Intake characteristics and pumping loss in the intake stroke of a novel small scale opposed rotary piston engine

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Abstract:

Compact and high power density internal combustion engines are attracting much attention for the applications to hybrid vehicles, aiming at decreasing fuel consumption and exhaust emissions. A novel opposed rotary piston engine, whose cyclic period is 360° crank angle, is designed as the power of hybrid vehicles. Intake process of the internal combustion engines significantly affects brake thermal efficiency and rated power. In this paper, 3D simulation of a opposed rotary piston engine is conducted over different engine speeds of 3000 and 5000 RPM, in order to analyze the intake characteristics and pumping loss in the intake stroke. The results indicate that the in-cylinder pressure distributions change significantly in the intake process, which results from the variations of cylinder volume and fresh air flow rates. The minimum in-cylinder pressure is approximately 0.3 and 0.2 bar for 3000 and 5000 RPM, respectively, with the combustion chamber volume being ~ 0.04 L correspondingly. The maximum velocity of the fresh air in the intake process is higher than 150 m/s. Intake ports 2 and 3 dominate the mass flow of the combustion chambers, in addition, the contribution of mass flow from intake port 1 decreases with the engine speed. The pumping loss in the intake process increases from 1.35 to 4.39 kW when the engine speed increases from 3000 to 5000 RPM.

Keywords: opposed rotary piston engine; intake characteristics; in-cylinder pressure distributions; pumping loss

1. Introduction

Under the pressure of fossil fuel consumption and environmental pollutions caused by conventional vehicles, lots of measures are taken to increase the energy efficiency and decrease the emissions, from the viewpoints of fuel types, combustions, energy recycle, thermal management, and after-treatments (Kovacs et al., 2019; Kuszewski, 2019; Sharma et al., 2019; Uusitalo et al., 2016). Such as, the carbon monoxide decreased by 14 % if 10% volume of *Jatropha curcas* blends with 90% diesel fuel, meantime, the hydrocarbon decreases by 16% (Mofijur Rahman et al., 2014). Additionally, much attention has been paid to hybrid and electric vehicles due to their low emission factors and high energy efficiency (García et al., 2019; Liu et al., 2019; Luin et al., 2019; Zhang et al., 2019). However, the technologies of electric vehicles need to be further improved to meet customers' requirements, especially for the energy density of batteries, charging duration, and low temperature performance (Fischer et al., 2009; Hamut et al., 2014; Kwade et al., 2018; Schmuck et

al., 2018). Hybrid vehicle technologies are more mature than electric vehicles, but they require compact, high power density, and high thermal efficiency engines to be the energy sources. This gives the opportunities of the development of other forms of internal combustion engines, such as the Wankel engines (Shi et al., 2019a) which have a high power density and are free of the complicate crank connecting rod system. The cyclic period of Wankel engines is 360° crank angle (CA), which is half of the conventional reciprocating engines, contributing to a high power density (Su et al., 2018). Injection stratified charging combustion was investigated by Chen et al. based on a nature gas-diesel Wankel engine using a 3D simulation tool (Chen et al., 2019). In the fuel-air mixing process, the natural gas movement is sensitive to the airflow. Meantime, the flame propagation speed increases with the nature gas blending ratio, which conduces to increasing brake thermal efficiency. However, the emissions from Wankel engines are great issues due to incomplete combustion in cylinders. The combustion chambers of Wankel engines are long and narrow, which leads to a serious quenching effect, resulting of much unburnt fuel being emitted into the atmosphere (Yang et al., 2018). The commonly used approaches to decrease the quenching effect are focused on increasing the flame propagation speed and decreasing the quenching distance (Shi et al., 2019b), which can be achieved by hydrogen addition (Fedyanov et al., 2019; Ozcanli et al., 2018). In addition, hydrogen addition increases the engine brake thermal efficiency by increasing the flame propagation and enhancing combustion completeness, such that the CA0-10 and CA10-90 were significantly shortened (Ji et al., 2016). Also it was indicated by Amrouche et al., such that the brake specific energy consumption decreases from 21.5 MJ/(k·Wh) to 14.5 MJ/(k·Wh) when the hydrogen energy addition increases from 0% to 10% (Amrouche et al., 2018). The coefficient of variance in indicated mean effective pressure also drops from 13% to 1.5%. Due to high flame propagation speed, the peak combustion temperature increases, which leads to higher nitrogen oxides (NO_x) emission. The cold start issues of Wankel engines needs to be addressed because of high specific surface area of combustion chambers. Xu et al. decreased the exhaust emissions of a Wankel engine during cold start by controlling the excess air ratio (Xu et al., 2018). Meanwhile, the high specific surface area translates in a thermodynamic disadvantage that inherently leads to a lower efficiency than a traditional reciprocating engine (Boretta et al., 2015).

Similar to Wankel engines, opposed rotary piston (ORP) engines, being a novel type of piston engines, are free of complicated crank connecting rod devices. Meantime, narrow and long combustion chambers are avoided, which is better than Wankel engines. The performance of a ORP

engine with variable geometry compression ratio was analyzed using thermodynamic numerical simulation method in the investigation (Ticona et al., 2015). In addition, Miller cycle was applied to the engine to achieve a high thermal efficiency, however, it had a penalty of power density. Variable compression ratio in ORP engines were also investigated in the work (Guarato et al., 2016), such that high compression ratio effectively increases the fuel economy. The ORP engines, with changeable compression ratio, have a high flexibility of fuel types, since different types of fuel have different knock tendency and auto-ignition temperature. Zou et al. investigated the performance of a ORP engine using both simulation and experiment approaches (Zou et al., 2014). The tested maximum in-cylinder pressure of a prototype, with naturally aspirated intake systems, is approximately 43 bar over 2000 RPM and 1.2 excess air coefficient, and the power density reaches 77.8 kW/L. In the prototype, the end faces of the pistons are flat, which leads to a small range of spark timing, which provides a small potential of improving brake thermal efficiency and decreasing exhaust emissions by adjusting sparking timing under different operating conditions (Duan et al., 2019; Li et al., 2019; Zhu et al., 2018). Other researches of the ORP engines, such as the work (Deng et al., 2013) investigated the piston velocity and acceleration in the engine operating process using a mathematical method, and the authors (Deng et al., 2013) hold the opinion that the power density of the ORP engine increases with piston numbers under the conditions of same engine size. However, this opinion neglects the effect of intake duration on charging efficiency which dominates the engine power output. Since more pistons (means higher engine displacement) lead to a shorter duration of intake and exhaust process, resulting of a low charging efficiency. As shown in the work (Heywood, 1988), the charging efficiency of the engine decreases significantly when the intake and exhaust durations (related to engine speed) are lower than a threshold. Liu et al. applied the ORP structure to an air compressor by changing the intake and exhaust ports timing, with the maximum in-cylinder pressure reaching 13 bar (Liu et al., 2015). A similar type of engine named centric reciprocating engine (CRE) (Otechos) was designed, such that a pair of rotors move back and forth, which is different from Wankel and conventional reciprocating engines. The power output of the engine with a displacement of 1.15 L is approximately 200 house power, which is corresponding to a 2.3 L conventional reciprocating engine.

The cyclic period of the ORP engine is 360° CA, and each stroke (intake, compression, power, and exhaust) lasts $\sim 90^\circ$ CA. The durations (in time series) of the intake and exhaust process in ORP

engines are much shorter than conventional reciprocating engines, which may lead to a high pumping loss despite the effective surface area of the intake and exhaust ports are huge. As shown in the work (Gao et al., 2019), the layout of three intake ports and a big exhaust port is adopted to increase the charging efficiency. Gao et al. explored the performance of a ORP engine over various engine speeds, using 1D and 3D simulation approaches (Gao et al., 2019). In that work, the merits of the ORP engine were analyzed based on P-V diagrams, such that it is with a high power density, smooth rotation, a low noisy level, and free of crank connecting rod system. Further, the power density of the ORP engine was evaluated using GT-suite software under naturally aspirated intake systems. The power density reaches 80 kW/L, which is almost twice of a conventional reciprocating engine with a same displacement. In the ORP engine (Gao et al., 2019), the exhaust port closes slightly before top dead center (TDC), and intake ports open slightly after TDC, which causes more pumping loss in intake and exhaust strokes. The work conducted by Gao et al. is mainly focused on the evaluation of the overall performance and comparison with a conventional reciprocating piston engine (Gao et al., 2019). However, the intake characteristics and pumping loss are neglected although they are different from conventional reciprocating engines. The intake stroke has a significant impact on brake thermal efficiency and power output, as indicated in the work (Ciulli, 1993). Based on the above discussion, the intake characteristics and pumping loss in the intake stroke of ORP engines are limited to date, although they are significantly important to brake thermal efficiency and fuel consumption of the engine.

The investigation in this paper is mainly focused on the intake characteristics and pumping loss of a ORP engine in the intake stroke by using a 3D simulation method. This paper is organized as the follows: (1) the structures and operation theory of the ORP engine are described, and compared with conventional reciprocating engines; (2) the dynamic mesh, calculation stability and convergence of the 3D simulation are researched to ensure a high precision of simulation results; (3) the pressure distributions and streamlines in combustion chambers during intake are researched over different engine speeds (3000 and 5000 RPM); (4) the mass flow rates of the intake ports under different engine speeds are analyzed; (5) the pumping loss of the ORP engines caused by the intake process are explored by analyzing P-V diagrams.

2. Materials and methods

The flowchart of the investigation in this paper is shown in Figure 1. In this part, the detailed

structures and operation process of the ORP engine are introduced firstly; then, the theory of 3D simulation is introduced.

Figure 1 Flowchart of the investigation

2.1 Structures and operation process of opposed rotary piston engines

The basic parameters of the ORP engines are shown in Table 1, they are also listed in the authors' previous work (Gao et al., 2019). This novel engine is a four cylinders, four stroke, naturally aspirated spark ignition engine, with 3 intake ports, 1 exhaust port, and 2 spark plugs. The operating process includes intake stroke, compression stroke, power stroke, and exhaust stroke. The compression ratio is the same level with the conventional spark ignition engines. Currently, it is a naturally aspirated engine, however, the authors hold the opinion that it is more suitable for turbocharging since there is not early opening of the intake ports nor late closing of the exhaust port. The intake and exhaust systems are different from conventional reciprocating engines, whose exhaust and intake valve timing has overlaps in order to increase the charging efficiency and brake thermal efficiency (Parvate-Patil et al., 2004).

Table 1 Specifications of the opposed rotary piston engine

Specifications	Values
Ignition method	Spark ignition
Stroke number	4
Cylinder number	4
Spark plug number	2
Compression ratio	9.73
Displacement/ L	0.5468

Intake system	Naturally aspirated
Fuel type	Gasoline
Bore/ mm	59.5
Chord length of the piston end face/ mm	59.0
Geometric angle of both end faces in piston/ °	56
Crank angle per stroke/ °	90
Intake port number	3
Exhaust port number	1

The ORP engine is composed of four cylinders, two engine blocks, two shafts and two spark plugs, as shown in Figure 1. Pistons 1 and 3 are connected using shaft 1, while pistons 2 and 4 are connected using shaft 2. The geometrical angle between the centers of pistons 1 and 3 is 180° CA, the angle is the same for pistons 2 and 4. The combustion chamber of the engine is consisted of inner surfaces of the engine blocks, surfaces of both shafts, and the end surfaces of adjacent pistons. The three intake ports are located at the engine blocks separately, which contributes to form a strong intensity flow in the intake stroke although it can lead to slightly high pumping loss. The particular piston-shaft structure makes each stroke last 90° CA in a cyclic period (360° CA), meantime it ensures 120° geometrical angle for each stroke, which is achieved by two shafts respectively connecting with each two opposed pistons, as shown in the authors' previous work (Gao et al., 2019).

(a) Engine block

(b) Pistons and shafts connections

(c) Detached figure

(d) Assembly of engine parts

Figure 1 Structures of the ORP engine

Figure 1 Structures of the opposed rotary piston engine

The top dead centre (TDC) is the positions where the combustion chamber has the minimum volume; bottom dead centre (BDC) is corresponding to maximum combustion chamber volume, which is the same definitions as the conventional engines. However, the crank angle in ORP engines is defined by the centre positions of combustion chambers rather than piston positions since each combustion chamber is related to adjacent pistons. The fluid domains are shown in Figure 2, where two combustion chambers are at TDC positions. As shown in the figure, the bowls (indicated in Figure 1(d)) in pistons significantly enlarge the range of spark timing, which allows to optimize in-cylinder combustion in order to increase brake thermal efficiency and decrease exhaust emissions under various engine conditions. Different from conventional spark ignition engines, two spark plugs are used in ORP engines, which ensures the sparking stability and contributes to decrease the cycle-to-cycle variations, especially when a large exhaust gas recirculation (EGR) rate is adopted. Meantime, it has potentials of decreasing knock tendency, being benefit from the decreased flame propagation distances, which is achieved by opposed positions of the two spark plugs. The bowls also strength the in-cylinder flow intensity, conducting to improve combustion efficiency.

Figure 2 Fluid domains of opposed rotary piston engine

The movements of the two shafts are in sine mode, as shown in the authors' previous work (Gao et al., 2019). The movements of the dynamic fluid domains are shown in Figure 3. The volume of the combustion chambers 1 and 3 is all the same, since pistons 1 and 3 are connected using shaft 1, while pistons 2 and 4 are connected using shaft 2. The cyclic period of the ORP engine is 360° CA, with two TDCs and BDCs positions. As shown in the figure, the exhaust and intake ports opening time is not overlapped, which will lead to more residual gas in cylinders than conventional reciprocating engines if they have the same displacement. However, the residual gas contributes to internal EGR to decrease NO_x formations. Intake port 1, port 2, and port 3 work successively after pistons pass the exhaust port. While it should be noted that these three intake ports cannot work simultaneously (at most two intake ports at the same time, e.g. intake ports 1 and 2; or intake ports 2 and 3). Video S1 shows how does the engine work.

Pistons and
engine
blocks

Figure 3 Movements of pistons and fluid domains

2.2 3D simulation theory

In this paper, ANSYS FLUENT software is used to simulate the unsteady flow in the ORP engine, such as streamlines and pressure distributions in the combustion chambers. The 3D simulations provide the approach of analyzing the intake characteristic and pumping loss, further, improving engine thermal efficiency and power outputs. ANSYS FLUENT is based on the finite volume method. Reynolds Averaged Navier-Stoke (RANS) turbulence model is the renormalization group

(RNG) k-epsilon model, which improve the accuracy of predicting rapidly strained flows and swirling flows (Wei et al., 2015). The transport equations of k and ε are shown in Equations 1 and 2.

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + P_k + P_b - \rho \varepsilon - Y_M + S_k \quad (1)$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} (P_k + C_{3\varepsilon} P_b) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} + S_\varepsilon \quad (2)$$

t is time; ρ is the working fluid density; k is the turbulent kinetic energy; x_i is a position factor in the i_{th} direction; u_i is a velocity factor in the i_{th} direction; x_j is a position factor in the j_{th} direction; μ is viscosity; μ_t is turbulent viscosity; P_k is the generation of turbulence kinetic energy due to the mean velocity gradients; P_b is the generation of turbulence kinetic energy due to buoyancy; ε is the dissipation; Y_M is the contribution of the fluctuating dilatation in the compressible turbulence to the overall dissipation rate; S_k , S_ε is user-defined source terms; $C_{1\varepsilon} = 1.44$, $C_{2\varepsilon} = 1.92$, $C_{3\varepsilon} = -0.33$, $\sigma_k = 1.393$, $\sigma_\varepsilon = 1.393$.

The turbulent viscosity is defined in Equation 3, and P_k is defined in Equations 4 and 5.

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (3)$$

$$P_k = -\rho \overline{u_i' u_j'} \frac{\partial u_j}{\partial x_i} \quad (4)$$

$$P_k = -u_t S^2 \quad (5)$$

Where, $C_\mu = 0.09$, u_i' is fluctuating component of velocity in i_{th} direction; u_j is velocity factor in the j_{th} direction; u_j' : fluctuating component of velocity in j_{th} direction; S is the modulus of the mean rate-of-strain tensor, it was defined as Equation 6. P_b is defined in Equation 7.

$$S \equiv \sqrt{2S_{ij}S_{ij}} \quad (6)$$

$$P_b \equiv \beta g_i \frac{u_t}{Pr_t} \frac{\partial T}{\partial x_i} \quad (7)$$

Where, β is coefficient of thermal expansion; g_i is the component of the gravitational vector in the i_{th} direction; Pr_t is the turbulent Prandtl number for energy. For the standard and realisable-models, the default value of Pr_t is 0.85.

The coefficient of thermal expansion, β , is shown in Equation 8.

$$\beta \equiv -\frac{1}{\rho} \left(\frac{\partial \rho}{\partial T} \right)_p \quad (8)$$

The governing equation of computational fluid dynamics is defined in Equation 9.

$$\frac{d}{dt} \int \rho \phi dV + \int \rho \phi \left(\vec{u} - \vec{u}_g \right) d\vec{A} = \int \Gamma \nabla \phi d\vec{A} + \int S_\phi dV \quad (9)$$

Where, ϕ is general scalar; dV is the boundary control volume; ρ is the working fluid density; $\vec{u}$ and $\vec{u}_g$ are the flow velocity and the grid velocity of the moving mesh, respectively; $\vec{A}$ is surface area vector; Γ is diffusion coefficient; $\nabla\phi$ is gradient of ϕ ; S_ϕ is the source term.

2.3 3D Computational mesh

High quality mesh is the foundation to ensure the precision of 3D simulation results. Table 2 shows the element numbers of the fluid sub-domains and the overall mesh quality. The minimum orthogonal, maximum ortho skew, and maximum aspect ratio are 0.337, 0.663, and 16.726, respectively. There are 16 bowls and 4 cylinders fluid sub-domains in total, and the mesh numbers of these sub-domains are slightly different individually. Table 2 only lists element numbers in one of them (bowls and cylinders). The hexahedral meshes are applied to all the sub-domains. In the engine operating process, cylinder volume changes greatly, so dynamic mesh is used in the unsteady flow simulation. The cylinder fluid mesh changes in layers during the piston movements.

Table 2 Element numbers of the sub-domain fluids and overall mesh quality

Sub-domain	Elements
Intake pipe 1	12210
Intake pipe 2	12210
Intake pipe 3	16632
Exhaust port	40664
Spark plug (each one)	18202
Bowl (each one)	37718
Cylinder (initial stage)	53376
Minimum orthogonal	0.337
Maximum ortho skew	0.663
Maximum aspect ratio	16.726

2.4 Boundary conditions of 3D simulation

The boundary conditions of 3D simulation are shown in Table 3. As shown in the work (Gao et al., 2019), the charging efficiency of the ORP engine drops significantly when the engine speed is higher than 3000 RPM, meantime, 5000 RPM engine speed is corresponding to the rated speed. So, 3000 and 5000 RPM are chosen to investigate the intake characteristics and pumping loss. The inlet

and outlet pressure are set as 1.0 bar. In the simulation, one cyclic period includes 2000 time steps that the time step is 10^{-6} and 0.6×10^{-6} s respectively for 3000 and 5000 RPM. In this work, combustion sub-model is not taken into consideration, however, a high cylinder wall temperature is set to simulate the combustion situations.

Table 3 Boundary conditions for CFD simulation

Boundary conditions	Values
Inlet pressure/ bar	1.0
Inlet temperature/ K	300
Outlet pressure/ bar	1.0
Outlet temperature/ K	300
Cylinder wall temperature/ K	350
Inlet and outlet wall temperature/ K	300
Roughness height of piston and cylinder surface/ m	10^{-6}
Rotation speed/ RPM	3000, 5000
Working fluid	Fresh air
Load/ %	100

2.5 Stability and convergence of CFD calculations

Mesh number and converge residuals have a significant impact on the precision of simulation results. Table 4 shows the effect of the mesh number, time step, and converge residuals on the simulation results. The boundary conditions are the same as 3000 RPM scenario in Table 3. The maximum difference of the average in-cylinder pressures is 10 Pa for the five given scenarios. The converge residuals listed in the table show a limited influence on the simulation results. In this work, grid number 779324, time step 10^{-5} s, and residual 10^{-6} are chosen. The simulation is considered to be converged when the residual is lower than 10^{-6} . Table 5 provides the details of solver settings in FLUENT software.

Table 4 The simulation results among different scenarios: 3000 RPM

Mesh number	Residuals	Time step/ s	Average in-cylinder pressure at the end of intake process/ Pa
593246	10^{-6}	10^{-5}	99599
779324	5×10^{-6}	10^{-5}	99595

	10^{-6}	2×10^{-5}	99598
	10^{-6}	10^{-5}	99593
	10^{-6}	0.5×10^{-5}	99589
	0.3×10^{-6}	10^{-5}	99591
102672	10^{-6}	10^{-5}	99597

Table 5 Solver settings in FLUENT software

Specifications		Descriptions
Pressure-velocity coupling		Simple
Spatial discretization	Gradient	Least squares cell based
	Pressure	Standard
	Density	Second order upwind
	Momentum	Second order upwind
	Turbulent kinetic energy	First order upwind
	Turbulent dissipation rate	First order upwind
	Energy	Second order upwind
Transient formulation		First order implicit

3. Results and discussion

The engine power output and brake thermal efficiency significantly depend on the volume profiles of combustion chambers. Combustion chamber volume variations as the series of time steps are shown in Figure 4. Time step is 10^{-6} and 0.6×10^{-6} s for 3000 and 5000 RPM, respectively. The combustion chamber volume variations are similar to conventional reciprocating engines, such that they are in sine mode. The operating process of the ORP engine includes intake stroke, compression stroke, power stroke, and exhaust stroke. However, the duration of each stroke (in crank angle) in conventional engines is more than twice of that in the ORP engine. The combustion chamber volume profile around TDC is flatter than that of conventional reciprocating engines, as shown in the authors' previous work (Gao et al., 2019). It will result of a high percentage of constant volume combustion, which contributes to a high brake thermal efficiency of the ORP engine. The volume profile of combustion chamber 1 is symmetrical with combustion chamber 2, which means that combustion chamber 1 is at TDC when combustion chamber 2 is at BDC.

Figure 4 Combustion chamber volume variations: 3000 RPM

Due to the short duration of each stroke, the heat loss in the combustion stroke will significantly decreased, which conduces to improve the brake thermal efficiency. However, the high power density of the ORP engine will cause a serious thermal load for the engine, so the cooling system may be different from the conventional reciprocating engines. Meanwhile, the sealing issue between the two shaft should be addressed, or it can cause high exhaust emissions due to quenching and crevice effects.

3.1 In-cylinder pressure distributions and streamlines in the intake stroke

Pressure distributions in the combustion chambers and the mass flow rates dominate the pumping loss of intake stroke and charging efficiency to a great extent. The pressure distributions and streamlines in the symmetric plane of the combustion chambers are shown in Figures 5~8. Because the bowls are located at the margins of the pistons, the symmetric plane doesn't cross the bowls. The bowls can be seen in the figures are their surfaces, such that the colour indicates the fluid velocity on the bowl surfaces. The bowl surfaces in the figure are aimed at making the combustion chamber fluid domains more clear. As mentioned in Section 2, the intake and exhaust port structures cause more residual gas in combustion chambers, which leads to a high pressure at the start of the intake stroke (intake port 1). So, a small part of the gas in the combustion chambers will flow back into intake pipe 1. The back flow is more serious over 5000 RPM, and has a longer duration than 3000 RPM scenario, as shown in the figures of pressure distributions and streamlines. At the end of the intake process for intake port 1, the in-cylinder pressure decreases significantly, which causes a

strong fluid flow (e.g. velocity). The mass flow rate will be analyzed in the following part. There is an overlap for intake ports 1 and 2, so the fluid from both intake ports collides, which contributes to the formation of a strong intensity flow and vortexes, resulting of a high combustion rate during the combustion stroke. The fluid velocity is more than 150 m/s for both 3000 and 5000 RPM when intake port 2 opens. The fluid velocity in the ORP engine is much higher than that in conventional reciprocating engines, where the maximum velocity is lower than 100 m/s (Pandey and Roy, 2012). As the fluid flows into the combustion chambers rapidly, the in-cylinder pressure increases and the fluid velocity from intake port 3 decreases slightly. At the end of the intake port 3 closing, the fluid velocity is approximately 80 m/s for 3000 RPM, however, it is still higher than 150 m/s for 5000 RPM. Such that the charging efficiency of the engine can be improved through the layout of the three intake ports. The operating process of individual combustion chamber is interacted by the others for conventional reciprocating engines. It is caused by fluid distributions among different combustion chambers, and pressure waves in intake manifolds. Since two cylinders are in intake stroke simultaneously for a four cylinder engine (Gao et al., 2019), although they are not synchronous. The pressure waves in the exhaust system will also be different from the conventional reciprocating engines, such that double-channel turbines may not be suitable for the ORP engines.

Figure 5 Pressure distributions in the intake stroke over time steps for 3000 RPM

Figure 6 Streamlines in the intake stroke over time steps for 3000 RPM

Figure 7 Pressure distributions in the intake stroke over time steps for 5000 RPM

Figure 8 Streamlines in the intake process over time steps for 5000 RPM

Charging efficiency of the ORP engine is approximately 96% and 82% for 3000 and 5000 RPM, respectively, as shown in the authors' previous work (Gao et al., 2019). The charging efficiency of the ORP engine is slightly higher than that of conventional reciprocating engines at low and medium engine speed range. As shown in the work (Dorić et al., 2011), the charging efficiency of a conventional spark ignition engine is approximately 94% over 3000 RPM, and it decreases to 82% when the engine speed increases to 5000 RPM. Because the intake port area of the ORP engine is bigger than that of conventional reciprocating engines, where the port layout and area are limited by the spark plug and the cylinder cap area. However, the intake ports, exhaust port, and spark plugs are located at different positions of the engine blocks. Meanwhile, the separation of the spark plugs and exhaust port effectively decreases the hot points caused by the exhaust port in the combustion process. It decreases the knock tendency, since hot points are one of the main reasons resulting of engine knock (Zhen et al., 2012). Less hot points and shorter flame propagation distance of the ORP engine drop the possibility of engine knock, which is helpful for increasing compression ratio to improve the engine thermal efficiency.

3.2 Mass flow rate analysis in the intake stroke

Fluid mass flow rate in the intake stroke affects the engine charging efficiency and power output. In this part, fluid mass flow rate of the intake ports are analyzed. In the commonly used methods, the mass flow rates in the intake stroke are presented as the series of crank angle, however, the authors make it as the function of time steps in this paper, which helps reader to estimate the mass flow level in the whole intake stroke. Figure 9 shows the mass flow rates of intake ports. As for the intake port 1, the mass flow rates are consistent with in-cylinder pressure distributions and streamlines. The contribution of the intake port 1 to the cylinder fluid mass flow is smaller than the other two, but it ensures an overall high fluid mass flow rate of the intake ports when piston goes through the position of intake port 2. The mass flow rate of intake port 2 is the highest that the value

almost reaches 0.13 kg/s. The mass flow rate almost linearly decreases with time steps after the maximum value. When the mass flow rate of intake port 2 reaches the maximum value, the intake port 3 opens gradually. The fluid flow intensity decreases slightly for intake port 3, however, the opening of port 3 lasts a longer duration (in both geometrical angle and time) than intake ports 1 and 2.

Figure 9 Mass flow rates of intake ports over 3000 RPM

The fluid mass flow rates of combustion chambers in the intake stroke over 3000 RPM are shown in Figure 10. As can be seen, the mass flow rate profiles of combustion chambers present multi-peaks in the intake process, which results from the joint actions of the three intake ports. The overall tendency of the mass flow rates in the four combustion chambers are the same despite of little differences. The maximum flow rate of the combustion chambers is approximately 0.14 kg/s, which is higher than any of the maximum mass flow rates in individual intake ports. In the research (Kesgin, 2005), the mass flow rate profiles in the intake stroke are in sine mode for a conventional reciprocating engine, whose bore and stroke are 190 and 220 mm (approximately 6.23 L displacement), and the maximum fluid mass flow rate is 1.2 kg/s. Similar to the authors' study, at the beginning of the intake process, backflow phenomenon happens. Also, the maximum flow rate is 0.014 kg/s in the intake process for a spark ignition engine with 0.35 L displacement (Kumar and Nagarajan, 2012).

Figure 10 Fluid mass flow rates of combustion chambers over 3000 RPM

Engine speed has a significant impact on the in-cylinder mass flow, which is mainly caused by the formation of low pressure zones. High engine speeds will cause big volume change rates so that fluid cannot flow into the cylinders in a short time. Figure 11 shows the mass flow rates of the three intake ports over 5000 RPM. At the beginning of the intake process for intake port 1, the fluid backflow intensity under 5000 RPM is much stronger than that of 3000 RPM, which causes more residual gas in the combustion chambers to flow into the intake pipe 1. Intake port 1 makes a quite small contribution to the cylinder charging efficiency over high engine speed conditions. However, it helps a lot under low speed situations, as shown in reference (Gao et al., 2019). The mass flow rate profiles of intake ports 2 and 3 change significantly when the engine speed increases from 3000 to 5000 RPM. The fluid velocity reaches to the maximum value in a short time (short crank angle) in the process of intake port 2 opening, and the velocity decreases slowly when it closes. Additionally, the maximum flow rate of intake port 2 under 5000 RPM is higher than the value in 3000 RPM. The phenomenon is much more obvious for intake port 3 such that the maximum mass flow rate lasts approximately 300 time steps, which leads to more contribution of intake port 3 to the cylinder mass flow. As can be seen, the mass flow of intake ports 2 and 3 are almost the same for 3000 RPM, however, it is much higher for intake port 3 than that of intake port 2 under 5000 RPM conditions. The mass flow in the intake ports of conventional reciprocating engines was also investigated in the work (Adomeit et al., 2011), meantime, the flow stability was also analyzed. Figure 12 shows the fluid mass flow rates of combustion chambers over 5000 RPM. The maximum mass flow rate reaches 0.21 kg/s, which is 50% higher than the value in 3000 RPM. The average mass flow rate for a reciprocating engine, whose displacement is 0.05 L, is approximately 12.5 kg/h under 5000 RPM conditions (Bozza et al., 2008).

Figure 11 Mass flow rates of intake ports over 5000 RPM

Figure 12 Fluid mass flow rates of combustion chambers over 5000 RPM

3.3 Analysis of pumping loss in the intake stroke

In this paper, in-cylinder combustion is not considered, only a high cylinder wall temperature is set. Figures 13 and 14 show the P-V diagram in one of the engine cylinders, under 3000 and 5000 RPM, respectively. As can be seen, the P-V lines in the compression and expansion strokes are not overlapped such that the P-V line in the compression stroke is above the one in the expansion stroke. In the compression stroke, the in-cylinder temperature increases rapidly, and it will be higher than

the values of cylinder walls, which leads to heat loss through cylinder walls, with the results of a slightly low in-cylinder pressure in the expansion stroke.

Figure 13 P-V diagram of combustion chamber 1 over 3000 RPM without combustion

Figure 14 P-V diagram of combustion chamber 1 over 5000 RPM without combustion

The pumping loss in the intake process significantly influences the brake thermal efficiency and engine power outputs. As shown in the research (Galindo et al., 2010) that the brake thermal efficiency of the engine increases with the drop of pumping mean effective pressure (PMEP), which is demonstrated based on a two-stage turbocharged architecture. Figure 15 shows the P-V diagram of the intake process, and P-V diagram is a visualized approach to show the level of pumping loss. As the exhaust port closes slightly before TDC ($\sim 2^\circ\text{CA}$), which causes the in-cylinder pressure to be higher than the intake pipe at the beginning of the intake stroke, leading to more pumping loss and low power output. As shown in the work (Kesgin, 2005), residual gas in combustion chambers will cause net work loss. The average minimum pressure in the intake stroke is approximately 0.3 bar, which happens in the process of intake port 2 operating. Then, the pressure increases gradually

due to fluid flowing into the combustion chambers in a high velocity, such that the in-cylinder pressure almost reaches 1.0 bar. Gheorghiu et al. analyzed the pressure variations of a conventional reciprocating engine in the intake process, where the in-cylinder pressure is in the range of 0.87~1.02 bar under 2498 RPM conditions (Gheorghiu, 2001). Luo et al. also investigated the pumping loss of a reciprocating piston engine, such that the minimum pressure in the combustion chamber is 0.62 bar over 4500 RPM (Luo et al., 2003), which is higher than the ORP engine. Since the ORP engine has a small crank angle (a short time) in the intake stroke, it leads to fast changes of combustion chamber volumes. While the maximum pressure of is 0.89 bar for the engine (Luo et al., 2003). When the combustion chamber volume of the ORP engine is huger than 0.08 L, the in-cylinder pressure is higher than 0.7 bar. At that position, there is already large amounts of fresh air in the combustion chambers. There is a little difference of the in-cylinder pressure for the four combustion chambers, since the impact of small errors in the shaft rotation profiles will be enlarged at TDC (due to a small combustion chamber volume). Majority of the pumping loss is generated before the combustion chamber volume 0.075 L. However, for the turbocharged engines, the intake pressure of the engine is higher than the exhaust pressure at parts of the engine operating points (Kesgin, 2005), which will lead to a negative pumping loss of the engine. The authors think turbocharging technology should be applied to the ORP engine, which will be researched in the future work.

Figure 15 P-V diagram of the intake process over 3000 RPM

The in-cylinder pressure is related to the volume changes and mass flow rates of the combustion chambers. Figure 16 shows the combustion chamber volume change rates over 3000 RPM. The volume change rate is the highest when the volume is in the range of 0.065~0.11 L, however, the in-cylinder pressure is the highest. It means the fluid mass flow rate of combustion chambers dominates the pressure in P-V diagram. The volume change rates are in medium level around 0.04 L, where cylinders experience the lowest pressure.

Figure 16 Combustion chamber volume change rates over 3000 RPM

In-cylinder pressures in the intake stroke greatly depend on engine speeds. Figure 17 shows the P-V diagram in the intake process over 5000 RPM. The in-cylinder pressures under 5000 RPM are completely different from those of 3000 RPM. The in-cylinder pressure at the beginning of the intake process is more than 1.5 bar, and decreases rapidly due to the expansion of combustion chambers. The minimum in-cylinder pressure is approximately 0.2 bar, being corresponding to 0.035 L combustion chamber volume, such that the pressure is lower than that of 3000 RPM. Then, the in-cylinder pressure increases gradually. The in-cylinder pressure changes slightly after it increases to 0.73 bar, where the combustion chamber volume is approximately 0.1 L. Low in-cylinder pressure at the end of the intake stroke means a low charging efficiency, further, a small engine power outputs. The pumping loss is more than half of the overall value when the combustion chamber volume is smaller than 0.075 L. The pumping loss of the engines can be optimized by adjusting the intake valve timing, as demonstrated in the work (Hara et al., 1985). Figure 18 shows the combustion chamber volume change rates over 5000 RPM.

Figure 17 P-V diagram of intake process over 5000 RPM

Figure 18 Combustion chamber volume change rates over 5000 RPM

Figure 19 shows the pumping loss in the intake stroke in a cyclic period over different rotation speeds. The average power loss in the intake stroke is shown in Table 6. The power loss over 5000 RPM is 4.39 kW, which is more than 3 times higher than that of 3000 RPM (1.35 kW). It is caused by a low in-cylinder pressure of 5000 RPM, with less fresh air flowing into the cylinders immediately due to short time. It means the pumping loss in the intake stroke is approximately 10% of the rated power, which is ~ 40 kW as shown in the work (Gao et al., 2019). The pumping loss calculated in this work is under full load conditions, while it changes significantly with load variations. The pumping loss was also shown in the work (Ciulli, 1993), it is approximately three times higher for 9.5 bar brake mean effective pressure (BMEP) than that of 0.3 bar BMEP situation.

Figure 19 Pumping loss over combustion chamber volumes in the intake process

Table 6 Average power loss in the intake stroke

	3000 RPM	5000 RPM
Value/ kW	1.35	4.39

3.4 Implications for theory on cleaner production

Hybrid vehicle is an alternative mode during the process of transitions from the internal combustion engine vehicles to electric vehicles, which imposes more requirements for the internal combustion engines in order to fully exhibit the merits of hybrid vehicles. The ORP engine is aimed at being the power of hybrid vehicles, which can effectively decrease the vehicle energy consumption and emissions. Downsized engine is required for the application of hybrid vehicle to decrease the vehicle mass and for the purpose of layout. The power density of the ORP engine is almost twice of the conventional reciprocating engines in theory, which will contribute to decrease the engine size. In addition, the thermal efficiency of the ORP engine is proved to be slightly higher than reciprocating engines. Meanwhile, the ORP engine has less movement parts than reciprocating engines, which effectively decrease the engine cost. High thermal efficiency and small size of the ORP engines will lead the hybrid vehicle to be more sustainable.

4. Conclusions

In this paper, the intake characteristics and power loss of the ORP engine in the intake stroke are analyzed using 3D simulation. Dynamic mesh is applied to simulate the unsteady flow over different scenarios. Two independent rotation profiles are set as the boundary conditions of the two shafts. The contribution of this work clearly addresses the intake characteristics and pumping loss in the intake stroke, also they are compared with published results to dig out the potentials of improving the brake thermal efficiency and decreasing exhaust emissions.

Due to particular structure of the ORP engine, it has two geometrical TDCs and BDCs such that each stroke of the engine is approximately 90° CA, which leads to the differences of the intake characteristics and pumping loss between the ORP engine and conventional reciprocating engines. At the beginning of the intake stroke, slight backflow happens, and its intensity increases with the engine speed, which is mainly caused by the early closing of exhaust port and late opening of intake ports. The particular characteristic can be used for the internal EGR to decrease the NO_x formations.

The fluid velocity at the end of the intake stroke is higher than 150 m/s for 5000 RPM, which implies the possibility of increasing charging efficiency by late closing of intake ports. Intake ports 2 and 3 dominate the mass flow in the intake stroke, while the intake port 1 is mainly the transition process from exhaust stroke to intake stroke. More than half of the pumping loss is generated in the first half of the intake stroke. The average pumping power loss is 1.35 and 4.39 kW respectively for 3000 and 5000 RPM, such that the value is approximately 10% of the engine power output. It poses the necessity of decreasing the pumping loss by re-structure the intake ports.

In the authors' next work, the structure of the intake ports will be optimized, by re-organizing the layout of the three intake ports to retard the closing timing of the intake ports. It is mainly focused on decreasing the pumping loss and increasing the charging efficiency in order to increase the brake thermal efficiency.

Support information

Video S1 shows how does the engine work.

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Abbreviation

BDC: bottom dead centre

BMEP: brake mean effective pressure

CA: crank angle

CA0-10: crank angle duration from spark discharge to 10% heat release of the total fuel (°CA)

CA10-90: crank angle duration from 10% to 90% heat release of the total fuel (°CA)

CRE: centric reciprocating engine

EGR: exhaust gas recirculation

NO_x: nitrogen oxides

ORP: opposed rotary piston

PMEP: pumping mean effective pressure

RANS: Reynolds Averaged Navier-Stokes

RNG: renormalization group

TDC: top dead center

Notation:

$\vec{\rightarrow}_A$: surface area vector

g_i : component of the gravitational vector in the i_{th} direction

i : surface element number

j : time step

k : turbulent kinetic energy

P_b : generation of turbulence kinetic energy due to buoyancy

P_k : generation of turbulence kinetic energy due to the mean velocity gradients

P_{r_t} : turbulent Prandtl number for energy

S : modulus of the mean rate-of-strain tensor

S_k, S_ϵ : user-defined source terms

S_{ij} : rate of strain tensor

S_ϕ : source term.

t : time

T : temperature

$\vec{\rightarrow}_u$: flow velocity

$\vec{\rightarrow}_{u_g}$: grid velocity of the moving mesh

u_i : velocity factor in the i_{th} direction

u_i' : fluctuating component of velocity in i_{th} direction

u_j : velocity factor in the j_{th} direction

u_j' : fluctuating component of velocity in j_{th} direction

V : control volume

x_i : position factor in the i_{th} direction

x_j : position factor in the j_{th} direction

Y_M : contribution of the fluctuating dilatation in the compressible turbulence to the overall dissipation rate

β : coefficient of thermal expansion

ε : dissipation

ρ : working fluid density

μ : viscosity

μ_t : turbulent viscosity

ϕ : general scalar

Γ : diffusion coefficient

$\nabla\phi$: gradient of ϕ

$\sigma_k, \sigma_\varepsilon, C_{1\varepsilon}, C_{2\varepsilon}, C_{3\varepsilon}, C_\mu$: constants

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