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2-Stroke Scavenging in Conventional and Minimally-Modified 4-Stroke Engines for Heavy Duty Applications at Low to Medium Speeds

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Abstract: The transformation of a standard 4-stroke cylinder head into a torque-improved and gradually more efficient 2-stroke design is discussed. The concept with an effective loop scavenging via an extended inlet valve holds promise for engines at low- to medium-rotational speeds for typical designs of conventional 4-stroke cylinder heads. Calculations, flow simulations, and visualizations of experimental flows in relevant geometries and time scales indicate feasibility, followed by a small engine demonstration. Based on presumably long-forgotten and outdated patents, and the central topic of this contribution, an additional jockey rides on the inlet valve's disk (facing away from the combustion chamber) and reshapes the in-cylinder flow into a reverted tumble. A quick gas exchange with a well-suppressed shortcut into the open exhaust is approached. For overall mechanical efficiency, the required charge pressure for scavenging is of paramount importance due to the short scavenging time and the intake's reduced cross-section. Herein, still acceptable charging pressures are reported for scavenging periods equivalent to low or medium rotational speeds, as characteristic for heavy-duty applications. Using widely available components (charger, direct injection, variable camshaft angles) an increased engine efficiency is suggested due to the 2-stroke's downsizing effect (relatively less internal friction as well as the promise of more torque and a decreased size).

Keywords: two-stroke engine; scavenging; in-cylinder flow

1. Introduction

It is well known [1–3] that a more "ideal" combustion engine would alternatingly compress and expand (2-stroke) rather than execute an additional idle stroke for gas exchange with no net output (on the contrary, internal friction is realized), i.e., the common 4-stroke. However, a well-known issue for 2-stroke operation is the demand for a quick and well-separated gas replacement (scavenging) around the bottom dead center of the piston [4,5]. Nowadays, engines with the highest fuel efficiency (above 50%) are 2-stroke, for example, large ship diesels with their relatively long stroke utilizing uniflow scavenging with inlet ports in the lower cylinder wall [6]. In addition, very compact 2-stroke engines with poor fuel efficiency and poor exhaust composition (for handheld machinery: Leaf blowers or chainsaws, etc.) are based on inlet and outlet ports in the cylinder walls. For a number of reasons such as wear [7], lubricant film in contact with the ports' high-speed flow (particle emission, oil consumption), thermal problems, prolonged piston with more mass required, etc., such ports in the cylinder walls [8] are also undesirable for medium-sized, long-lasting, preferably clean and efficient engines.

Therefore, 2-stroke concepts with well-established poppet valves for inlet and outlet in the cylinder head have been investigated. The figure below (Figure 1) schematically displays the three operational sectors of this approach over a full rotation of the crankshaft.

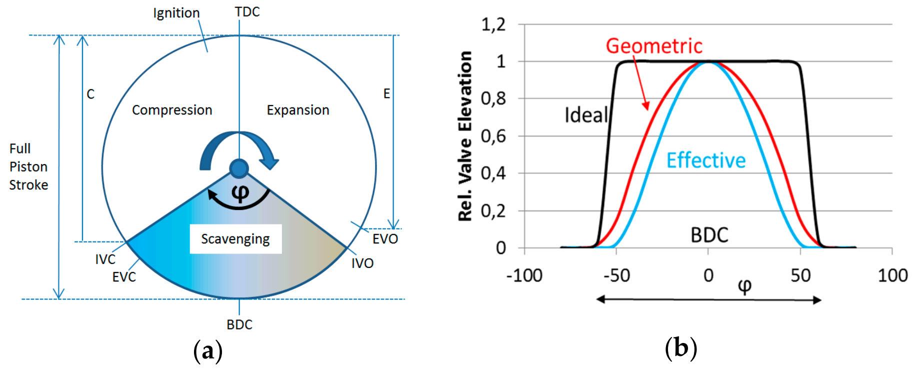


Figure 1. (**a**) 360° crankshaft cycle for a 2-stroke engine with conventional poppet valves. The full rotation can be separated into three sectors: Compression (including ignition), expansion, and scavenging. As a challenge, the scavenging angle φ is rather limited to leave a sufficient compression stroke C and expansion stroke E with respect to the fully available piston stroke. (**b**) Normalized valve elevation within φ. The effective cross-section (blue) for scavenging is smaller than the geometrical valve stroke (red) and accounts for only 50% of an ideal (=unrealistic) valve with instantaneous open/close characteristics (black). The abbreviations are as follows: TDC = piston's top dead center, E = expansion stroke, EVO = exhaust valve opening, IVO = intake valve opening, BDC = bottom dead center, EVC = exhaust valve closing, IVC = intake valve closing, and C = compression stroke. The scavenging angle φ is determined by the camshaft profile and strongly determines the whole approach.

While compression, ignition, and expansion do not principally deviate from 4-stroke engines, the scavenging process poses (a) quantitative and (b) qualitative challenges. An increased scavenging angle φ over-proportionally reduces the power-relevant strokes C and E, and the advantage over a 4-stroke can be easily lost, as calculated below. Thus, the applicable φ must be much shorter (definitely less than 180°) than an intake stroke of a 4-stroke engine. On the other hand, too short of a φ directly reduces the effective time-cross-section (Figure 1b) for scavenging and demands a strongly increased pumping work (pressure) from a charger, reducing the engine's net output and efficiency.

The scavenging process is also delicate in terms of quality. Unlike the well-separated exhaust and intake stroke in a 4-stroke engine, the provided fresh gas (i) partially mixes with the exhaust gas, (ii) does not completely expel the residual exhaust gas, and (iii) partially enters the exhaust without contributing to the engine's combustion. These effects are strongly determined by the in-cylinder flow during scavenging.

This contribution addresses the quantitatively limiting scavenging angle φ and the qualitative in-cylinder flow.

A conventional 4-stroke cylinder head without modifications is not very suitable for 2-stroke operation; the loss of fresh gas into the simultaneously open outlet (a shortcut) is very strong, as is well-known [5] and shown below. The issue of poor-quality scavenging was gradually relieved with modified geometries for the cylinder head and the piston. These geometries, e.g., smaller inlet valves in a vertical intake port [8–11] and/or edges in the combustion chamber and piston crown [12,13] or stepped pistons [14] are, however, not very attractive in terms of (i) considerable re-engineering with relatively higher engine designs and (ii) a potentially affected combustion (due to rugged and increased inner surfaces of the combustion chamber).

A significant loss of fresh air into the exhaust demands an over-proportionally increased filling volume at full load (more than the displacement). This necessitates more effort from a charger to the cost of overall efficiency. Furthermore, the exhaust gas volume increases (necessitating a larger cross-section for the exhaust system), and the exhaust's oxygen content is increased (confusing, e.g., a lambda probe [5]).

It is the goal of this contribution to (i) largely conserve the established design of 4-stroke cylinder heads, the piston, the piston drive, and the shape of the combustion chamber and (ii) to nevertheless provide an efficient gas exchange for 2-stroke operation. From the viewpoint of engineering and production, every change for an established design is undesirable and prolongs or hampers implementation. With direct fuel injection [15,16] (standard for diesel and widespread nowadays for gasoline engines) the concerns regarding fuel losses appear manageable. The general promise is more power at relatively less internal friction for a typical heavy duty engine at low- to medium-rotational speed, where the limited φ for scavenging (Figure 1) is less critical due to more available time, i.e., less pumping work is demanded from a slower scavenging flow. Camshaft phasing for dynamic adjustments of EVO, IVO, EVC and IVC (Figure 1) could be particularly helpful and is, nowadays, widespread in engines for vehicles.

2. A Presumably Forgotten Solution

A potentially attractive concept for a qualitatively advantageous 2-stroke scavenging in an almost normal 4-stroke design is presented via a modified intake valve (Figure 2) which, by itself, induces a directed flow instead of changing the complete cylinder head and intake geometry [8–11] or the combustion chamber [12,13]. It must be noted that a modified inlet valve for 2-stroke operation is not a new idea but was already treated by many and mostly outdated patent applications, including a very related invention from Jeong et al. [17] in 1996. Nevertheless, even much older inventions point to virtually the same approach. These ideas were, however, hardly realized. Several other requisites, desirable for clean and efficient 2-stroke operation, were not matured or available at that time: Direct injection, camshaft phasing, efficient chargers, and electronic engine control. Further concerns were a premature wear of the valve´s guide bush due to the asymmetrical shape of the valve, which induces oscillating bending torque to the valve's stem.

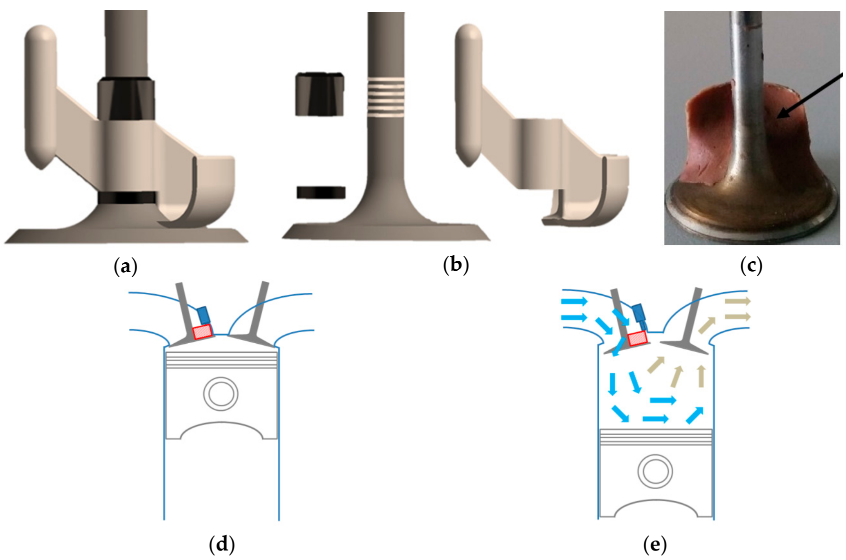


Figure 2. (**a**,**b**) schematic of the almost original inlet valve with an additional jockey. The complex shape of the jockey is vertically guided with, e.g., a radially pressed-on sleeve (black pieces). The jockey is held in a fixed angular position (guiding structure in the inlet channel not shown) and thus will not follow the valve´s rotation. The jockey redirects the fresh gas and a certain angle of the valve disk is almost completely obstructed. (**c**) Experimental inlet valve for flow visualizations with attached 120° jockey structure. (**d**) Arrangement in a nearly conventional 4-stroke engine design at TDC. (**e**) Schematic of scavenging near BDC; the flow-steering jockey is marked in red and the angular guiding structure in dark blue.

A particular concern is too large of an obstruction of the inlet flow, which then results in excessive pumping losses for scavenging: A central topic of this work.

The almost unchanged intake valve is equipped with an additional part (i.e., the "jockey," Figure 2) which follows the valve's vertical movements. The jockey is not in contact with the combustion, i.e., no distinct heat problem is expected in the cool inlet flow. Furthermore, no delicate sealing requirements apply to the jockey. The intake valve may rotate when the jockey is kept in a fixed angular position, e.g., with a guiding tongue (or pin) inserted in the inlet channel (Figure 2d,e).

Small graphite pins in the jockey's guiding sleeve could provide sufficient lubrication against wear due to the slowly rotating valve. In addition, a graphite inlay in the inlet´s angular guiding structure could prevent wear. Uneven mass-distribution of the jockey with respect to the valve´s axis—inducing critical bending torque to the stem—could be countered or compensated by (a) a thickened valve stem (presumably undesirable), (b) a preferably light jockey, or (c) streamlined counterweights as shown in Figure 2a,b.

As already claimed in the old patents [17], the shape of the jockey obstructs the annular gap of the open valve toward the nearby and open outlet, and it entirely re-directs the inlet flow toward the cylinder wall away from the opposing outlet valve. The effectively blocked circumference is herein denoted as the obstruction angle α. The experimental jockey structure in Figure 2c results to $\alpha \approx 120°$. As is schematically shown in Figure 2e and closely related to the vertical intake port design [11], fresh gas then descends down to the piston (in this phase near BDC) and U-turns toward the outlet region (loop scavenging), thereby more effectively replacing the cylinder's volume before entering the outlet. The directional nozzle, formed by the valve disk and jockey and effective in the last possible placement of the flow, can enforce a comparably intense tumbling, as shown below. An angular momentum or tumbling of the filling gas is achieved, which is appreciated in many engine concepts and might be further exploited (but this is not the scope of this contribution). Furthermore, the cold and distinct stream covers the piston surface and provides some extra cooling for this highly-stressed part.

It should be noted that a fixed guiding structure within the intake [8–11] (instead of a jockey) cannot perform in the same way; the annular gap of the radially re-directing valve disk would still be fully opened toward the very close outlet and premature losses would still occur [10,11], as is also simulated below.

Naturally, the reduced φ for gas exchange together with the intake's further reduced cross-section—both with respect to a conventional 4-stroke—would inhibit high engine speeds or would demand higher charging pressures at high speed. As we will see, however, moderate pressures (say, 30 kPa or 0.3 bars) appear sufficient for medium speed. The small-capacity test engine runs at low rpm even with zero charging, filled solely by the momentum of the exhaust gas column.

3. Principal Calculations for a Symmetrical Scavenging Around BDC

For an analysis, we begin with the idealized assumption that perfect gas exchange occurs within the scavenging period φ (here centered at BDC as indicated in Figure 1), and no work is spent on internal friction or fresh gas pumping. The relative power output with respect to a 4-stroke engine with the same displacement (also with no internal friction) is then:

$$\frac{P_{2\text{-}stroke}}{P_{4\text{-}stroke}} = \left(1 + \cos\left(\frac{\varphi}{2}\right)\right) \tag{1}$$

and would be 2 for $\varphi = 0°$ (an infinitesimally quick scavenging at BDC) and 1 for $\varphi = 180°$ (equivalent to the halved stroke at a doubled combustion frequency). Thus, while $\varphi = 0°$ is obviously not feasible, increasing φ too much will finally annihilate any potential advantage.

When introducing an engine's mechanical losses (internal friction, valve train, pumping work, and other auxiliary devices) a 4-stroke's output can be assumed to be decreased to 80%. This equals a lump sum of 20% for internal friction [18,19] or 0.2 when normalizing the 4-stroke's internal power output to 1.

Two systemic deviations induce higher losses in the proposed 2-stroke design. One additional loss is caused by the valve train, which now cycles at the crankshaft´s speed. In a 4-stroke, the valve train's energy cost W_{Valve} is typically smaller than 0.03 [18,19], and we double this number for 2-stroke operation to 0.06, i.e., the herein assumed loss from the valve train approaches 50% of the losses from together the piston assembly and crank train (these are much more massive parts with larger surfaces in frictional contact).

Another mechanical loss of utmost importance here is caused by the required pumping work W_P for scavenging, which is essentially determined by the flow's time-cross-section (Figure 1b). Unlike a 4-stroke, the pumping work must be provided by an additional device, e.g., a blower or supercharger. At a given rotational speed of the engine, the W_P losses strongly increase at a smaller φ and a further reduced cross-section, as is herein applied for the intake valves:

$$W_P = W_0\left(\frac{180^\circ}{\varphi}\right)^2\left(1+\cos\left(\frac{\varphi}{2}\right)\right)^2 \tag{2}$$

The flow's required speed is reciprocal to φ [in Equation (2), the first term in parentheses] and proportional to the remaining displacement, which decreases with φ (Figure 1a) (the second term in parentheses). The kinetic energy of the flow, however, increases with the square of the speed. The referential W_0 must be determined from flow simulations and/or flow measurements of the representative arrangements. W_0 is herein defined as the required work for providing half of the displacement [φ = 180°, Equation (1)] within the period of φ = 180° at a given rotational speed. Values for W_0 are derived from the subsequent simulation and experiment.

Furthermore, and unlike in a 4-stroke, the gas replacement (G) in a 2-stroke will never be at 100%, i.e., all exhaust gas being perfectly replaced by fresh gas. Instead, a scavenging efficiency G of 80% and 90% is calculated herein [10,11], which will further and directly reduce the engine's available power.

With the 4-stroke's normalized power = 1 (without friction or any mechanical losses), Equation (1) for the equivalent 2-stroke engine at the same speed and with mechanical losses and the limited scavenging efficiency G now writes as:

$$P_{2\text{-}stroke} = G\left(1+\cos\left(\frac{\varphi}{2}\right)\right) - 0.16 - W_{Valve} - W_0\left(\frac{180^\circ}{\varphi}\right)^2\left(1+\cos\left(\frac{\varphi}{2}\right)\right)^2 \tag{3}$$

The 4-stroke's lump sum of losses (20% or 0.2) was replaced by 0.16, since the 4-stroke's valve train losses (0.03) and pumping work (0.01) is separately accounted as W_{Valve} = 0.06 and W_P [Equation (2)]. The subsequent Figure 3 shows the image of Equation (3) as a function of φ and a different G and W_0.

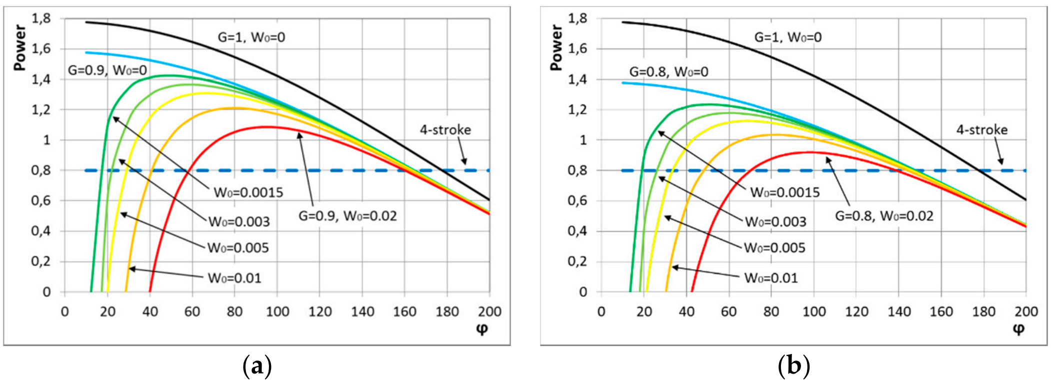


Figure 3. Calculated power or torque [Equation (3)] of the proposed 2-stroke design with poppet valves, with respect to a 4-stroke with the same displacement. Besides the mechanically pre-adjusted φ, the referential pumping work W_0 and the scavenging efficiency G largely determine a potential gain or loss. (**a**) G = 0.9. (**b**) G = 0.8.

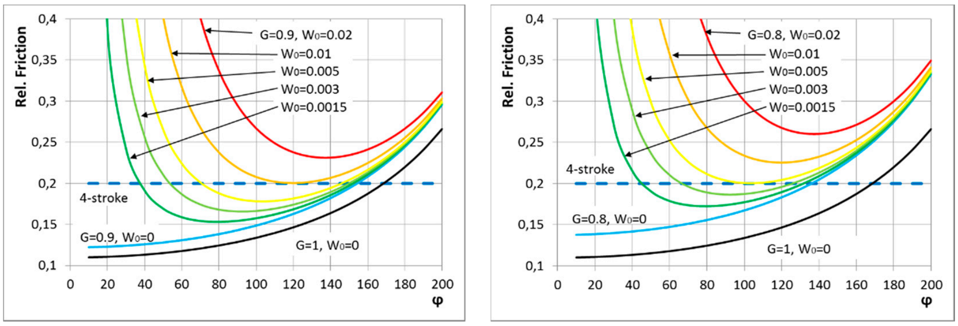
The potential gain or loss with respect to an equivalent 4-stroke with power output = 0.8 strongly depends on the gas replacement efficiency G and the referential pumping work W_0. Moreover, a sufficiently small W_0 would allow for more (=excessive) fresh gas and then a further increased G.

Since the required pumping work basically increases with the square of the engine's rotational speed, the W_0 in Figure 3 can also be read as proportional to the square of the engine's speed. As proposed, an advantage over the 4-stroke occurs for preferably low- to medium-speeds, equivalent to small W_0. For a more exact quantification, a representative number for W_0 is still missing at this point.

While the absolute mechanical losses per single crankshaft revolution are inherently higher than in a 4-stroke (even for W_0 = 0 the W_{Valve} is higher for 2-stroke), the relative friction F or relative mechanical loss is the herein relevant quantity for efficiency considerations:

$$F = \frac{0.16 + W_{Valve} + W_0\left(\frac{180^\circ}{\varphi}\right)^2\left(1+\cos\left(\frac{\varphi}{2}\right)\right)^2}{G\left(1+\cos\left(\frac{\varphi}{2}\right)\right)} \tag{4}$$

F should become smaller than 0.2 (the referential 4-stroke) to achieve a systemic advantage, simply more power (Figure 3) is not sufficient in terms of fuel economy. Again, the referential pumping work W_0 is of paramount importance. Figure 4 images Equation (4) and suggests an advantage at sufficiently low speeds: Ultimately, the referential pumping work W_0 becomes small enough at lower speeds, even at small time-cross-sections φ.



(**a**) (**b**)

Figure 4. Relative mechanical losses F at (**a**) G = 0.9 and (**b**) G = 0.8. The referential pumping work W_0 and the scavenging efficiency G largely determine a potential advantage or disadvantage with respect to a 4-stroke.

4. Flow Simulations

In-cylinder flow determines the qualitative scavenging efficiency G and was studied with simulations (Figure 5) for a schematic gasoline cylinder head (pent roof combustion chamber) with only two valves. Arrangements with four valves are shown in the experiments below. It must be noted that such 3D flow simulations are generally not very reliable for high speed and very turbulent phenomena. The simulations in Figure 5 (ANSYS academic license simulation software, ANSYS, Inc., Canonsburg, USA) show the quickly propagating fresh gas and corresponding millisecond time-scale for a schematic 500 cm^3 cylinder, which is prefilled with air at 300 K. A two-phase simulation with hot and already moving exhaust gas is more demanding, but (nevertheless) cannot promise much more reliable results.

A rapid inlet flow is enforced at 100 m/s. Such speed is required to provide the full displacement (500 cm^3) within 3 milliseconds for the simulated intake's cross-section. Therefore, in principle, at 5000 rpm (83 Hz, equivalent to 12 milliseconds for a full revolution), the complete displacement could be exchanged within 3 ms, equivalent to 90° of crankshaft angle around BDC (Figure 1).

Typical for 2-stroke operation, and herein simulated, both valves are open and the piston is situated near the BDC (Figure 2e).

For a conventional and unchanged 4-stroke cylinder head without jockey (Figure 5a), the inlet flow prefers the shortest route to the open outlet and hardly approaches the cylinder volume, i.e., scavenging is poor. A significant portion of exhaust gas would remain in the cylinder and *G* is unsatisfying. The calculated pressure drop of the rapid flow, from inlet channel to exhaust, is 40 kPa.

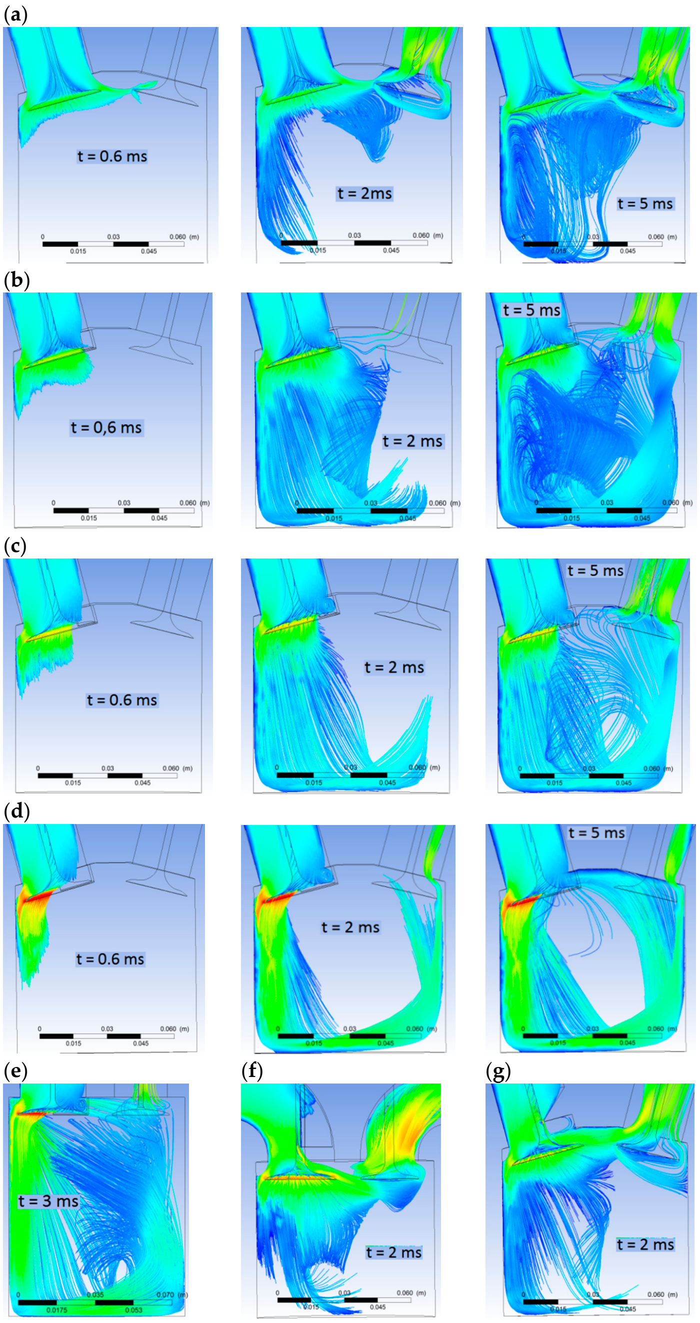


Figure 5. Flow simulations for a schematic 2-valve 500 cm^3 cylinder with propagating fresh air, forced to 100 m/s through the intake; images are color coded for velocity (dark blue: 50 m/s, green: 175 m/s, yellow: 250 m/s). (**a**) Unchanged inlet provides poor scavenging with premature and high losses. (**b**) Obstruction angle $\alpha = 90°$: Loop scavenging with less premature loss. (**c**) More distinct tumbling for $\alpha = 120°$. (**d**) Pregnant tumbling for $\alpha = 180°$. (**e**) Jockey for a schematic diesel cylinder with prolonged bore and flat head. (**f**,**g**) Fixed directors in the inlet do not perform in these simulations.

G improves with a schematic jockey on the inlet valve (Figure 2); a certain circumference of the valve's ring gap is obstructed toward the outlet valve. Different obstruction angles α are simulated. An α = 90° (Figure 5b) already enforces a reverted tumble with a prolonged U-turn path and low initial losses of the fresh gas, all the way expelling the exhaust gas. The calculated pressure drop is moderately increased to 55 kPa.

An intermediate α = 120° (Figure 5c) results in more distinctive tumbling with zero premature loss and a 70 kPa pressure drop.

In Figure 5d, the obstruction α was set to 180°. A very directed and high-speed tumble occurs in the simulation. Conversely, a stronger obstruction is present and requires higher charging pressures: 90 kPa in the simulation.

From simulations, we can see that the jockey also performs for diesel designs (Figure 5e, here α = 180°) with a prolonged bore and flat cylinder head.

Note that (Figure 5f,g) fixed directors in the inlet [9–11] do not perform well, at least in these simulations with various shapes; the unshielded valve disk effectively redirects a large portion directly toward the outlet.

For determining the very important W_0 from these simulations, the introduced cylinder in Figure 5a–d with 500 cm^2 and at 5000 rpm is assumed to deliver 20 kW output in ordinary 4-stroke operation, i.e., 25 kW internal power is present before internal friction and mechanical losses. This equals 300 J output for a 360° crankshaft cycle. As defined for W_0, at φ = 180° the required fresh gas volume is halved and the scavenging time is doubled, demanding only 1/4 of the applied 100 m/s in the simulation. The required driving pressure, therefore, reduces to 1/16 and results in 3.5 kPa (α = 90°), 4.4 kPa (α = 120°), and 5.6 kPa (α = 180°). The scavenging energy for 250 cm^3 is then 0.9 J (α = 90°), 1.1 J (α = 120°), and 1.4 J (α = 180°). These energies have to be doubled due to the effective time-cross-section (Figure 1b) of the valve.

For α = 120°, the W_0 at 5000 rpm can be calculated to 2.2 J/300 J = 0.0075, and this is represented by the orange trace (W_0 = 0.01) in the Figures 3 and 4. Thus, although additional power can be expected at 5000 rpm, the fuel efficiency—even with an ideal charger—would be lower than in an ordinary 4-stroke operation. This is changed with a reduced rotational speed: W_0 becomes 0.002 (25 % of the previous value) at 2500 rpm and now corresponds to the green traces in Figures 3 and 4, pointing to a superior efficiency and power (or torque) at φ = 80 to 100°.

Although the presentations in Figure 5 are instructive, flow simulations are not guaranteed to be the same as reality. Fortunately, the scavenging effects are still quick and effective with more modest charging in the subsequent experiments. This is at least partially supported by four valves and with a relatively higher cross-section in comparison to Figure 5, i.e., the practical W_0 for a given α is further reduced.

5. Scavenging Flow Experiments

The characteristic gas exchange is visualized on a representative time scale (0, 2, 3.5 and 5 ms) and in practical geometries with cold air (Figure 6). To the benefit of this cold air approach, it should be noted that (i) the real exhaust gas would have a more distinct momentum toward the outlet, and (ii) its density is significantly lower than the herein present cold air due to its higher temperature. The gas exchange G is generally higher in terms of relative masses than in relative volumes.

A conventional and unchanged 4-valve cylinder head (pent roof gasoline 4-stroke with horizontal intake ports) (Figure 6a) was completed with a transparent (Perspex) 500 cm^3; cylinder (Figure 6b) with a flat top (=the surface of the piston). The four valves are all open with the characteristic stroke of the valves, which is 8 mm. The inlet air flow was driven with a 15 kPa blower and the intake was instantaneously smoked with a small firecracker. Note that the density of the smoke deviates from shot-to-shot. Relevant here is the trace of propagating smoke (the "fresh gas"), which more or less effectively expels the clean air (the "exhaust gas"). As a further difficulty, the optimum angular orientation of a jockey is not clear for four valves, as it was for the two opposing valves in Figure 5.

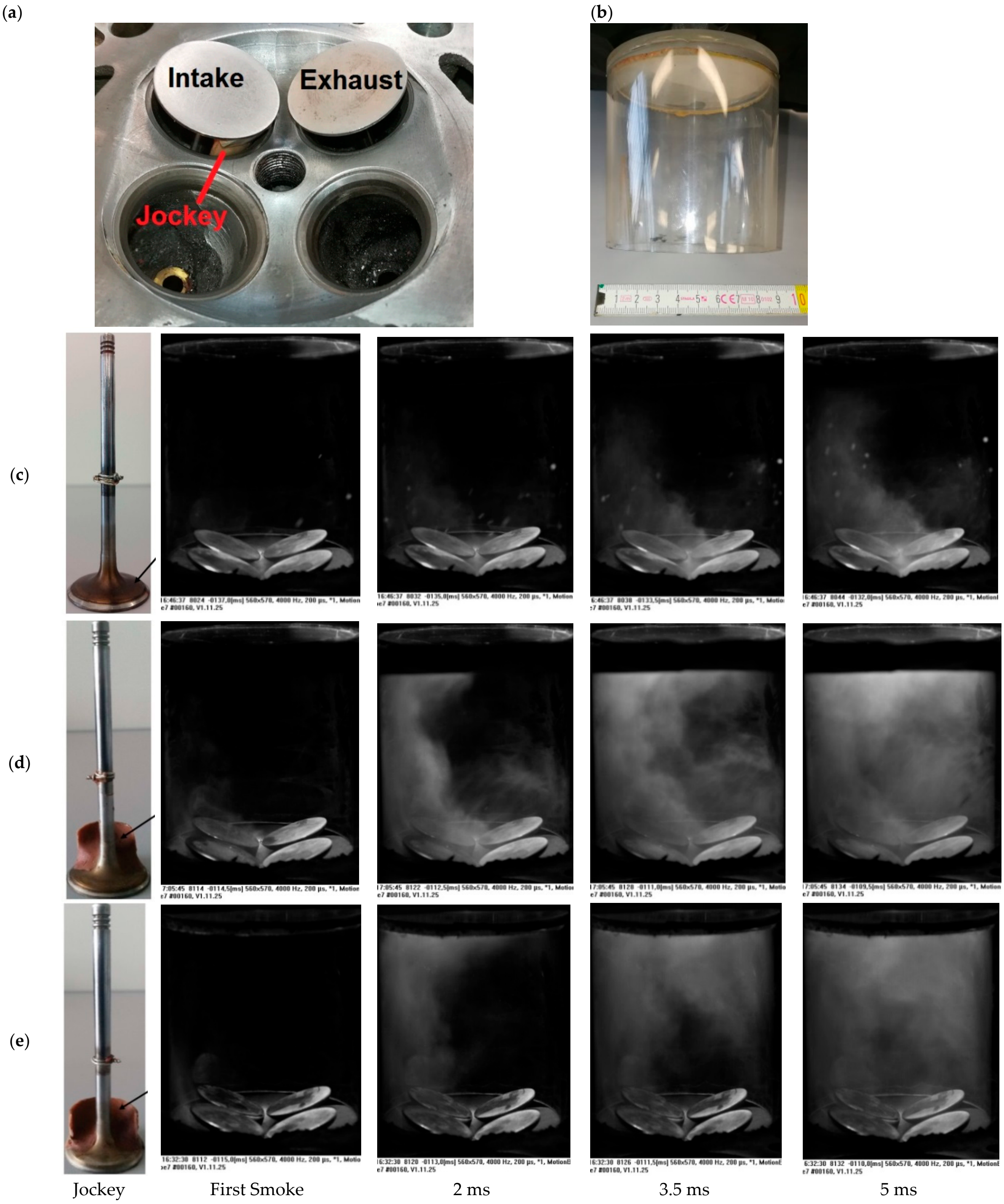


Figure 6. Flow-visualization inside a conventional 500 cm^3 4-valve cylinder (4-stroke gasoline) at 0 ms, 2 ms, 3.5 and 5 ms. (**a**) Cylinder head equipped with one outlet valve and one inlet valve with experimental jockey structure. Characteristic valve stroke is 8 mm. (**b**) Transparent perspex cylinder, scale in cm. (**c**) Unchanged and original inlet valves (left pair); poor scavenging with high loss. (**d**) α = 120° and (**e**) α = 180°. The redirected flow much better approaches cylinder volume within 5 ms. Direct shortcut into the exhaust is entirely suppressed in (**e**) and very effective in (**c**). The charging occurred with 15 kPa, which is less than calculated in the simulations in Figure 5.

The propagating smoke was recorded with a high-speed camera (0.2 ms time resolution). The inlet valves were partly equipped with a jockey structure (Figure 2c) formed from sealing putty and sufficient for this cold experiment without significant forces.

As is clearly visible in the high-speed photographs in Figure 6c, the original inlet valves without jockeys provide almost no gas exchange in the cylinder within 5 ms, and G is poor. The smoky "fresh" gas effectively exits through the nearby exhaust and is hardly apparent in the cylinder's volume, related to Figure 5a.

Scavenging is improved with jockeys on the inlet valves (α = 120° for Figure 6d and 180° for Figure 6e). As presumed from the simulations (Figure 5c,d), the smoke performs a reverted tumbling (more distinct in Figure 6e with α = 180°) and within 5 ms, approaches a significant portion of the cylinder's capacity, although generally driven with just 15 kPa (150 mbar). However, the volumetric flow metering for the most obstructed case (Figure 6e) indicated less than 500 cm^3: with 15 kPa, just 380 cm^3 were replaced in 5 ms. The theoretically required charging power for such a single cylinder in 2-stroke operation at 3000 rpm and with 380 cm^3 scavenging can be calculated to 0.5 kW.

We use the data from the most obstructed case in Figure 6e for determining the referential W_0: at φ = 180° (equivalent to 10 ms at 3000 rpm) the provided 380 cm^3 within 5 ms is more than generous for the remaining 250 cm^3 and just 1/3 of the speed is required and then 1/9 of the charging pressure is sufficient, i.e., 1.7 kPa. The required scavenging energy becomes 0.45 J, to be doubled to 0.9 J due to the valves' reduced time-cross-section (Figure 1b). The referential W_0 at 3000 rpm is then 0.9 J/300 J = 0.003. This number points to the light green traces in Figures 3 and 4. Thus, even at 3000 rpm, increased torque and increased efficiency can be expected for φ = 80 to 100°.

Remarkably, the derived values for W_0 from the simulation and the experiment are very similar: The simulated data (W_0 = 0.002) is calculated at 2500 rpm, α = 120°, and only a single intake valve. The experiment (W_0 = 0.003) is calculated at 3000 rpm, a more obstructing α = 180°, and two intake valves.

The circumferential speed of the practical tumble in Figure 6e can be estimated to 8 cm/2 ms = 40 m/s. Similar results are obtained when tracing the multiple high-speed photographs in more detail. This is far less than the 175 m/s in Figure 5d with the same α; the pressure drop (or applied pressures) in Figure 5d is several times higher, and the speedy flow reaches the outlet after a U-turn in distinctively less than 2 ms. In Figure 6e, the first smoke arrives at the outlet after 4 ms.

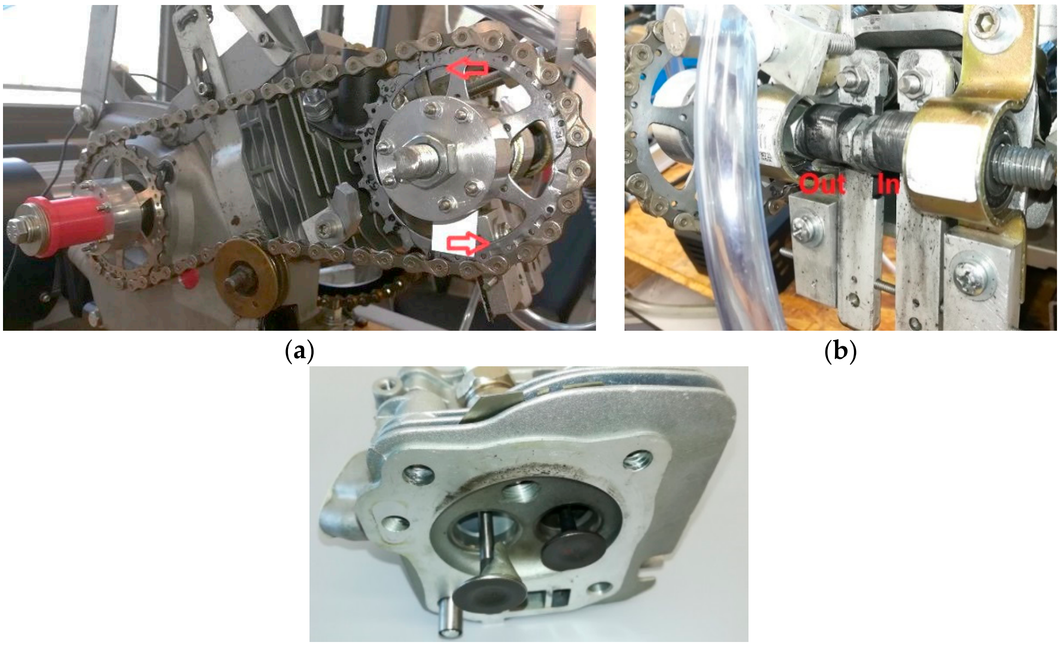
Nevertheless, all three of these more practical scenarios appear to correlate to the simulations in Figure 5a,c,d even though four valves are present.

6. Experimental Engine

The purpose of this section is the demonstration of the 2-stroke engine's principle feasibility and the jockey's principal benefit. A well-engineered and mature engine cannot be realized in this stadium and within our limited capabilities. In addition, the applied measurement (self-construction) for torque, and particularly fuel consumption, is not very accurate.

A small (196 cm^3), single-cylinder, and simple 4-stroke engine was modified in our laboratory (Figure 7). Most changes apply to the self-made valve train and camshaft which now has to rotate at double speed (2-stroke). The valve overlap must be much higher than in 4-stroke applications and approaches 80%. The valves' opening angle φ is set to 120°. The adjacent jockey on the (here) non-rotating inlet valve was formed from a heat resistant (200 °C) epoxy sealant.

Direct fuel injection would be very desirable for the concept but was not feasible for this small engine and our limited capabilities. Instead, a metered propane flow was continuously injected into the inlet channel. A gradually adjustable and mild charging (max. 30 kPa or 0.3 bar) provided the fresh air.



(**a**) (**b**)

(**c**)

Figure 7. (**a**) 1:1 camshaft drive for the 2-stroke. (**b**) Camshaft with highly overlapping lobes. (**c**) Head with elevated valves and a jockey made from heat resistant epoxy attached to inlet valve.

The stable idle speed of this small 2-stroke engine is as low as 400 rpm. At 1000 rpm, a torque of more than 15 Nm is already available, which is equivalent to 1.65 kW. The original, and much more mature, 4-stroke design of this engine delivers just 13 Nm at a higher speed (>2000 rpm).

High speed was not approached with the demonstrator; the valve train appears too delicate. At 1.65 kW output (1000 rpm), the metered propane accounts for about 7 kW worth of combustion heat and an overall efficiency of 23% is the result, which, however, appears quite optimistic for this small engine, even without direct injection. The potentially largest accounting inaccuracy is in the propane metering.

As to be expected (Figures 5 and 6), the modified engine performs much worse without the jockey—at least two times as much fresh gas, including stoichiometric amounts of propane, have to be provided to achieve only 8 Nm of torque at 1000 rpm. Furthermore, with the applied jockey, the engine runs at various valve angles even with zero charge pressure (blower off), filled by the momentum of the exhaust gases. A self-sustaining run without the jockey was hardly possible.

7. Summary and Conclusions

In theory, the experiment, and as already claimed in old patents [17], a suitable jockey on the inlet valve considerably reshapes the inlet flow and thereby allows an advantageous gas exchange in 2-stroke engines built on the basis of conventional, well-established, and minimally-modified 4-stroke cylinder heads. The flow can be realized with even zero initial shortcuts (Figure 6e) and would provide some additional cooling for the piston. A fixed streaming guide inside the inlet channel is inferior to the jockey (Figure 5f,g). As the required charging pressures—indicated by W_0—are still small enough, the jockey promises efficient and high-torque 2-stroke engines at low- to medium-rotational speeds, pointing to heavy duty applications.

The relatively high speed of the valves (doubled frequency) is, to a large portion, compensated for by the addressed lower rotational speed. The concept presumably applies for all kinds of fuels when directly injected (Figure 5e).

The jockey itself, as the only new part (all other components are well-known and in mass-production), is not endangered by heat or distinct sealing requirements, and a light and balanced solution (Figure 2) or shaped sheet may be sufficient.

A variable camshaft phasing or valve control, also in mass-production, appears particularly attractive for optimized operation over extended engine loads and speeds. Valves' timing (Figure 1) could be tuned for turbochargers.

As previously suggested by other workers [9,10], the engine could be modified to be switched between 2-stroke (for low rpm, more torque and efficiency) and 4-stroke (for high rpm, full load). In 4-stroke operation, the jockey could be halted in an upper position which would then not distinctively obstruct the valve's annular inlet gap.

Important for long-lasting and heavy duty machines and different to 4-strokes, the piston pin in a 2-stroke design bears uninterrupted and unidirectional pressures and this may cause problems with the lubrication film. This well-known problem is sometimes countered with roller bearings.

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