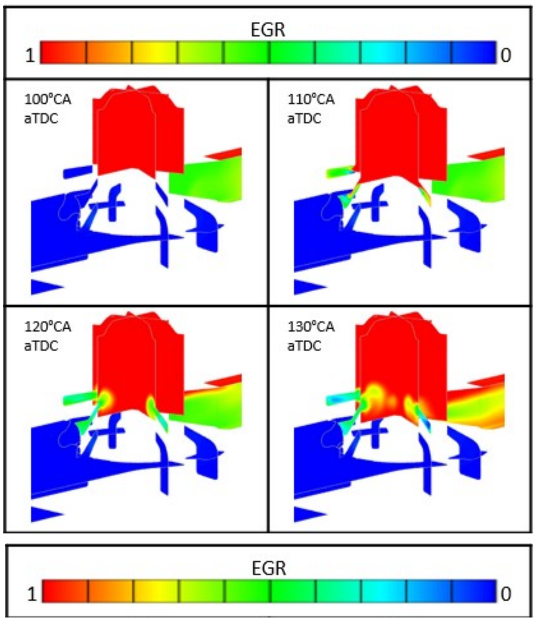


Figure 36: scavenging process loop concept (100-130°CA)

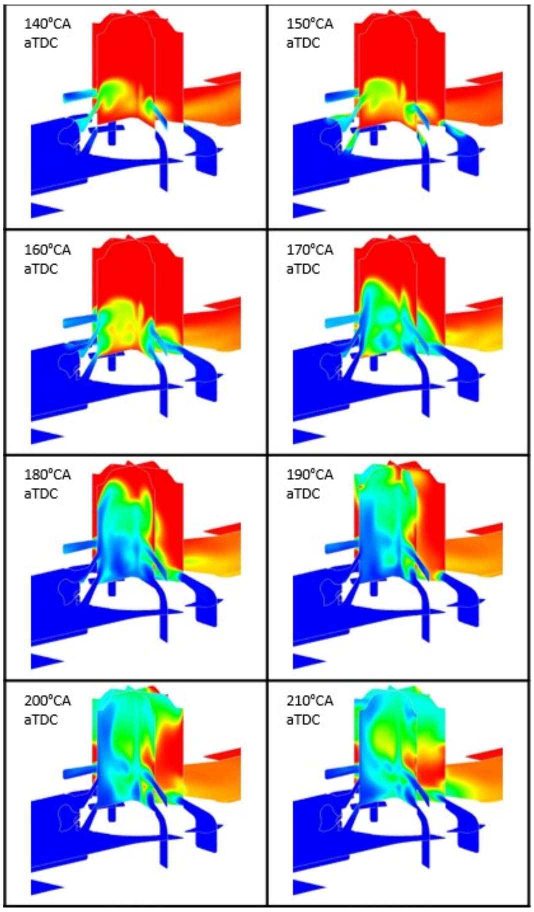


Figure 37: scavenging process loop concept (140-210°CA)

After 130° at Figure 37, the main and valve-controlled intake ports open and the scavenging process starts. Between 160°-170° a. TDC, some residual gas mixed with air is emitted through the exhaust port, which implicates a reduction of the trapping efficiency.

After 230° a. TDC, the main intake port is closed and the additional charging via the auxiliary ports is visible.

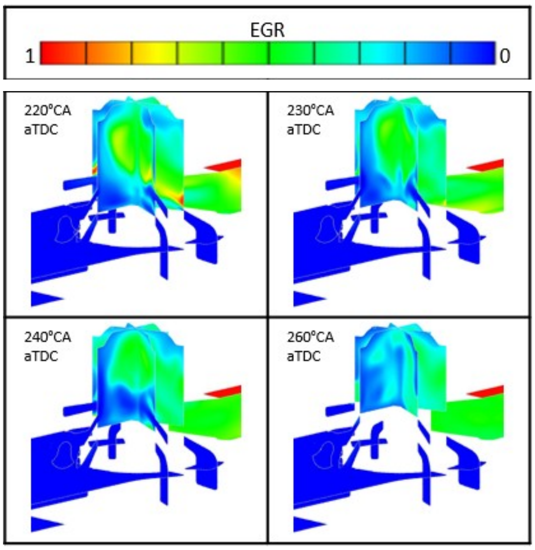


Figure 38: scavenging process loop concept (220-260°CA)

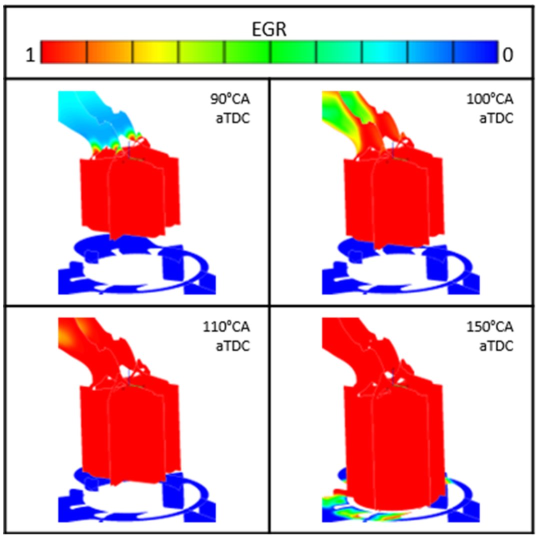


Figure 39: scavenging process uniflow concept (90-150° CA)

At the first timestep (90° a. TDC) in Figure 39, the intake ports at the bottom of the cylinder are also entirely filled with pure air (blue) and the exhaust port at the cylinder head is filled with a mixture of residual gas and air of the previous simulation cycle (light blue). The cylinder is completely filled with residual gas (red). The exhaust valves begin to move, and the residual gas emanates through the exhaust port. As can be seen at 150° a. TDC, there is a pushback of the pure air into the intake ports and therefore the scavenging begins with a delay.
For the next timesteps, the scavenging-gas flows directly towards the exhaust port and displaces the residual gas. From 200°-210° a. TDC, there is a direction reversal of the in-cylinder charge towards the piston. That's also the point of time of the highest tumble ratio (see Figure 34). Figure 40 demonstrates one big advantage of the uniflow design, the geometrical separation of the intake and exhaust port. Therefore, a short-circuit scavenging can be avoided to a predetermined limit.

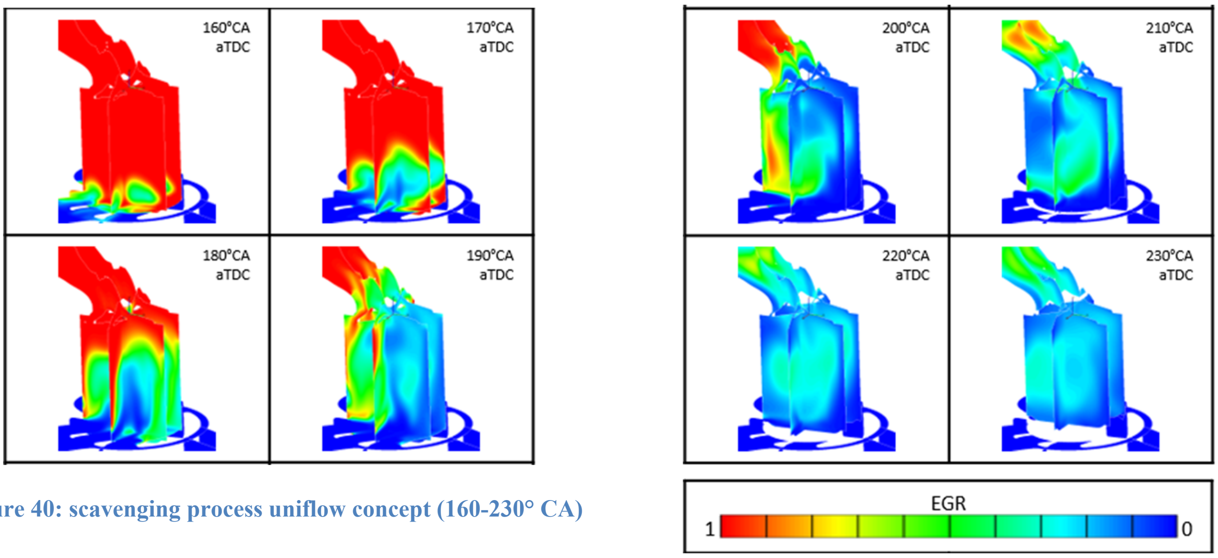


Figure 40: scavenging process uniflow concept (160-230° CA)

Summary/Conclusions

In this study, 3D-CFD simulations were performed to compare different two-stroke concepts during the scavenging and compression phase in order to evaluate the scavenging characteristics and the in-cylinder charge motion. A loop design was simulated with and without an auxiliary intake valve-controlled port to figure out the influence of intake timing control. The uniflow design was simulated with 2 and 4 exhaust valves and with a swirl and tumble intake concept.
All in all, the results of the loop concepts in 3D-CFD simulation clarified that there is a limit on the reachable trapping efficiency because of short-circuit scavenging at the beginning of the scavenging process caused by the auxiliary valve-controlled ports.
The simulation of the uniflow design illustrated that a satisfying scavenging characteristic is possible with a tumble intake concept and an exhaust port with 2 valves. The results show that this concept promises higher potential to achieve an acceptable trapping efficiency with tolerable residual gas content.
The next step should be an optimization of the uniflow tumble concept with 1D-CFD in cooperation with 3D-CFD to increase the trapping efficiency and prepare a prototype for testbench verification.

3.1.3 Experimentelle Untersuchungen "Experimental Analysis of a Uniflow Scavenged Two-Stroke Concept" [52]

To proof the potential of a uniflow scavenged engine in comparison to a classic loop scavenged principle, a reference engine was built with the same geometrical dimensions (bore and stroke). Both engines use the same crankshaft, conrod, piston and crankcase with the same oil, fuel and blower periphery system. As a special requirement for both setups, no resonance exhaust muffler was used for packaging reasons in automotive application. Therefore, only the developed and designed cylinder and cylinder head were different between the scavenging concepts. In Table 12 the initial engine information is displayed. Both concepts have piston-controlled intake ports, which have been optimized for each engine with a tumble oriented scavenging flow. The intake port timing is identical. The uniflow prototype has a valve-controlled exhaust port at the cylinder head with 2 valves and a maximum valve lift of 8 mm and an opening duration of 118°CA. The exhaust of the reference loop engine is conventionally controlled by the moving piston. Both concepts are provided with a high-pressure direct injection system with up to 350 bar pressure.

Table 12: Initial engine data

Concept	Uniflow	Loop
stroke	90 mm	
bore	84 mm	
displaced volume	498 cm^3	
compression ratio	9.81	11.95
effective compression ratio	8.56	8.15
intake	ports (tumble oriented)	
intake open	131°CA a. TDC	
intake close	228°CA a. TDC	
exhaust	2 valves (8 mm lift)	port
exhaust open	90°CA a. TDC	101°CA a. TDC
exhaust close	208°CA a. TDC	259°CA a. TDC
mixture formation	direct injection	

The exhaust of the uniflow scavenged prototype is controlled by a camshaft with two valves at the cylinder head. Therefore, it is possible to implement an asymmetric timing of the exhaust by the general position of the camshaft. As can be seen in the port-timing diagrams in Figure 41 and Figure 42, the exhaust (red bar) of the uniflow prototype closes significantly earlier than the intake port, which is controlled by the moving piston. In order to generate the same effective compression ratio as the reference loop engine concept, the cylinder head of the uniflow engine has been adjusted with head gaskets to reduce the geometric compression ratio. The earliest injection timing (yellow bar) of the loop scavenged engine is limited by the overlap between exhaust port open and start of injection. A big overlap would result in fuel scavenging short cut. There is a small overlap between the injection timing and the opened exhaust at the loop reference engine, but 3D-CFD simulation results verified that this overlap has no significant influence on direct fuel scavenging losses.

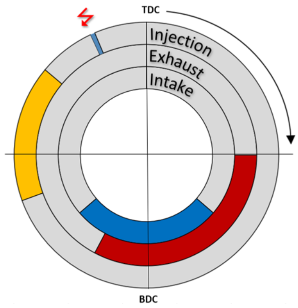


Figure 41: Port timing uniflow

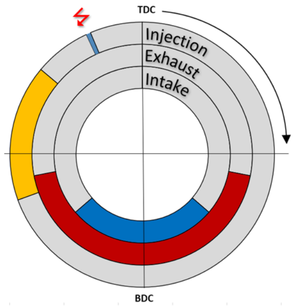


Figure 42: Port timing loop

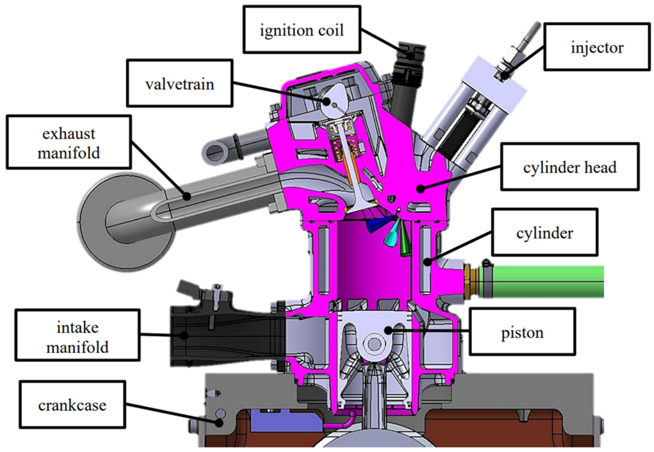


Figure 43: Uniflow research engine

Testbench setup

To investigate the different behaviours of these engine concepts, a development environment with different external peripheries was set up on the testbench (see Figure 44). As mentioned before, both engines are implemented on the same crankcase and crankshaft, which is connected with a clutch to an electric dyno. The engine control is managed by an open ECU and adjustments to the engine parameters like ignition, injection timing / duration and the supply peripheries of fuel, air and water can be conducted via a laptop during testbench operation. The testbench control system records all necessary data like temperatures, pressures etc. and manages the testbench conditions.

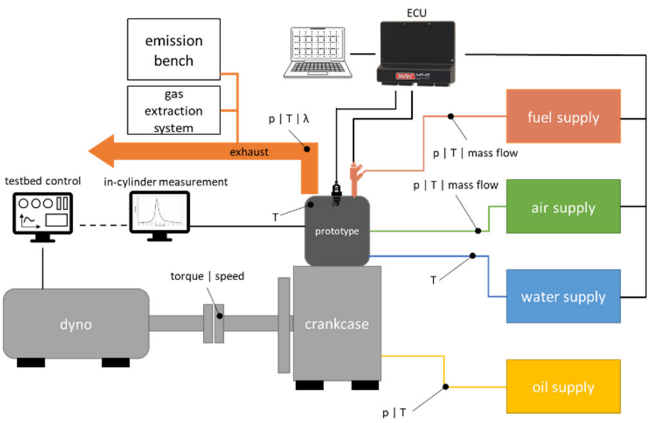


Figure 44: Global testbench setup

Testbench results

Comparison between loop and uniflow concept

To compare both concepts following boundary conditions were set up. First, the injection timing was set to late (110°CA b. TDC) to avoid direct scavenging losses of fuel towards the exhaust, especially for the loop concept because of the symmetric exhaust port timing (see Figure 42). The SOI was still before closing the exhaust port completely, but results of 3D-CFD simulation indicate that no scavenging losses should appear. Furthermore, both engines operate with the same standard gasoline fuel at an injection pressure of 300 bar. The ignition timing was adjusted in order to reach a MFB50% of 7-8°CA a. TDC (for efficiency reasons) and each load point was set to the same fuel mass flow to compare with the same available convertible energy. The engine parameters are summarized in Table 13.

Table 13: Engine parameters

concept	Uniflow	Loop
SOI	110°CA b. TDC	
ignition	adjusted for MFB50% of 7-8°CA a. TDC	
fuel pressure	300 bar	
fuel	Super 95	

The following graphs of global exhaust lambda and air trapping efficiency illustrate the engine behaviour of both concepts at the speed of 2000 and 3000 rpm and an IMEP between 4.5 – 9 bar. At the same IMEP and same fuel mass flow it can be seen, that a lambda = 1 concept could not be achieved for the same boundary conditions with a standard loop concept. Figure 45 illustrates the resulting lambda at the exhaust muffler. At low load and speed the resulting lambda of the loop concept has a value of 1.5, whereas the uniflow type achieves a lambda of 1.05 at the exhaust muffler. With increasing the load, the resulting exhaust lambda increases continually. By increasing the speed, the loop concept could improve its scavenging efficiency, whereas the resulting exhaust lambda enhanced to a lower level. Mainly, the lack of exhaust gas dynamic of a resonance muffler is responsible for this behaviour of the loop concept. In comparison, the uniflow concept achieves for both speeds nearly lambda = 1 at the exhaust muffler with the used parameters. Only at higher loads and speed the lambda increases, which requires an adjustment of the exhaust parameters.

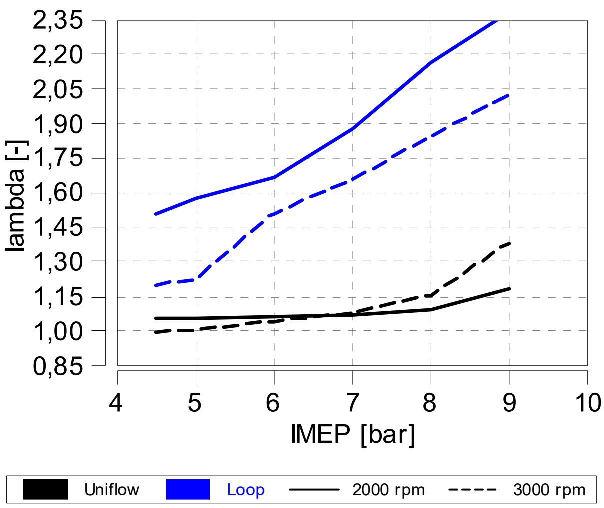


Figure 45: Global exhaust lambda

Figure 46 displays the characteristic of the air trapping efficiency for both concepts. The uniflow concept achieves a high trapping efficiency above 90 % in a wide operating range without adjusting the exhaust parameters. By increasing the load at 2000 rpm the trapping efficiency is nearly stable. Only over 8 bar IMEP the TE decreases slightly to a level of 81 %. With higher speed the TE actually reached a value of 98 % at low load but the decrease begins constantly with increasing the load. As mentioned before, the high lambda of the loop type can also be seen in the TE. Without a resonance exhaust muffler system, a maximum TE of 80 % could only be reached with low load. The higher speed has a positive impact on the TE and, therewith, also on the exhaust lambda.

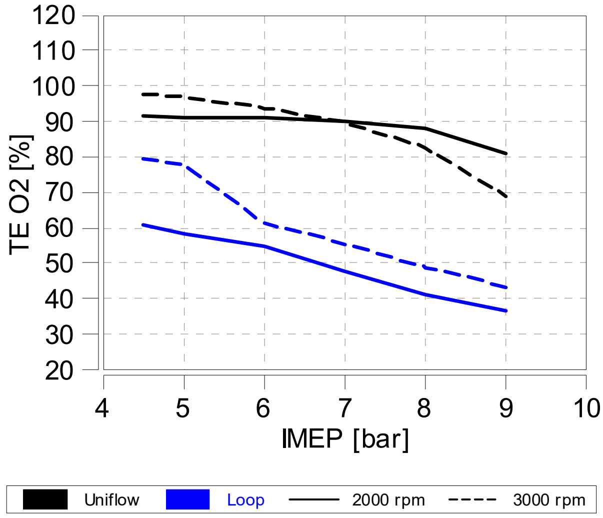


Figure 46: Air trapping efficiency

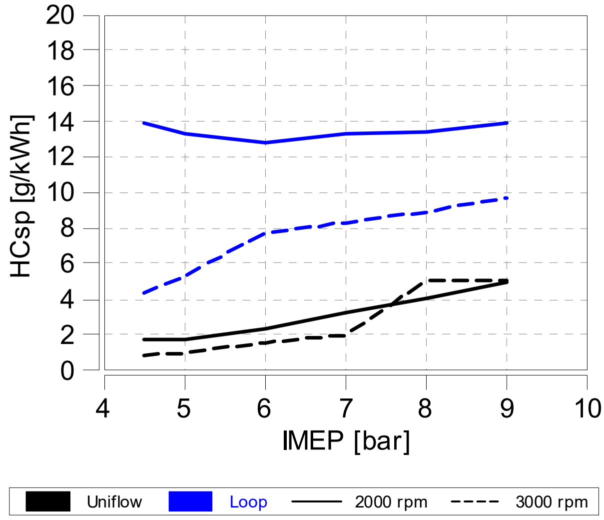


Figure 47: Specific HC emissions

The following Figure 47 - Figure 49 illustrate the specific HC, CO and NO_x raw emissions of both concepts in detail. It can be seen, that the uniflow concept generally reaches low specific emissions, especially HC and CO. The HC-emissions of the loop concept are 2 - 8 times higher, depending on speed and load. Comparing to Figure 45, which shows a lean global exhaust lambda of the loop concept, the CO-emissions are on a higher level. This suggests, that the in-cylinder lambda of the loop scavenged engine is much richer than that of the uniflow type. Therefore, also higher HC-emissions result from incomplete combustions due to lack of oxygen. The behaviour of a richer in-cylinder lambda can also be observed for NOx-emissions, especially with rising load. The NO_x-emissions are decreasing, which is a typical behaviour for rich combustion. The theoretical NO_x peak is between 1.05 -1.1. At the uniflow concept, the global lambda is also rising with higher loads, because of rising air scavenging losses, but also an increase of the NO_x-emissions occurs with higher combustion temperatures.

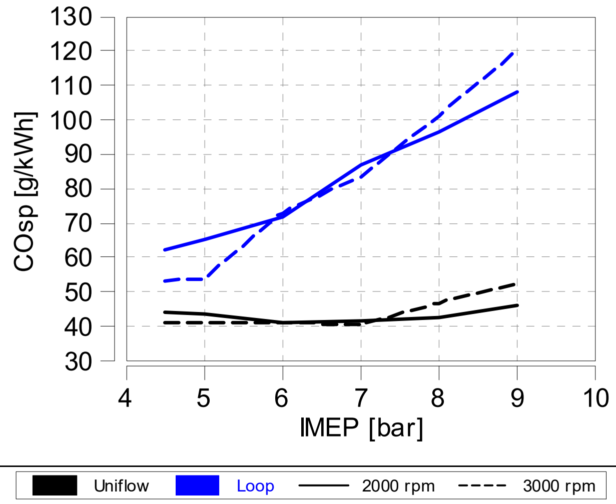


Figure 48: Specific CO emissions

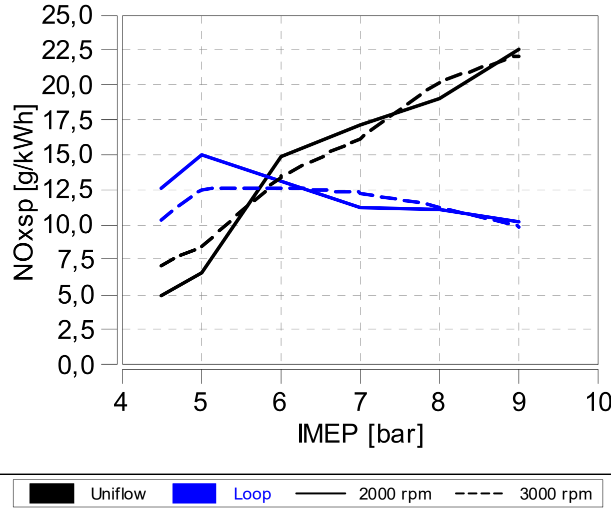


Figure 49: Specific NOx emissions

Figure 50 and Figure 51 illustrate the combustion stability using the coefficient of variation COV of the IMEP and the resulting knock index for each concept. The concept of the uniflow scavenging prototype has a higher IMEP COV and knock index. However, a detailed study of the mixture preparation by variation of the injection timing showed, that the used late SOI has a negative effect on the IMEP COV and the knock index for this concept. This is assumed to be due to worse homogenization of the uniflow concept in comparison to the loop scavenged concept for late SOI operation mode and the chosen injector spray type. Earlier fuel injection results in a better or the same IMEP COV as the loop concept. The loop scavenged concept shows an increased knock index for low load at high speed due to low charge motion and subsequently low turbulence as wells as a higher residual gas content in the cylinder.

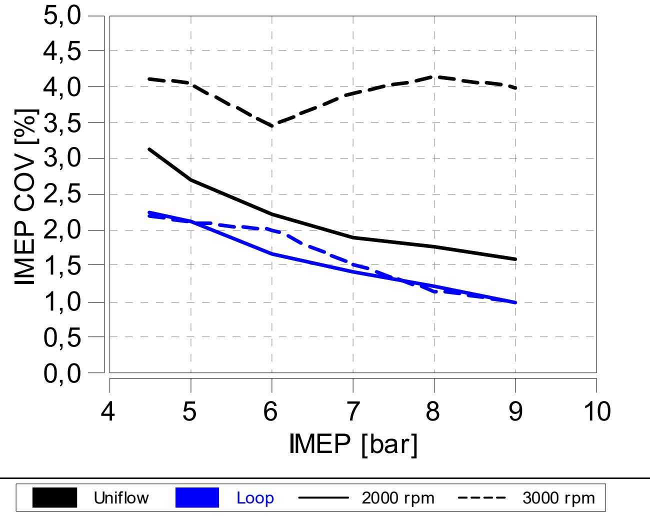


Figure 50: IMEP COV

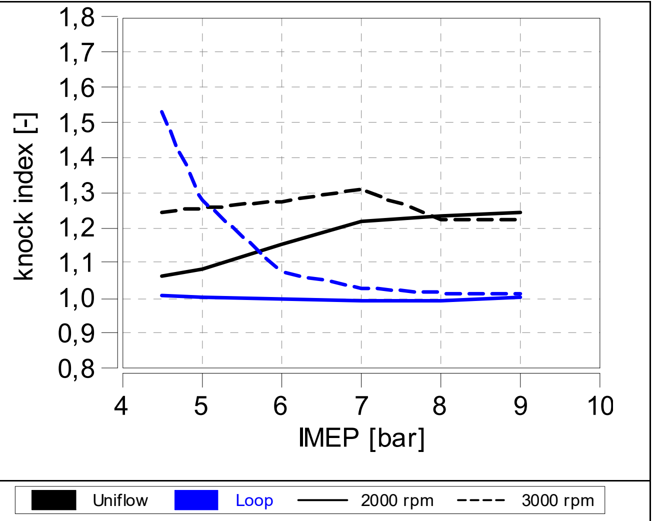


Figure 51: Knock index

Significant differences between these two scavenging concepts can also be seen in the spark plug and exhaust temperatures in Figure 52 and Figure 53. Although the spark plug position is different for both concepts and the spark plug temperature is therefore not direct comparable, it illustrates clearly that the temperatures at the head and therefore of the spark plug are higher at the uniflow concept because of the proximity of the exhaust port. This fact has possible advantages by minimizing soot deposits on the spark plug but it could cause a thermal overload or combustion anomalies like pre-ignition.

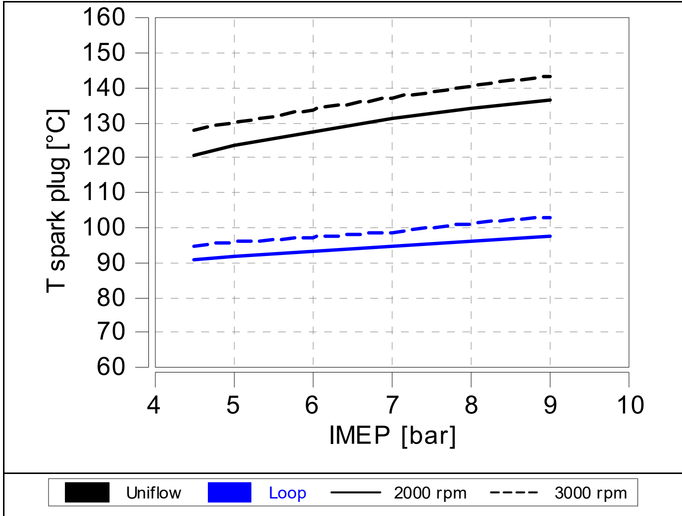


Figure 52: Spark plug temperature

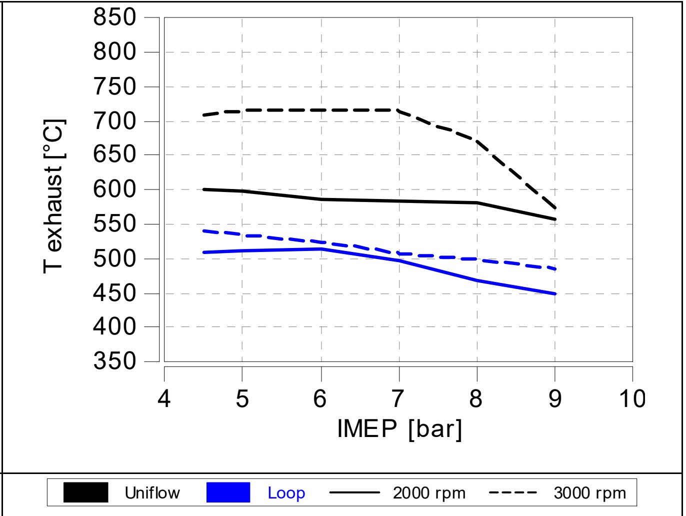


Figure 53: Exhaust temperature

The earlier exhaust open timing of the uniflow concept result in significantly higher exhaust temperatures than the ones of the loop concept. This is also a result of the higher trapping efficiency, because less cold air from the scavenging process will be mixed at the exhaust stream which decreases the temperature. At 3000 rpm and a higher load than 7 bar IMEP, the exhaust temperature of the uniflow concept is decreasing rapidly. This fact can be explained by a decreasing air trapping efficiency and mixing with cold air at the exhaust stream.

Operation with exhaust gas aftertreatment system

For a usage of this concept, a suitable aftertreatment system for a lambda = 1 operation is necessary. Therefore, a conventional 3-way catalyst of a serial motorcycle with similar engine dimensions was used. In Table 14 basic data of the used catalyst can be seen.

Table 14: Data of catalyst

Cover diameter	mm	72
cover length	mm	120
matrix diameter	mm	70
matrix length	mm	115
volume	l	0,443
cell density	cpsi	400

To evaluate the influence of the parameters load, exhaust valve lift and exhaust valve opening timing on the engine equipped with an exhaust gas aftertreatment a complete DoE with the following parameters of Table 15 was performed.

Table 15: List of variable parameters

Speed	rpm	1000 - 3000
IMEP load	bar	2.5 - 9
valve lift (duration)	mm (°CA)	7 (110) / 8 (118)
exhaust valve opening	°CA a. TDC	95 - 110
SOI	°CA a. TDC	100 - 160

Following engine parameters were kept constant for this test.

- Global lambda (1)
- MFB50% (7 - 8 °CA a. TDC)
- Fuel pressure (300 bar)
- Fuel (synthetic blend)

The testbench data was analysed with AVL CAMEO and mathematical models of the engine output data were compiled to optimize for each load point and speed as follows:

- Minimize ISFC
- Minimize HC emissions
- Minimize NO_x emissions

There are more than one optimization criteria, therefore it is not possible to fulfill the best value of each parameter in each point. With this optimization process, a pareto front was generated for each operation point which consists of points with possible optimal parameters. Out of these pareto fronts the best guess solution using engineers experience were taken as a optimum and shown in the subsequent graphs for a map between 1000 – 3000 rpm and 2.5 – 9 bar.

As can be seen in Figure 54, the possibility of a lambda = 1 concept can be reached in the whole investigated operation area. At 3000 rpm, a maximum power of 22 kW was reached with this fuel.

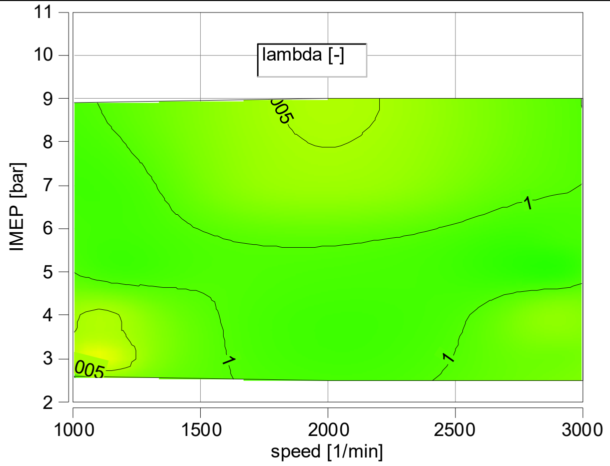


Figure 54: Global exhaust lambda

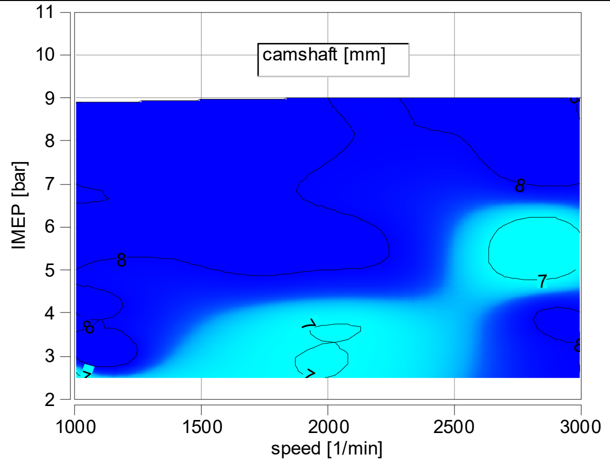


Figure 55: Valve lift

In Figure 55 and Figure 56 the necessary parameters of the exhaust, in detail the used camshaft and the exhaust opening timing, are displayed. It shows that for an operation in the whole a map, a variability of the scavenging process is needed to ensure an optimized scavenging process and minimize air scavenging losses. At low load, a smaller cam lift of 7 mm was needed for best conditions and the exhaust opening timing was shifted to an earlier opening. For remaining operation points the 8 mm cam lift was sufficient. Only at 3000 rpm and an IMEP between 4 – 5 bar, a spot was localized, where the 7 mm camshaft with an early opening delivers a better performance for the defined optimization criteria. At this spot, a possible strong influence of the intake and exhaust pressure dynamics could be the reason for this behaviour on the single cylinder research engine. With increasing load, a shift to a later opening of the exhaust opening timing showed the best performance for the scavenging process.

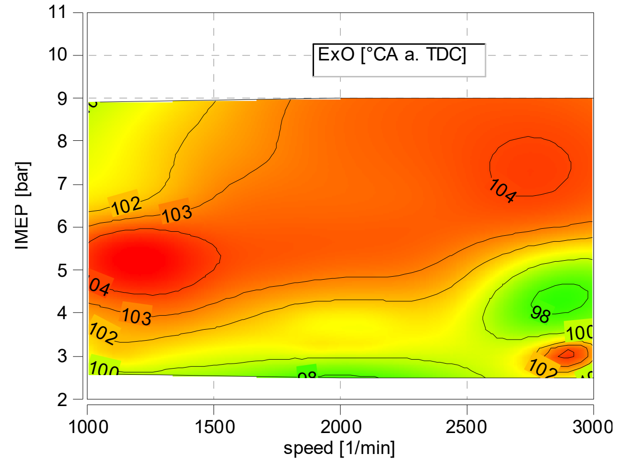


Figure 56: Exhaust valve opening

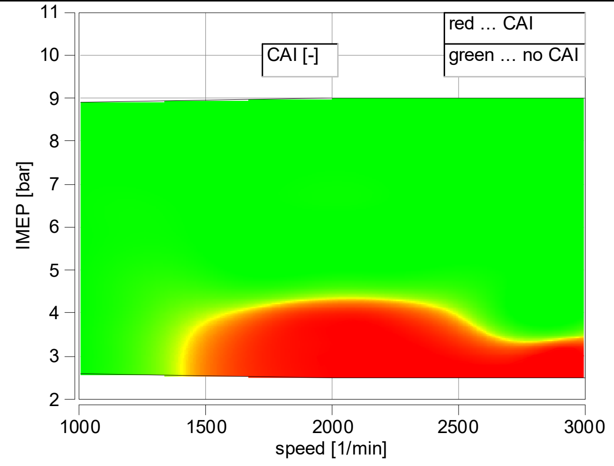


Figure 57: Controlled auto ignition

During the test, in the area between 2000 and 3000 rpm and low load conditions a special operational behavior was detected in, which can be seen as red area in Figure 57. This area is called “controlled auto ignition” (CAI) in which no decisive influence of the ignition timing on the combustion process is present. To evaluate this engine behavior, the position of the ignition was placed at TDC, where the combustion was already in progress and the mode (CAI) was analyzed and added for visualization to the testbench data at the post-processing. In this area, the combustion process can be influenced by the injection timing only. An early injection timing (170°CA b. TDC) causes a rapid increase of the in-cylinder pressure in the near of TDC with a MFB50% of 3°CA a. TDC. By shifting the injection to late (120°CA b. TDC) the MFB50% can be moved by 10°CA to a later position.

This behavior indicates, that the injected fuel interacts with the remaining hot residual gas content and/or hot parts in the cylinder and starts pre-reactions which leads to a self-ignition. It needs to be noticed, that this mode could not be reached at 1000 rpm with the variation range of the parameters of Table 15. The measured exhaust temperature at this speed is lower (up to 100°C) than in higher rpm ranges, indicating a lower in-cylinder gas temperature which possibly causes an insufficient chemical pre-reaction for this CAI mode. But in general, this combustion mode can be extended to lower speed areas also. For visualization, the burning duration between MFB10%-MFB50% is illustrated in Figure 58 and the border of the red CAI region is added as dotted black line. It can be seen, that the combustion process in the CAI region is quite fast and gets slower in the higher loads. This passage between these two zones, where the burning duration is still on a short level, could be interpreted as a spark assisted CAI mode.

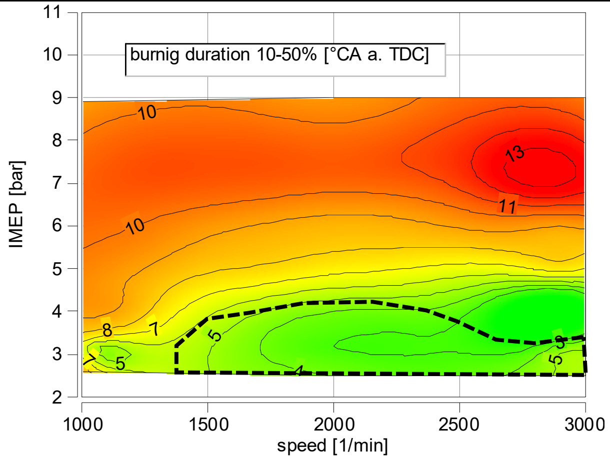


Figure 58: Burning duration MFB10%-MFB50%

Table 16 contains the measured values of pre-catalyst and after the catalyst emission and the conversion rate of the used catalyst in one operation point. A reduction of NO_x and CO was obtained above 80 % and for THC a rate of 69 % was achieved in that load point and with the used alternative fuel.

Table 16: Emissions data pre-/after catalyst

Speed	IMEP	Lambda	CAT	THC	NOx	CO
rpm	bar	-	-	ppm	ppm	vol.%
2000	5	1	pre	1244	972	1,30
			after	379	136	0,26
conversion rate			%	69	86	80

Tail pipe HC and NO_x-emissions (after catalyst), are shown in Figure 59 and Figure 60. Overall, the HC emissions are on a low level, while around an operating point of 1000rpm / 5 bar IMEP a spot with higher HC-emissions was measured as a result of bad mixture preparation.

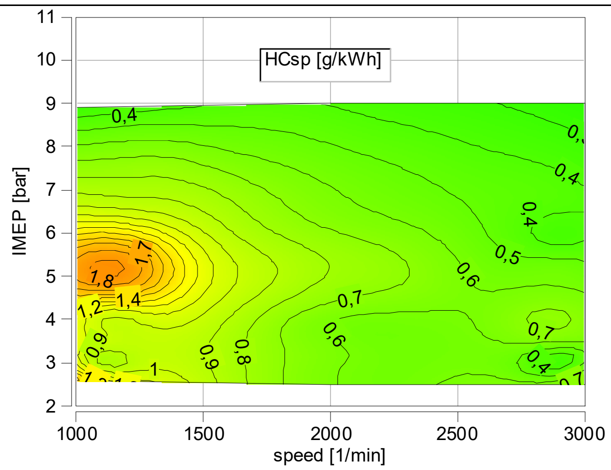


Figure 59: Specific HC-emissions after catalyst

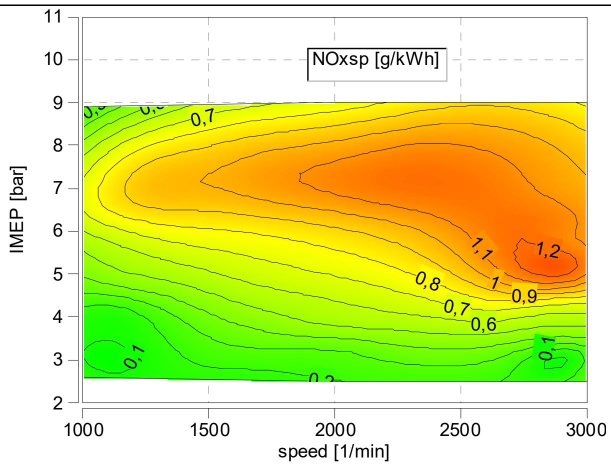


Figure 60: Specific NO_x-emissions after catalyst

With increasing load, the NO_x-emissions are rising, which is related to a decrease of the air trapping efficiency (see Figure 46) and consequently higher O_2 concentration in the exhaust gas, leading to lower catalyst conversion rates. The highest spot of the remaining NO_x-emissions is at 3000 rpm and 4-5 bar IMEP, which is the same spot where the 7 mm cam lift was used.

Figure 61 illustrates the indicated specific fuel consumption, which is corrected for the difference of lower heat value (~6%) between the synthetic fuel compared to standard gasoline. It can be seen, that this concept reaches low indicated fuel consumption, especially in lower load conditions where the CAI mode (black dotted line) and the spark assigned CAI mode (gray dotted line) with a maximum duration between MFB10%-MFB50% of 7°CA are used.

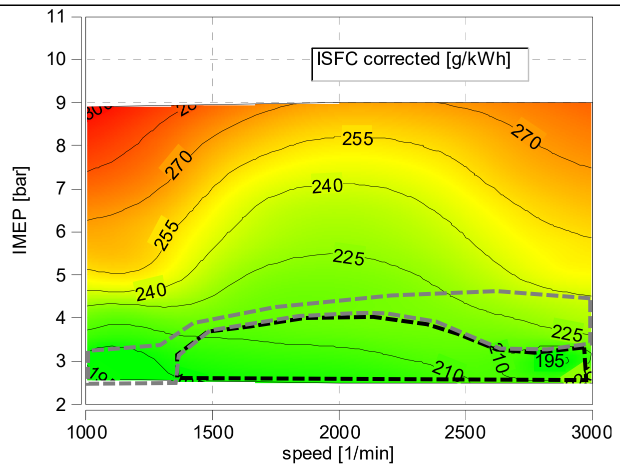


Figure 61: ISFC corrected

Summary/Conclusions

The first prototype has proven, that this uniflow scavenged two-stroke concept can reach a lambda = 1 operation in a wide range with comparable low raw emissions. In order to evaluate the potential for automotive applications, further investigations are necessary and a balancing of increased cost and effort due to increased complexity from valve train variability is necessary.

4 Ergebnisse und Schlussfolgerungen

The development of a modern two-stroke engine for current applications in the automotive sector, as an example as a hybrid compound, includes ambitious challenges and priorities to be a real alternative to a state of the art four-stroke engine. The chosen concept needs to meet durability requirements at least at the same level and must have advantages in terms of packaging, weight with a cost-effective and efficient exhaust aftertreatment throughout the entire operational range. A uniflow scavenged two-stroke engine with external scavenging blower, high-pressure direct injection system and a separate lubrication was identified and developed as a promising concept. To get a proof of this new concept in comparison to a classic loop scavenged two-stroke, two geometrically identical engines, in terms of stroke and bore, but with different scavenging concepts were performed on the testbench under the same boundary conditions. Due to the used scavenging method of a uniflow concept and the additional possibilities of influencing the charge cycle, advantages in terms of air trapping and lower emissions have been demonstrated. Furthermore, the prototype was adapted with a 3-way catalyst as aftertreatment system and operated at a lambda = 1 condition.

The raw emission level and the tail-pipe emissions are low compared to a standard loop scavenged two-stroke engine, but for an application in a hybrid automotive they are still too high. Especially at higher loads, the trapping efficiency needs to be increased to convert the NO_x-emissions to a lower level. The used CAI mode in lower load areas is a promising feature for a good fuel consumption and low emissions. For automotive application, this CAI mode operation area has to be extended towards lower speed ranges and as well as higher loads.

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6 Abkürzungen

PHEV	plugin hybrid electric vehicle
CO	carbon monoxide
HC	hydrocarbons
NOx	nitrogen oxides
M_{sr}	reference mass swept volume (compression + stroke volume)
M_{dr}	reference mass swept volume (compression volume)
M_{as}	mass of fresh charge
M_{sl}	mass of scavenging losses
M_{tas}	mass of trapped air delivered
M_{tr}	total mass of charge retained at exhaust closure
M_{ta}	mass of trapped air in cylinder
M_{cg}	mass after combustion
SR	scavenging ratio
DR	delivery ratio
TE	trapping efficiency
SE	scavenging efficiency
SL	scavenging losses
EGR	exhaust gas recirculation
CAT	catalyst
CAI	controlled auto ignition
TDC	top dead center
BDC	bottom dead center
IO	intake open
IC	intake close
EO	exhaust open
EC	exhaust close
GDI	gasoline direct injection
REx	range extender

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