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INVESTIGATION INTO THE EFFECT OF BORE/STROKE RATIO ON A SINGLE CYLINDER TWO STROKE OPPOSED PISTON ENGINE

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Abstract

Opposed-piston (OP) engine's promising fuel efficiency has attracted the interest of automotive industry in the recent years. The opposed-piston two-stroke (OP2S) engine technology heightens this fuel efficiency benefit and offers advances in structure, power density and thermal efficiency whilst sustaining its lower cost and weight. Today thermodynamic modelling remains an indispensable and cost effective route in the development and optimisation of internal combustion engines (ICEs). To achieve this goal, the OP2S engine is simulated and validated against experimental results in AVL Boost™, which is hailed as one of the most reliable and advanced engine simulation tools. Detailed analyses of the piston dynamics, heat release, scavenging and heat transfers are highlighted in discrete sections of this paper. Having compared distinct heat release models, the Wiebe 2-Zone model emerged efficacious in replicating the heat release characteristics of the PAMAR™ engine. In comparing the numerical and experimental results, the simulation revealed minimal differences in peak pressure, peak temperature and maximum pressure raise rate, under ±2.5% differences for indicated power, IMEP, indicated thermal efficiency (ITE) and ISFC. Subsequently, confidence taken from the validated numerical model is then deployed to investigate the effect of stroke-to-bore (S/B) ratio on OP2S performance. Three combinations of S/B ratios (0.5, 1.25, and 1.69) with identical swept volume are analysed in this study. Utilisation of the validated model ensured the standardisation of intake, exhaust and the combustion systems in order to isolate the effects of S/B ratio. Results indicate that heat losses decrease with increasing S/B ratio because of the reduced surface area-to-volume in the cylinder. Consequently, an improvement in ITE and mechanical efficiency is observed with reduced ISFC for higher S/B ratios. A tendency of upsurge in combustion efficiency is also evident for higher S/B ratio due to reduced heat transfer near minimum volume of the combustion chamber.

Keywords: *Opposed-Piston, Two-stroke, AVL Boost, Thermodynamic modelling*

1. Introduction

Over the past few decades, the automotive industry has faced unprecedented challenges due to its high-energy consumption and pollutant emissions. With soaring demand for high-efficiency and emissions-compliant powertrains, substantiation of radical advances in reciprocating engine technology has proved indispensable [3, 8, 17]. The opposed-piston two-stroke engine (OP2S) concept can be tracked back to late 1800's. OP2S concept saw a rapid development in the first half of the 20th century across many countries for wide range of applications. However, stringent regulations on emissions halted the growth of two-stroke engines in the latter half despite its issues with emissions. The OP configuration inherits its exceptional engine balance due to asymmetric motion of the pistons [12, 18]. The advantages of opposed-piston engines are described in the book, Opposed Piston Engines: Evolution, Use, and Future Applications [12]: "OP engines evolved because of their ease of manufacture, excellent balance, and competitive performance and fuel efficiency relative to comparable four-stroke engines….With the progressive development of OP engines…other significant advantages also emerged…Among these advantages were cutting-edge specific output, high specific torque, very high power density, and very high power-to-bulk ratio".

The first OP engine, engineered by Wittig, deemed to have appeared for public use in Germany in 1878, although it was only commercialised during the 1900s for land, marine and aviation purposes. Prolific OP engines have typically functioned on a two-stroke cycle, as they were prominent in achieving high thermal efficiency as well as high power density [12]. Fundamental thermal efficiency of OP engine as well as its low emissions, compactness and low manufacturing costs make it a substantial candidate for future commercial and passenger vehicles [14]. In addition, modern computational tools, fuel system advancements (high-pressure common rail) and electronic control technology of ICEs have paved the way for successful modernisation of OP engines [9].

The key purpose of this paper is to gain a comprehensive understanding of the software AVL Boost™ and create a 1D thermodynamic model of the PAMAR™ OP2S engine. Initially, the 1.00L single cylinder engine is introduced, followed by a description of the numerical methodology of the engine simulation. Successively, the model validation is accomplished and simulation results are compared to experimental results of Primer1 engine. Finally, the validated model is utilised to demonstrate the effects of stroke-to-bore ratio on OP2S engines' heat loss, efficiency and fuel consumption.

2. Literature Review

Pinault and Flint [12], clarifies that Wittig's OP engine had a classic three-throw crank with the centre throw linked to the inner piston and operated on a four-stroke cycle. In 1888, Hugo Junkers and Oechelhaeuser revolutionised OP engines with the inclusion of two-stroke operation. This OP gas engine had a mechanical efficiency (ME) of 77% along with a 40% decrease in fuel consumption in comparison to a modern day four-stroke. In the early 1900s, OP engines were developed with a varying degree of success, notably the Doxford engines, being predominant until its decline in 1980s. A common rail injection system developed by Vickers integrated with optimised combustion resulted in a substantial increase in mechanical efficiency (ME) up to 82%.

A comprehensive thermodynamic analysis has been accomplished by Herold et al. [1] titled; "Thermodynamic Benefits of Opposed-Piston Two-Stroke Engines" exemplifies the fundamental efficiency advantage of OP2S over a four-stroke engine with comparable power output and geometry. The simulation results revealed a 10.4% reduction in indicated-specific fuel consumption (ISFC) than the four-stroke engine.

Another intriguing piece of work utilising three-dimensional (3D) computational fluid dynamics (CFD) along with state-of-the-art spray, turbulence and combustion models has been carried out on OP2S engines by Venugopal [17]. The research evaluates the effects of injection pattern design on piston thermal management via combined experimental and analytical approach.

A novel opposed-piston folded-crank train (OPFC) two-stroke range extended engine developed by Beijing Institute of Technology is presented in Zhao et al. [19].

Last of all, Warsaw University of Technology (WUT) is developing a new-type barrel engine called PAMAR-4 under a Polish-Norwegian research programme. This is a successor of PAMAR-3 (3.0L, 340 kW) engine built for aviation purposes, which achieved a test bench efficiency of 44% and has successfully functioned on propane, CNG, gasoline and diesel fuels. The wobble plate mechanisms deployed in this engine yields greater efficiency than the slider-crank crankshaft components. Opaliński et al. [10] refers to a comparison of gas-dynamics of this engine, which has been modelled and quantified using Ricardo WAVE (1D) and Ansys Fluent (3D CFD).

3. Numerical Modelling of Engine

Nowadays, thermodynamic modelling has become an integral process in the development and optimisation of ICEs [13]. It is a broad and fertile research area, considering the variety of analytical techniques present. As a result, laboratories and automotive stakeholders have produced their own thermodynamics models with a degree of complexity, scope and ease to use. Although, most of the processes in the engine are 3D, often 1D simulations are deployed to compromise for complexity and time constraints [2]. When it comes to numerical modelling of ICEs, there are three major 1D-engine simulation computer programs, namely GT Power, AVL Boost and Ricardo WAVE. Unfortunately, none of them has a built-in module capable of simulating an opposed-piston (OP) engine. [4]. However, AVL Boost is the favourite of the aforementioned, due its compatibility of defining relative piston motion profile, thus enabling to model the OP configuration [17]. Furthermore, it provides advanced options in terms of design, cycle simulation and heat release models, compared to other industrial modelling platforms [5]. Although numerous OP2S concepts have been universally modelled and examined by researchers in the recent past only a limited number of journals have exposed experimental data from fully developed engines. The prime focus has been vastly restricted to basic engine performance, as comprehensive analysis of intricate engine operations (in-cylinder gas motion / mixture formation) proves to be challenging [18].

4. Opposed-piston configuration

The engine uses a uni-flow scavenging with an injector placed on its cylinder liner. Gas ports are located at both ends of the cylinder, intake ports to deliver fresh charge into the cylinder from one side, and exhaust ports on the other side to get rid of burnt gas and residuals. Opening and closing of these ports are controlled by the reciprocating motion of pistons. Two pistons are deployed at opposite ends of the cylinder liner, often named as intake and exhaust pistons, respective to their effective control of port opening and closing. The reciprocation movements of piston are driven by what is called an opposed crank-connecting rod mechanism. The reciprocating movement accomplishes air-exchange with combination of the gas ports [7].

5. Numerical methodology

Figure 1 displays a schematic of the input data needed to implement a 1D simulation of OP2S in AVL Boost. Cylinder geometry, port timings, operating and initial conditions are identical to what exists in the actual engine. Piston motion, heat release, heat transfer and scavenging models are aligned accordingly to reproduce the heat release characteristics of PAMAR engine, and are discussed inclusively in the following sections. Factors that influence in-cylinder combustion characteristics like the piston bowl shape, injector technology, rail pressure and EGR capability are not reflected here as these are beyond the scope of this paper.

6. Engine cycle simulation

AVL BOOST version 2014.1 is used in this modelling paper. The main tool used in the software is the Workspace, which is composed of predefined elements that represent components of ICE. The 1D engine model is built within the workspace, selecting various elements from the element tree and is joined by pipe attachments to their desired connectors. The numerical model created to represent PAMAR single cylinder OP2S engine is shown in Fig. 2.

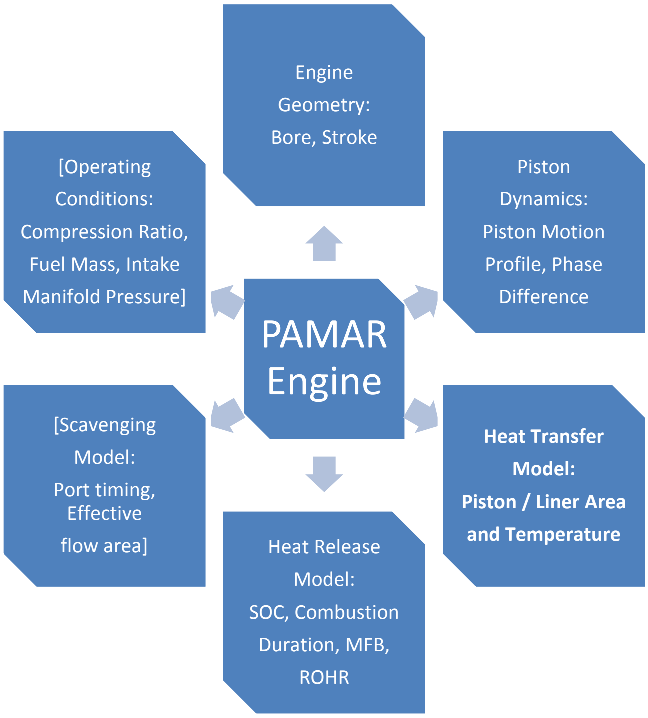

Fig. 1. Numerical model input data flow

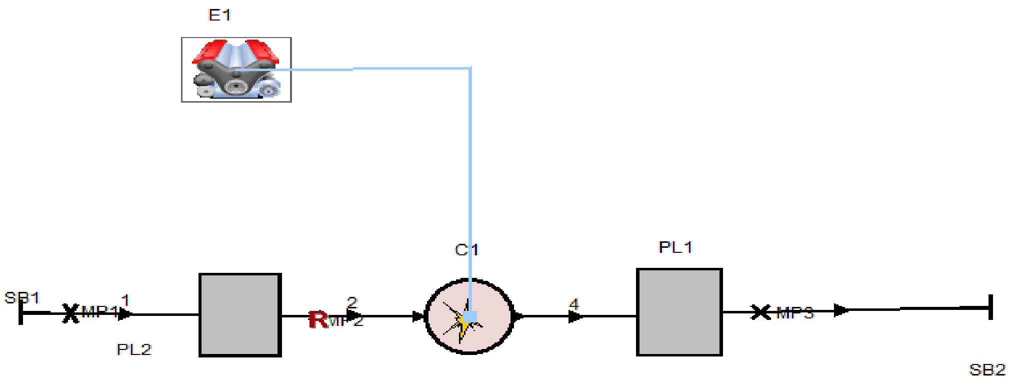

Fig. 2. AVL Boost simulation model

7. Piston dynamic model

Due to the presence of phase difference in the piston motion, both pistons do not arrive at their respective IDCs in harmony. Given the phase difference φ, the piston displacements are calculated as described in Ma et al. (2015).

$$X_{in} = (1 - \cos\left(\alpha - \frac{\varphi}{2}\right) + \frac{1}{2}\,\lambda sin^2\left(\alpha - \frac{\varphi}{2}\right))\,, \tag{1}$$

$$X_{ex} = r(1 - \cos\left(\alpha + \frac{\varphi}{2}\right) + \frac{1}{2}\,\lambda\, sin^2\left(\alpha + \frac{\varphi}{2}\right)), \tag{2}$$

where:
X_{in} – displacement of the intake piston,
X_{ex} – displacement of the exhaust piston,
r – Crank radius,
l – Length of the connecting rod,
α – equivalent crank angle,
$\lambda = r/l$ – ratio of crank radius to connecting rod.

Intake and exhaust pistons move towards each other face-to-face. However, the optional user defined piston motion in AVL Boost only allows the user to input a relative position of the piston motion profile. The connecting rod length of the engine is not specified in Naik et al., [9]. It is assumed that the connecting rod length is much greater than the crank radius. As $l >>> r$, λ tends to an infinitesimal value, causing the latter part of the piston motion equation to become insignificant [10]. The relative displacement of piston is indicated by the following equation.

$$X_{relative} = r\left(1 - \cos\left(\alpha - \frac{\varphi}{2}\right)\right) + r\left(1 - \cos\left(\alpha + \frac{\varphi}{2}\right)\right). \quad (3)$$

Ma et al. [8] demonstrates that with increasing piston motion difference, from 0ºCA to 18ºCA, the delivery ratio and scavenging efficiency increases progressively. However, the trapping efficiency increases initially and then declines after it achieves optimum efficiency at 15º CA, so does the indicated work. Henceforth, 15º is assumed for phase difference in our simulation. The phase difference also affects the maximum and minimum value of cylinder volume [8]. As the phase, difference increases the minimum distance between the pistons at IDC increases, the maximum distance at ODC decreases. Consequently, the compression ratio (CR) is adversely effected.

8. Effect of stroke-to-bore ratio on OP2S engine performance

The inquiry that this part of the paper attempts to answer is how do changes in the stroke-to-bore (S/B) ratio of an OP2S engine influence efficiency, fuel consumption and wall heat losses. The analysis was carried out on three combinations of bore and stroke to be contrived to yield a common swept volume of one litre with S/B ratios of 0.5, 1.25 and 1.69.

Tab. 1. Specification for S/B ratio investigation

	Stroke-to-Bore ratios		
Engine Specification	0.50	1.25	1.69
Bore (mm)	86.00	117.00	129.00
Stroke (mm)	172.00	93.60	76.50
Bore Surface Area (mm^2)	5808.80	10751.32	13069.81
Clearance height (mm)	12.24	6.61	5.44
2 x Piston Head	11617.61	21502.63	26139.62
Liner Area @ TDC	3307.28	2430.99	2204.86

In order to isolate the effect of S/B ratio, the simulations were executed with similar heat release rate HRR between the combinations. This is achieved by adapting the OP2S numerical model validated in the previous section. By this means the intake, exhaust and combustion systems are standardised. Thus, the effects of S/B ratio investigated are independent of the combustion system itself. Scavenging characteristics, heat transfer model, effective flow area and other operating conditions of the validated model were kept the same. A mutual piston motion profile is deployed across the S/B combinations to ensure the rate of change of volume in the cylinder is conserved Sandu et al. [15]. The assumption of flat piston head is incorporated in the analysis. The tabulated results below show negligible differences in peak cylinder pressure, temperature and MPRR, reassuring that heat release characteristics for the combination of S/B ratios remain the same.

Total wall heat flow defines heat rejected to the entire surface of the combustion chamber (piston and liner). As seen in Fig. 3, in-cylinder heat losses are largest for the lowest S/B ratio. A general trend of decrease in heat loss is evident with decreasing bore size (i.e. increasing S/B) as piston area available to dissipate the heat is reduced. The decrease in combustion chamber surface area-to-volume ratio with increasing S/B is seen as the predominant factor in such a trend [8]. With different combinations of S/B, similar temperature expansion occurs over a lower surface area for higher S/B ratios, hence the reduced heat loss [6, 16]. Wall heat losses increase with the engine cycle progressing into the expansion stroke, as burnt gasses are exposed to a greater surface area. It is apparent that maximum heat loss occurs at the very end of the combustion process. This is an indication of enhanced combustion efficiency, attributed towards the OP configuration. It is also to be noted from Fig. 3 that the peak is at minimum volume of the combustion chamber. The decrease in heat transfer reduces the heat rejected to the coolant, consequently leads to an increase in indicated thermal efficiency (ITE) as illustrated in Fig. 4(a). The change in ITE gets progressively better with increasing S/B ratio overall, the trend is non-linear. There is a steep increase in ITE from S/B ratio 0.5 to 1.25; however, the rate of change in ITE falls for S/B ratios above two.

Tab. 2. In-cylinder results from AVL BOOST simulation

	Stroke-to-Bore ratio		
Parameters	0.50	1.25	1.69
Peak Pressure	90.45	90.39	90.46
Peak Temperature	1515.75	1515.70	1517.86
IMEP	3.67	3.69	3.70
BMEP	2.51	2.53	2.54
ISFC (g/kwh)	202.22	199.60	200.21
Indicated Thermal Efficiency	0.41	0.41	0.41
Delivery ratio	7.45	4.40	7.45
Heat loss – Total (kJ)	-0.17	-0.15	-0.15
Indicated Torque	58.37	59.13	58.95
Mechanical Efficiency	68.38	68.57	68.68

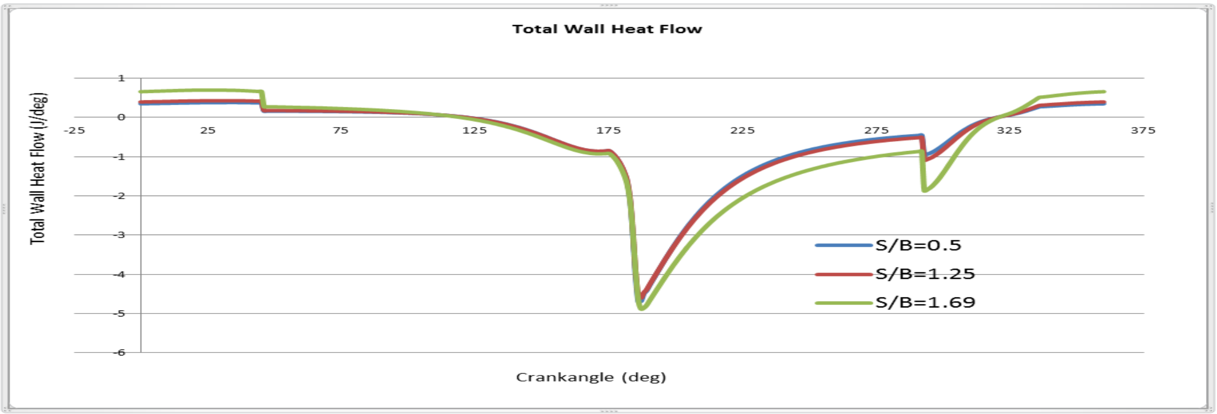

Fig 3. Wall heat flow from AVL BOOST simulation

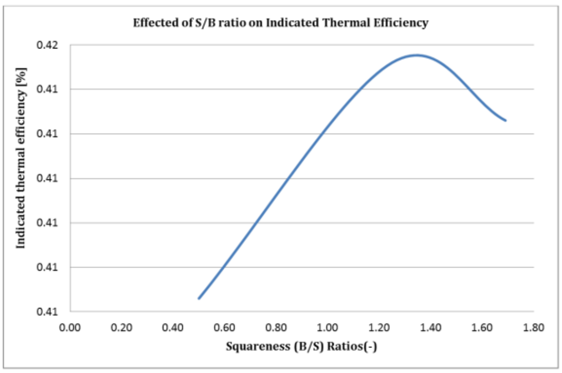

Fig. 4(a). Indicated thermal efficiency from AVL BOOST simulation

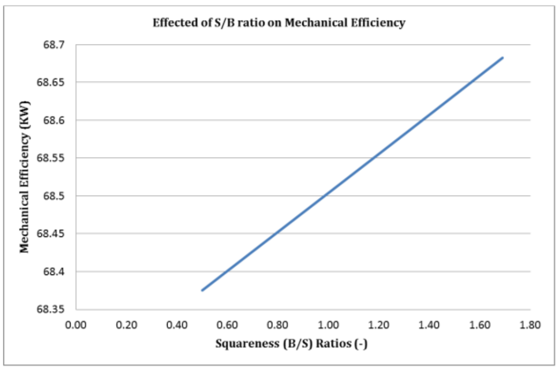

Fig. 4(b). Mechanical efficiency from AVL BOOST simulation

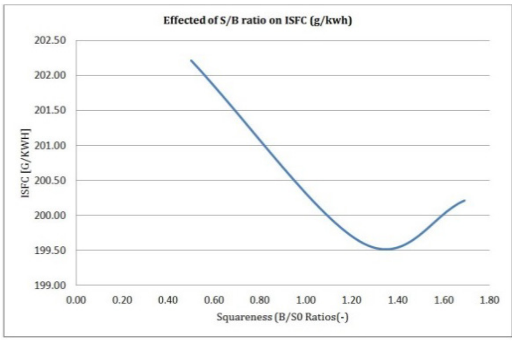

Fig 5(c). ISFC from AVL BOOST simulation

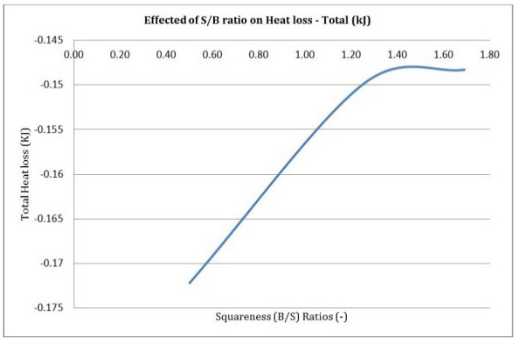

Fig 4(d). Total heat loss from AVL BOOST simulation

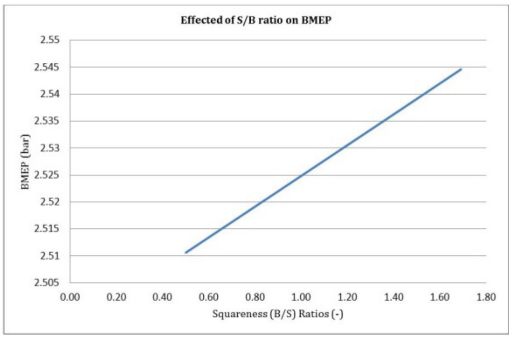

Fig 4(e). Brake Mean Effective Pressure from AVL BOOST simulation

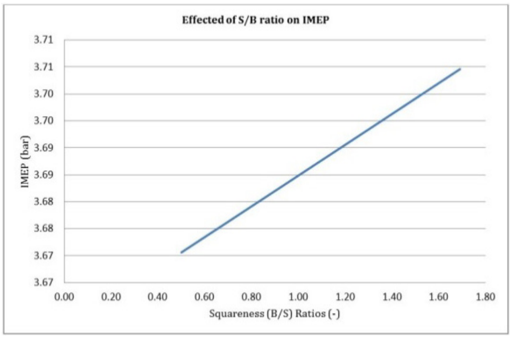

Fig 4(f). Indicated Mean Effective Pressure from AVL BOOST simulation

9. Conclusion

A brief history of the dawn of OP engines and its enduring research progress in the 21st century has been described. A comprehensive 1D thermodynamic modelling of the PAMAR™ single cylinder engine was performed in AVL BOOST to demonstrate fundamental engine performance. Wiebe, Wiebe 2-Zone and Double Wiebe heat release models were independently analysed. When it comes to compression ignition like combustion, Double Wiebe is the obvious choice in AVL BOOST. However, lack of information on fuel allotment produced unreliable results. Wiebe 2-Zone model unveiled its robustness over Wiebe and Double Wiebe models with its utilisation to validate the OP2S engine. With great confidence taken from the validation, three different S/B combinations of the OP2S engine for the same swept volume were investigated. The main conclusions of this analysis were that the heat losses decrease with increasing S/B ratio, due to the lower surface area-to-volume exposed, increase in ITE and mechanical efficiency was observed with increasing S/B ratios and improvement in fuel consumption is evident with increasing S/B ratio. Higher S/B ratio is desirable for applications that require better fuel efficiency.

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