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EFFECTS OF AIR INTAKE RUNNER LENGTH AND DIAMETER ON THE VOLUMETRIC EFFICIENCY OF SI ENGINE

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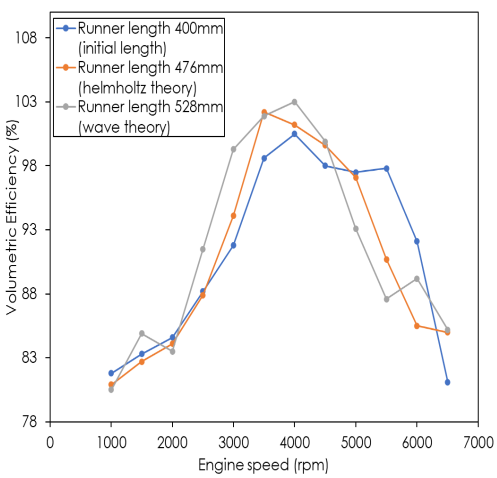
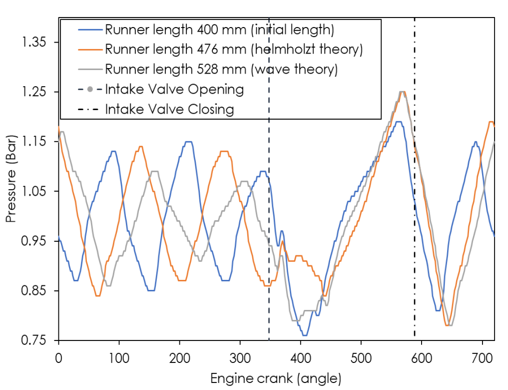
Graphical abstract





Abstract

The performance of an engine is significantly influenced by its volumetric efficiency which depends on how effectively air fills the engine cylinder. Low and inconsistent volumetric efficiency across different engine speeds can result from air pressure wave behaviour in the intake system. This study presents an approach in investigation on the impact of intake parameters to improve the performance and efficiency of the Proton CamPro 1.6L SI engine. The approach incorporates helmholtz resonance and wave propagation methods to improve air intake system. The objective is to determine the optimized intake runner length and diameter for improved engine efficiency. A combination of fluid flow simulation and 1D computational fluid dynamics was used to model intake behavior across engine speeds ranging from 1000 rpm to 6500 rpm. Various intake runner lengths and diameters were tested based on helmholtz and wave theory. Results showed that at mid-range rpm, volumetric efficiency improved by 8.71% when the intake runner length increased to 528 mm and by 5.34% when the diameter decreased to 34.7 mm. At high rpm, efficiency increased by 4.19% with a larger diameter of 44.4 mm and a shorter runner length of 400 mm. These findings suggest that intake tuning using resonance and wave propagation principles can significantly enhance engine performance. This study provides valuable insights into optimizing air intake design for better efficiency and performance delivery in naturally aspirated engines especially for the Proton CamPro engine.

Keywords: Intake runner length, intake runner diameter, wave propagation, volumetric efficiency, helmholtz resonator

Abstrak

Prestasi enjin sangat dipengaruhi oleh kecekapan volumetrik, yang bergantung kepada sejauh mana silinder enjin dapat diisi dengan udara secara berkesan. Kecekapan volumetrik yang rendah dan tidak konsisten pada pelbagai kelajuan enjin boleh disebabkan oleh gelombang tekanan udara dalam sistem pengambilan udara. Kajian ini membentangkan pendekatan dalam penyelidikan terhadap kesan parameter pengambilan udara bagi meningkatkan prestasi dan kecekapan enjin Proton CamPro 1.6L SI. Pendekatan ini menggabungkan kaedah resonans Helmholtz dan perambatan gelombang untuk menambah baik sistem pengambilan udara. Objektif kajian adalah untuk menentukan panjang dan diameter salur pengambilan yang dioptimumkan bagi meningkatkan kecekapan enjin. Gabungan simulasi aliran bendalir dan dinamik bendalir pengiraan 1D digunakan untuk memodelkan tingkah laku pengambilan udara pada kelajuan enjin antara 1000 rpm hingga 6500 rpm. Pelbagai panjang dan diameter salur pengambilan diuji berdasarkan teori resonans helmholtz dan gelombang. Hasil kajian menunjukkan bahawa pada kelajuan rpm pertengahan, kecekapan volumetrik meningkat sebanyak 8.71% apabila panjang salur pengambilan ditingkatkan kepada 528 mm, dan meningkat sebanyak 5.34% apabila diameter dikurangkan kepada 34.7 mm. Pada kelajuan rpm tinggi, kecekapan meningkat sebanyak 4.19% dengan diameter lebih besar iaitu 44.4 mm dan panjang salur lebih pendek iaitu 400 mm. Hasil kajian ini menunjukkan bahawa penalaan sistem pengambilan menggunakan prinsip resonansi dan perambatan gelombang boleh meningkatkan prestasi enjin secara signifikan. Kajian ini memberikan panduan yang berguna dalam mengoptimumkan reka bentuk sistem pengambilan udara untuk kecekapan dan prestasi yang lebih baik bagi enjin aspirasi semula jadi, khususnya enjin Proton CamPro.

Kata kunci: Panjang laluan penyerapan, diameter laluan penyerapan, penyebaran gelombang, kecekapan volumetrik, resonator helmholtz

1.0 INTRODUCTION

A lot of improvement was made by car tuner on the existent bought car. One of the criteria commonly improved is volumetric efficiency of the engine. Volumetric efficiency (VE) is the amount of air filled into the engine cylinder compared to its actual volume capacity. From the improved volumetric efficiency, the performance of the engine can be further improved. High volumetric efficiency helps engine to draw in more air in cylinder which improve combustion efficiency and thermal efficiency. Meanwhile, low volumetric efficiency could cause incomplete combustion and reduced thermal efficiency. This is supported by Talati *et al.* [1] which in their experimental study showed that variable intake length based on engine speed improves volumetric efficiency by 6.33% together with fuel consumption and thermal efficiency by 0.83% and 1.77% respectively. Based on study by Pahmi *et al.* [2] , they observed that improvement of volumetric efficiency was significantly affected by intake pressure wave during intake valve opening. This positive reflective wave can fill in more air into cylinder and contribute higher pressure peak of mean effective pressure in cylinder too. Thus, they concluded that intake pressure affected both volumetric efficiency and brake mean effective pressure (BMEP) of engine performance significantly.

In term of emission, Jemni *et al*. [3] stated that the intake manifold had significant impact on engine performance and emission and different design of intake manifold gave different result of volumetric efficiency and emission. The high volumetric efficiency engine would result in high engine performance and low emission. Raja & Selvam [4] claimed that intake manifold geometry consequently influenced the flow, air-fuel mixing and combustion in engine cylinder. This is due to the motion of fluid into combustion chamber is crucial to accelerate fuel evaporation and air-fuel mixing which increase combustion efficiency and reduce emission [5]. High velocity air flow can cause the turbulent flow within engine system which increases the thermodynamic heat transfer rates. As example, a study by Shin *et al.* [6] showed that a straight manifold design able to strengthen the in-cylinder flow which increased turbulent kinetic energy up 11% compared to curved intake manifold. The intake diameter is also one of the important parameter to get proper velocity that keep fuel suspended in air stream Breisacher *et al.* [7].

Improved volumetric efficiency allows engine to draw in larger volume of air into cylinder which contribute to more efficiency combustion and result in more power output for the same engine size. Thus, this allows improved volumetric efficiency of a smaller engine to give the same power output as larger and less efficiency engine which means this downsizing can improve on both fuel economy and emission [8].

Among the parameters tuned for volumetric efficiency improvment include throttle diameter, plenum volume, runner length, valve timing, and exhaust runner diameter. While these parameters are based on naturally aspirated induction engine, There are other two methods based on force induction engine which are turbocharger and supercharger. Turbocharger is one of method to improve volumetric efficiency by raising boost pressure [9]. Turbocharger is a forced induction device which force extra compressed air into the combustion chamber as to improve volumetric efficiency [10]. According to previous study, by feeding high density air into an engine, it produces better power and torque compared to natural aspirated engine [11]. Based on Bahng *et al*. [12], a downsized engines with turbocharged has a performance match to its larger counterpart as high specific output can be obtained with the intake boosting. Supercharger is other method to improve volumetric efficiency which use air compressor that drawn from the crankshaft to increase the pressure or density of air supplied to an internal combustion engine [13]. Same as turbocharger, it increasing power of engine by feeding high density air into the engine [14]. The improvement using supercharger has been studied by Cho [15], which found the combustion pressure, the rate of heat release and the burning rate of fuel-air mixture were higher than naturally aspirated engine. Thus, both output and torque performance were improved.

Other than that, parameters based on naturally aspirated induction engine also has a different impact on the volumetric efficiency improvement. A study done by Ghodke and Bari [16] showed that intake runner diameter and valve timing can improve volumetric efficiency of an engine especially when combining both. In their study, it was found that varying both parameters for each rpm had increment of 12% in volumetric efficiency.

This also supported by Bapiri and Sorusbay [17] as in their study, they stated that variable valve timing (VVT) is one of the solutions used in industries to solve problem of pumping losses during the intake of atmospheric gasses into cylinder for SI engines. Based on Tracy [18], variable valve timing was started developed in 1960s with an aim to improve volumetric efficiency, decrease NOx emissions and decrease pumping losses by allowing intake and exhaust valves to open earlier or later in 4-stroke engines.

According to Dresner and Barkan [19], variable valve timing yields a flatter maximum torque over engine speed curve when valve timing is optimized to engine speed [20]. The improvement was achieved through retarding the camshaft's timing at high rpm whereas advancing the camshaft's timing at low rpm.

Another study done by Aradhye & Bari [21] and Sawant *et al.* [22] showed that exhaust systems can be used to improve volumetric efficiency by increasing its exit diameter as it reduces the air flow restriction. Thus, it helps prevent air backflow into engine cylinder and improve the flowing of air out of the cylinder. In the latter study, tuning and varying both exhaust pipe diameter and exhaust valve timing can yield improvement an average of 4.69% in torque which result from volumetric efficiency improvement.

Other than that, it is also common that increasing the diameter of throttle body to improve the volumetric efficiency of the engine. A bigger area of diameter of throttle helps to allow more air flow into intake system. This is by reducing the restriction of air flow into the intake plenum. This is supported with a study done by Chauhan and Cho [23]. Based on their study, the improvement in throttle body design and operation also can help to improve the volumetric efficiency of an engine. This also is supported by Bordjane [24] which found that the volumetric efficiency can be improved by allowing the throttle body to move more air through certain opening angle of throttle valve into cylinder so piston's pressure can be increased. Furthermore, an invention by Glogovcsan [25], which has a constantly varied cross section of venturi in throttle body to increases the density of air flow into the manifold shown an improve in the acceleration of the engine.

Besides that, the volume of the intake plenum also has its impact on volumetric efficiency. Based on a study by Ceviz [26], volumetric efficiency can be improved between the range of 1700 rpm and 2600 rpm by increasing the plenum volume. It was found that an increase in plenum volume can increase the intake runner pressure which allows leaner mixture. This is supported by a study by de Souza *et al*. [27] that showed a new helmholtz resonance effect-based design of intake manifold can improve volumetric efficiency by 6% at 3500 rpm. This is because it has been noted that the cylinder volume is not fully occupied during intake period because of variation of volume and pressure drop along the system. Based on Malkhede & Khalane [28], volumetric efficiency was observed to be a function of engine speed and intake length especially at higher engine speed. In the study, they determined the optimum intake length based on Helmholtz resonator theory and acoustic theory.

Although prior research has studied the parameters to improve volumetric efficiency, it is difficult to determine the air wave behavior inside cylinder for different engine speed which affecting the volumetric efficiency. Thus, this research approach with helmholtz resonator and wave propagation method to investigate and compare the effects of length and diameter runner of the SI engine that impacts on volumetric efficiency. It was expected that this study can obtained the optimized air intake length and diameter for the engine to operate at optimum efficiency. Thus, it allows more cost-efficient improvement on volumetric efficiency can be conducted in future. It helps researchers to have a deeper understanding of the impact of air wave behavior in optimizing air flow of intake system which help improve volumetric efficiency of an SI engine.

2.0 METHODOLOGY

2.1 Engine Specification

This study was conducted based on CamPro 1.6L engine from Malaysia automobile manufacturer, Proton. Therefore, the engine specification used in the simulation is based on this engine. The constant value of components specification required to run this simulation are intake port, intake valve, engine cylinder, exhaust valve and exhaust port. These values would be constant throughout each simulation. The details of each value that is used in the simulation are as in Table 1.

Table 1 Specifications of the engine

Components	Specification value
Intake valve throat diameter	28 mm
Cylinder bore	76 mm
Cylinder stroke	88 mm
Cylinder con-rod length	180 mm
Compression ratio	10
Intake valve open	12° BTDC
Intake valve close	48° ABDC
Exhaust valve open	45° BBDC
Exhaust valve close	20° ATDC
Exhaust max lift	8 mm
Exhaust valve throat diameter	25 mm
Maximum power	82 kW at 6000 rpm
Maximum torque	148 Nm at 4000 rpm

Valve lift, valve timing and valve duration used for this simulation can be referred as in Figure 1 which include all four cylinders' phase. Based on the Figure 1, blue line indicated the inlet valve event while orange line indicated the exhaust valve event. The number above each curve indicated the cylinder number. The maximum opening point (°CA) for inlet and exhaust were 108° and -102.6° respectively from top dead center (TDC).

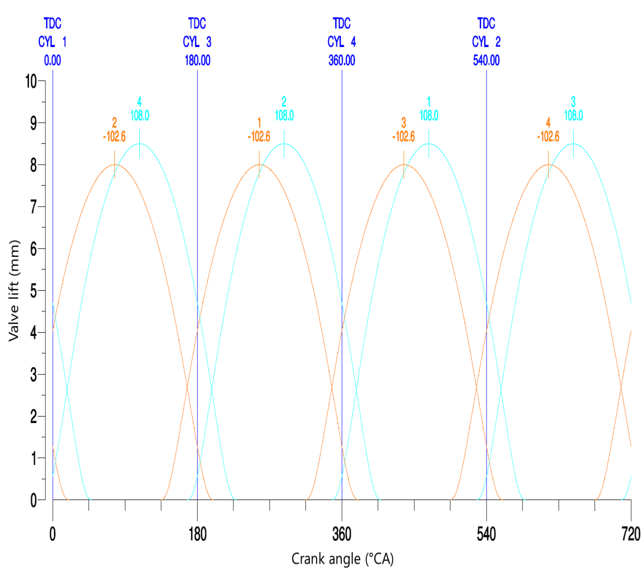


Figure 1 Valve timing diagram for all four cylinders

Figure 2 showed detail valve timing and duration diagram of a cylinder based on Table 1. According to Figure 2, intake valve open at 12° before top dead center (BTDC) and close at 48° after bottom dead center (ABDC). The exhaust valve open at 45° before bottom dead center (BBDC) and close at 20° after top dead center (ATDC).

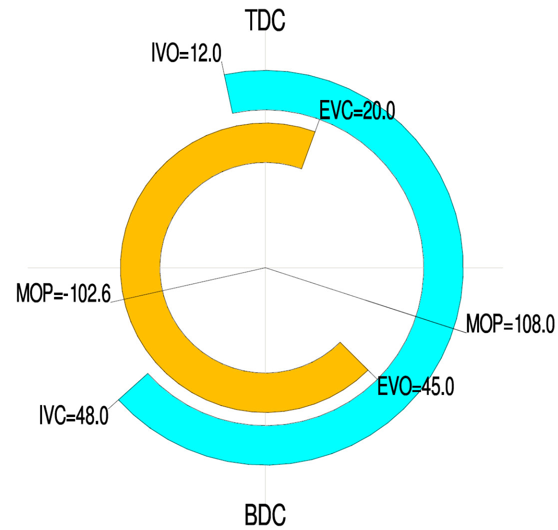


Figure 2 Intake and exhaust valve timing diagram of a cylinder

2.2 Configuration of Simulation

The simulation of this study was a steady-state simulation which had the analysis results based on stable engine operation and unchanging condition by time. In the simulation, the engine was simulated to run four stroke cycles which include intake, compression, power, and exhaust cycle. Thus, this simulation would include total of 720° crank angle that went through all four strokes cycle. These cycles are repeated for different rpm starting from 1000 rpm until 6500 rpm. The maximum engine speed of 6500 rpm was used for the test because the maximum power of the engine occurred slightly at 6000 rpm. The minimum engine speed of 1000 rpm was used as starting speed which same as the previous researcher experiment validation. Every rpm cycle, the amount of air fills up into the cylinder is calculated by fluid flow equation through 1D simulation. This calculation also includes air wave behavior inside runner before and after intake valve closing. Based on Table 2, the initial boundary condition of the air was 1 bar for inlet pressure and 20°C for inlet temperature which were reference point used by the simulation software. From this, the pressure and mass flow rate of air entering and exiting engine cylinder can be computed. Then, the volumetric efficiency can be calculated for each rpm range based on the trapped air inside cylinder once the intake valve closes. As for temperature, the calculation of VE was based on the input ambient temperature. Throughout this simulation, the air-fuel ratio was set to maintain at 14.7:1 ratio to allow complete or stoichiometric combustion occurred. This is crucial as the minimum air intake required for engine complete combustion depends on this ratio. Thus, the VE outcome achieved by this simulation were based on complete combustion. This setting can be referred in Table 2 which showed equivalence ratio equal to 1.

Table 2 Simulation boundary condition

Parameters	Specification value
Min. speed	1000 rpm
Max. speed	7000 rpm
Speed increment	500 rpm
Ambient air pressure	1 bar
Ambient air temperature	20° C
Inlet air pressure	1 bar
Inlet air temperature	20° C
Exit pressure	1 bar
Equivalence ratio	1

The simulation parameter setup in Lotus Engine simulation for this study is described in Table 3. Referring to Table 3 the intake runner length and diameter were optimized based on helmholtz resonance and wave propagation theory. Meanwhile, the initial runner is the actual parameters used in previous experimental study.

Table 3 Simulation parameter setup

Components	Intake runner length	Intake runner diameter
Initial engine runner	400 mm	39 mm
Helmholtz resonance runner	476 mm	34.7 mm
Wave propagation runner	528 mm	44.4 mm

For ensuring the accuracy and consistency of the simulation result, grid independence study was done as in Table 4. According to Table 4, there was no significant difference between Mesh 3 and Mesh 4 compared to other mesh type. Thus, Mesh 3 was used in this simulation as it provided a consistent result while save the computation time.

Table 4 Grid independence study

Mesh Type	Number of meshes	Volumetric efficiency
Mesh 1	164	87.58%
Mesh 2	337	88.73%
Mesh 3	655	89.39%
Mesh 4	1272	89.42%

For the flow of operations, this simulation starts with the engine cylinder rotating for induction cycle which draws in air from inlet goes to intake throttle, plenum, runner pipe, intake port, intake valve and then into engine cylinder. Next, the compression and power cycle occur based on crank angle rotation position. As a result, the air was pushed out to exhaust valve, exhaust port, exhaust plenum and until to exhaust exit. The efficiency of air drawn into the cylinder is calculated based on this operation for every cycle of rpm. The configuration of operation can be referred to as in Figure 3.

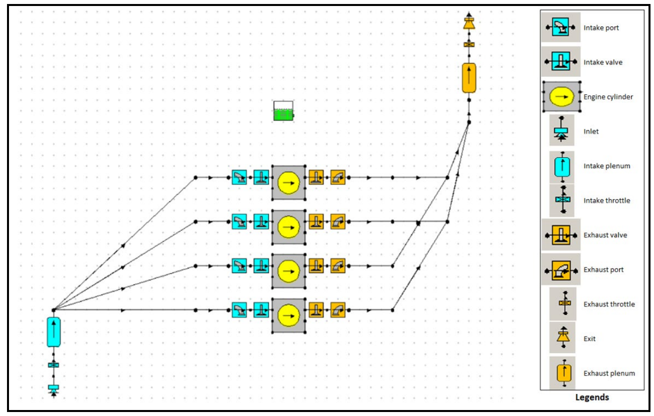


Figure 3 Configuration of Simulation in Lotus Engine Simulation

2.3 Equations

In this section, the equations that are required for the 1D simulation are explained based on equation of air fluid flow. The volumetric efficiency of an engine is strongly influenced by pressure wave phenomena that occur in engine intake system which is calculated based on wave equations. Thus, the condition of air within the intake system is determined by solving a set of conservation equations with wave equations for mass, momentum, and energy at each engine speed. These momentum, continuity, and energy equations are based on Lotus Engine Simulation Program and D. E. Winterbone & Pearson [29].

2.3.1 Volumetric Equation

This simulation determines the volumetric efficiency based on the ratio of the mass of air trapped in a cylinder to the mass of air that could be trapped within the swept volume as shown in Equation 1 [30]. Mass flow rate trapped in cylinder is determined by momentum, continuity, and energy equation.

$$\eta = \frac{M_i}{\frac{N}{2} V_s \rho_i} \qquad (1)$$

Where η is volumetric efficiency, M_i is mass flow rate, ρ_i is inlet density (based on ambient temperature and pressure), V_s is piston displacement and N is engine speed in revolution per unit time.

2.3.2 Momentum Equation

The momentum equation is also conservation law that has the sum of pressure forces and the shear forces applying on the control volume is equal to the sum of the rate of change of momentum within the control volume and the net flux of momentum out of it. This equation is as in Equation 2 [31].

$$-\frac{\partial(\rho F)}{\partial x}dx+\rho\frac{dF}{dx}dx-\frac{1}{2}pu^2 f\pi Ddx = \frac{\partial(upFdx)}{\partial t}+\frac{\partial(p\,Fu^2)}{\partial x}dx \quad (2)$$

Where F is flow cross-sectional area, ρ is density, p is momentum, u is initial velocity, f is friction coefficient, D is diameter of duct and t is time.

2.3.3 Continuity Equation.

The continuity equation states that the rate at which mass enters a system is equal to the rate at which mass leaves the system. The rate of change of mass within the control volume can be determined through the gradient of the mass flux, length of duct element, dx and its cross-sectional area, F [29].

$$\frac{\partial(pF)}{\partial t}+\frac{\partial(puF)}{\partial x}=0 \quad (3)$$

Where F is flow cross-sectional area, p is momentum, t is time, u is initial velocity and ∂x is length of element.

2.3.4 Energy Equation

The energy equation is derived by implementing the first law of thermodynamics to a controlled volume. Thus, the equation that have e_0 and h_0 represents the internal energy and enthalpy of fluid respectively as in Equation 5. The radial heat transfer from the gas to wall or vice versa can be implemented into this equation [32].

$$\frac{\partial(\rho e_0 F)}{\partial t}+\frac{\partial(\rho u h_0 F)}{\partial x}-q\rho F=0 \quad (4)$$

Where F is flow cross-sectional area ρ is density e_0 is internal energy h_0 is enthalpy t is time and u is initial velocity.

2.3.5 Helmholtz Resonator Equation

The helmholtz Resonator method is used to calculate the intake pipe dimensions required to achieve a desired tuning speed. The tuning speed of an engine is the speed at which the induction process matches the natural frequency of the combined pipe and cylinder system. The helmholtz resonator method considers the gas within the intake pipe as a finite incompressible mass. The volume of gas is considered as a spring with no inertia. Deceleration of the gas plug causes a peak in pressure at BDC. There are several ways in relation to the tuning speed. It can either be used to calculate the helmholtz speed directly, or it can be used to calculate the intake parameter required to provide tuning at a defined engine speed. The equation is as shown in Equation 5, 6 and 7 [31]

$$N=\frac{15a}{\pi}\sqrt{\frac{F_p}{L_p V_c}} \quad (5)$$

$$L_p=\frac{F_p}{V_c(\frac{N\pi}{15a})^2} \quad (6)$$

$$F_p=L_p V_c(\frac{N\pi}{15a})^2 \quad (7)$$

where N is engine speed (rev/min), F_p is flow pipe cross sectional area, L_p is Pipe length (m), and V_c is Mean cylinder volume, a is speed of sound, R is Gas constant and T is Temperature (K).

2.3.6 Wave Propagation

Engine volumetric efficiency improvement can also use a standard wave propagation equation to calculate the optimized intake length and diameter for a specified maximum power speed. Alternatively, it also can be used to determine the tuning speed for a specified intake parameter. By calculating the necessary intake pipe length and the speed of sound for the fluid, the optimized tuning speed can be determined. Meanwhile, by multiplying this time duration by the speed of sound, the wave propagation distance may be calculated. The intake length is half the total distance calculated as shown in Equation 8 [31]. For runner diameter, the relation of runner length to its diameter can be referred to as in Equation 9 [33].

$$L=0.5*\frac{60(120)}{360N}a \quad (8)$$

$$D=2((\frac{EVCD\ x\ 0.25\ x\ a\ x\ 2}{N\ x\ RV})-L) \quad (9)$$

where N is engine speed (rev/min), a is speed of sound, EVCD is effective valve close duration, RV is reflected value, L is length of intake runner, and D is diameter of intake runner.

3.0 RESULT AND DISCUSSION

In this section focuses on the results obtained from the simulation study using Lotus Engine Simulation software. The simulation was run on a 4-cylinder engine with different intake runner length and diameter. This setup was repeated to different engine speeds ranging from 1000 rpm to 6500 rpm. It starts with the validation of the simulation with previous research, then continued with the comparison between initial runner parameters, runner by helmholtz resonator theory and runner by wave propagation theory. The comparison is based on the volumetric efficiency, pressure wave and air mass flow rate in engine cylinder.

3.1 Validation of the Simulation

The validation of simulation is crucial to ensure the accuracy of this simulation model. Therefore, the result of this simulation was validated with study done by Mohiuddin and Rahman [34] that include both experiment and simulation result in their study. Their study was based on the same type and specification of engine as this current study. The specification of their study can be referred as in Table 1. This study simulation result and experimental result by Mohiuddin and Rahman [34] can be compared as in Figure 4. Based on Figure 4, it showed that only a slight variance between these two VE (%) especially at 3000 rpm to 4000 rpm. According to Mohiuddin and Rahman [34], the exceptional dip occurrence in Proton Campro engine was due to geometry of intake manifold design by the manufacturer which they studied for improvement. Overall, it showed a remarkably close result with an average of error is less than 3%.

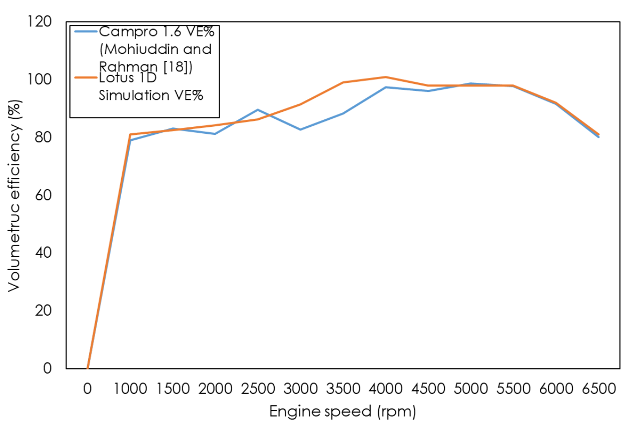


Figure 4 Volumetric efficiency comparison between experimental Mohiuddin and Rahman [34] and Lotus 1D simulation across a range of rpm

3.2 Intake Parameter based on Helmholtz Resonance & Wave Propagation

The helmholtz resonance and wave propagation method are used to estimate runner length of intake manifold so its natural frequency can be tuned with engine speed. Therefore, the tuned length and diameter of runner are based on the maximum speed of engine which is 6500 rpm that produced 79kW of horsepower as reference. This was because to maximize the engine powerband without compromise the performance of other lower RPM range. Thus, based on helmholtz resonance method, the length of the runner that matches its natural frequency with the engine speed is 476mm in length while wave propagation method has 528 mm of runner length for tuned natural frequency. As for the runner diameter that matches natural frequency with engine is 37.4 cm diameter for helmholtz resonance method while the runner diameter is 44.4 mm for wave propagation method.

3.3 Effect of Runner Length on Volumetric Efficiency

Based on Figure 5, optimized runner length 528 mm according to wave theory has VE improvement up to 8.17% at mid-range rpm between 3500 rpm to 4500 rpm and at tuned engine speed of 6500 rpm. However, the initial runner length 400 mm showed better VE between 5500 rpm and 6000 rpm. Therefore, it can be concluded the 528 mm is best used for low and mid-range rpm while 400 mm runner length is best used for high rpm range. This is supported by Bari and Sawant [35] which study showed the longer runner length improves performance at low engine speed range while short runner length improves performance at high engine speed. This result shows that tuning intake system for air flow optimizing needs at least two or more different specifications of tuning based on rpm for the best improvement of VE. Based on this study also, it showed that combining the initial runner length with wave theory runner length would result in the best VE performance for this engine.

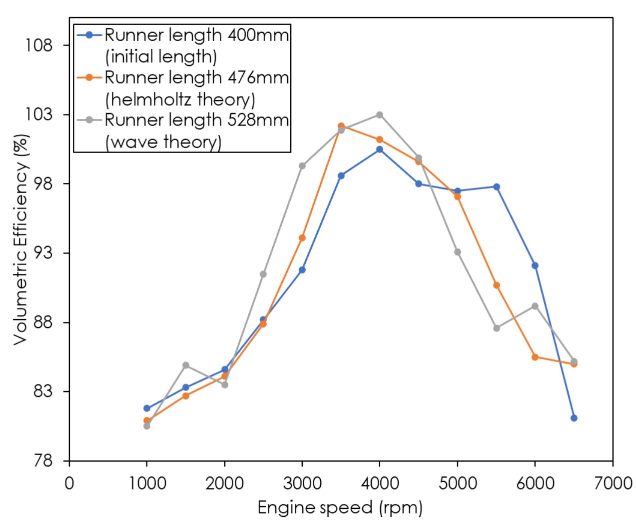


Figure 5 Optimized runner length between runner length 528 mm (wave propagation) and runner length 476 mm (helmholtz theory)

3.3.1 Effect of Runner Length on Pressure Wave

Figure 6 showed the details behavior of pressure wave occur at intake valve during engine full combustion cycle at 3500 rpm where the VE improvement occurred. The pressure wave result was based on the steady state condition by the simulation so the pressure result shown was based on repeated engine cycle of intake, compression, combustion and exhaust stroke. Thus, the pressure at 0° of crank angle was the continuation from 720° crank angle. According to this engine specification for cylinder 1, the intake valve opening occurs at 348 degrees and closing at 588 degrees for each cycle as in Figure 1. The opening and closing of valve cause the pressure wave as shown in Figure 6. The initial runner length has high pressure wave that reaches too early before intake valve opening while optimized runner length has its high-pressure wave reach right after intake valve opening. These are the reasons that optimized runner has improvement at mid-range rpm between 3500 rpm to 4500 rpm and 6500 rpm. It also showed that by tuning the length of the runner, the pressure can be delayed or fast forward.

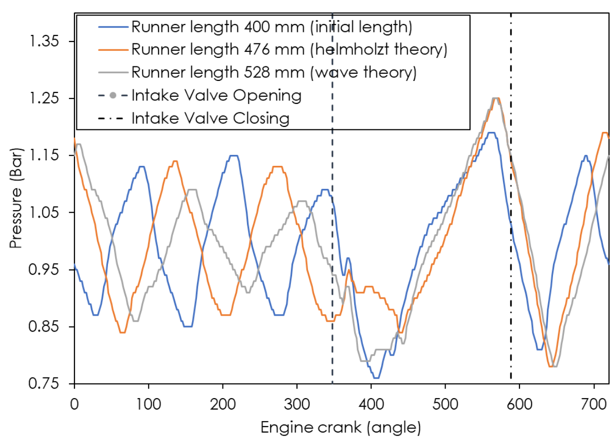


Figure 6 Pressure at inlet valve between runner length 39mm (initial runner), runner length 34.7 mm (Helmholtz theory) and runner length 44.4 mm (Wave theory) at 3500 rpm

Figure 7 showed the pressure wave occur at intake valve during engine full combustion cycle at 6000 rpm engine speed where the major VE improvement occurred. Based on Figure 7, both optimized runner lengths have high pressure waves that reach too early before intake valve opening which cause a great decline in VE performance. These are also the reasons that optimized runners have declined VE at rpm range between 5500 rpm to 6000 rpm. Pressure wave fluctuate along intake system until intake valve opened and the proper timing can result in improvement of volumetric efficiency [36]. By comparing both Figure 6 & 7, it shows that lower rpm of engine speed has higher frequency of wave while higher rpm of engine speed has lower frequency of pressure wave. Based on Figure 6, there was around four peak pressures before intake valve opening for 3500 rpm engine speed while Figure 7 showed that around two peak pressures before intake valve opening for 6000 rpm.

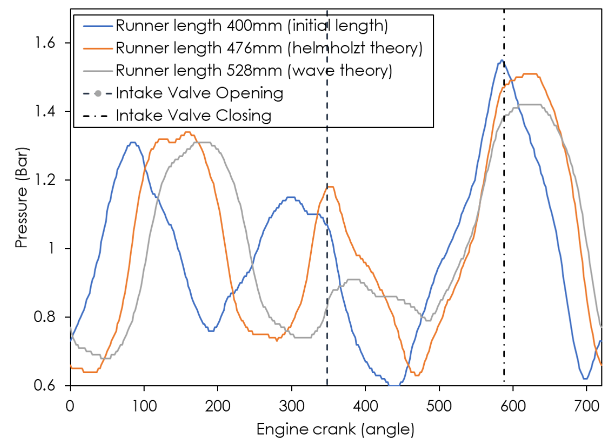


Figure 7 Pressure at inlet valve between runner length 39mm (initial runner), runner length 34.7 mm (Helmholtz theory) and runner length 44.4 mm (Wave theory) at 6000 rpm

3.3.2 Effect of Pressure Wave Timing on Mass Flow Rate

Figure 8 shows the impact of runner length and its pressure wave timing on mass flow rate of air entering engine cylinder at 3500 rpm. This is crucial as VE is the ratio of an actual volume flow rate of air into the cylinder to the rate at which the volume is displaced by the cylinder.

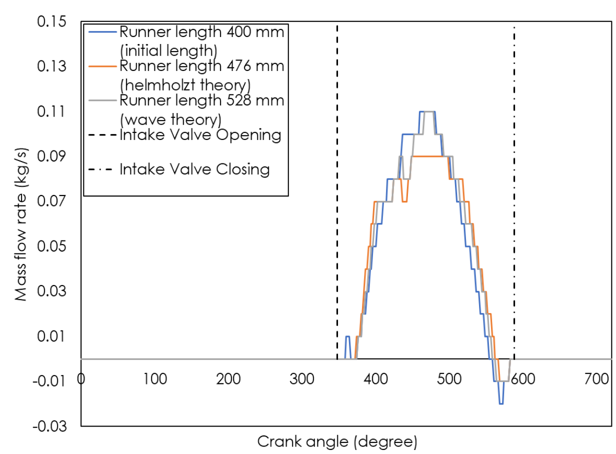


Figure 8 Mass flow rate at inlet valve between runner length 400 mm (initial runner), runner length 476 mm (helmholtz theory) and runner length 528 mm (wave theory) at 3500 rpm

Based on Figure 7, the high-pressure wave of initial runner length pressure wave reaches too early before intake valve opening which cause a high spike of reverse air flow between 550 and 600 crank angle degrees. This is the reason for the decline of VE as too much air flows out back from cylinder. This is supported by Sawant & Bari [37] which in their study showed that volumetric efficiency improved when the peak pressure wave occurrence was very close to the intake valve opening. According to Sammut & Alkidas [38], the efficiency of ongoing and next intake cycle was highly influenced by the back and forth propagation of pressure wave in intake runner. Meanwhile, both optimized runner lengths have lower amount of reverse air flow due to its high-pressure valve that reaches right after intake valve opening.

Figure 9 shows the impact of runner length and its pressure wave timing on mass flow rate of air entering engine cylinder at 6000 rpm.

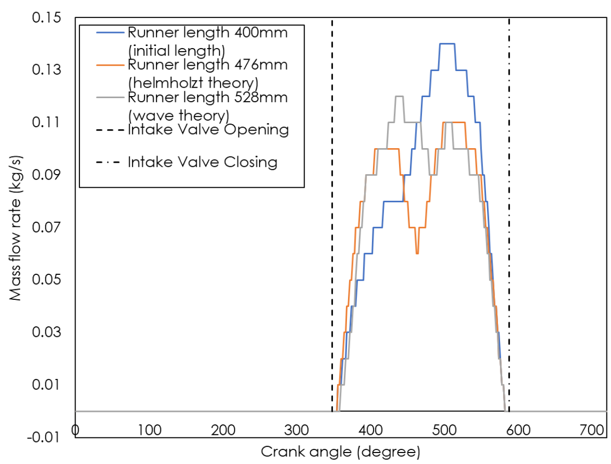


Figure 9 Mass flow rate at inlet valve between runner length 400 mm (initial runner), runner length 476 mm (Helmholtz theory) and runner length 528 mm (Wave theory) at 6000 rpm

Based on Figure 9, both initial runner length and optimized runner lengths have no reverse air flow from the cylinder. This is due to high engine speed which prevents air from reverse flow. The reverse flow condition tends to occur on low rpm range which is 4500 rpm and below. Although both runners have high-pressure reach too early before intake valve opening as in Figure 7, the initial runner length has higher amount of air mass flow rate due to its high-pressure wave timing is later than both optimized runner lengths. High engine speed condition prevents the reverse air flow which improve volumetric efficiency. This is supported by Ghodke & Bari [16] which highlighted in their study that high rpm had its air flow momentum sufficiently charge into opened intake valve and reduced the backflow of gaseous into the intake runner back. Other than that, a study by Allawi et al [39] also mentioned that waveform phenomenon due to engine cylinder and valve timing behavior can improve volumetric efficiency. At low engine speed, air flow was forced back inside intake runner due to cylinder and intake manifold equal pressure during intake opening. At high engine speed, the forced back of air flow was overcome by the continuous charging of high flow momentum of air due to increase piston speed. Thus, the pumping losses by backflow of air can be minimized and improve volumetric efficiency.

3.4 Effect of Runner Diameter on Volumetric Efficiency

Based on Figure 10, optimized runner diameter 34.7 mm according to helmholtz equation has VE improvement at mid-range rpm between 2000 rpm to 4500 rpm while optimized runner diameter 44.4 mm based on wave theory shows improvement at high-rpm range between 5500 rpm to 6500 rpm. However, initial runner diameter 39 mm showed better VE at 5000 rpm. Thus, it can be concluded that higher rpm requires bigger runner diameter while at same time pressure tuning play a crucial role in VE improvement. This is supported by Alves *et al*. [40] which study showed that VE improve by having bigger diameter of runner as the engine speed increased and tuned pressure wave arrival timing provide extra boost to VE performance. Based on the result in Figure 10, it shows that tuning intake diameter based on helmholtz resonance and wave propagation method need at least two different specifications of tuning for the best improvement of VE. Helmholtz runner diameter provides the best VE performance for low and mid-range rpm while wave theory runner diameter is best at high rpm engine speed.

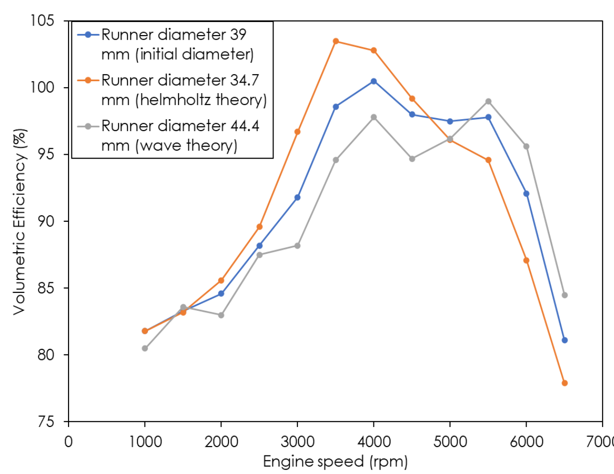


Figure 10 Optimized runner diameter between runner diameter 44.4 mm (Wave propagation) and runner diameter 34.7 mm (Helmholtz theory)

3.4.1 Effect of Runner Diameter on Pressure Wave

Figure 11 showed the details behavior of pressure wave occur at intake valve during engine full combustion cycle at 3500 rpm where the major VE improvement occurred. According to this engine specification, the intake valve opening occurs at 348 degrees and closing at 588 degrees for each cycle that causes the pressure wave as shown in Figure 11. The helmholtz runner diameter has the highest pressure of wave that reaches slightly earlier compared to other two diameter of runners. These proved as the reason for high improvement of volumetric efficiency for runner diameter 34.7 mm by helmholtz theory at mid-range rpm between 2000 rpm to 4500 rpm. It also found that by tuning the diameter of the runner, the pressure wave can only slightly be delayed or fast forward, but its impact is seen more on the increase of peak pressure of the wave. The increase and decrease of runner diameter, could affect the timing of pressure wave and its peak pressure. This is supported by Ghodke & Bari [16] which showed the same outcome as the high rpm require large flow diameter for better volumetric efficiency while low rpm need small flow diameter. Small flow diameter increases the velocity of flow that increase the momentum.

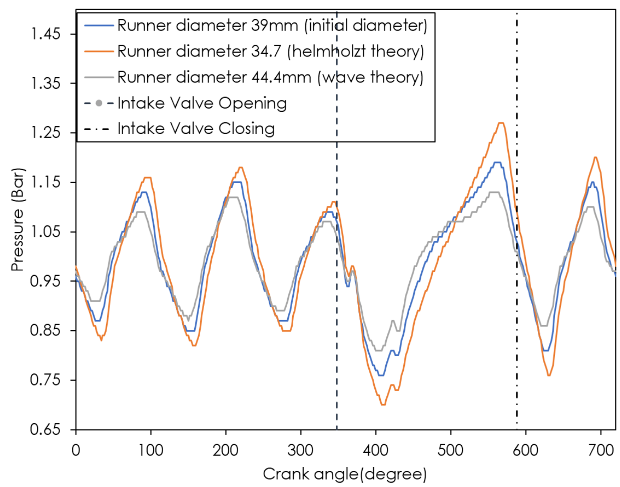


Figure 11 Pressure at inlet valve between runner diameter 39mm (initial runner), runner diameter 34.7 mm (Helmholtz theory) and runner diameter 44.4 mm (Wave theory) at 3500 rpm

Figure 12 showed the pressure wave occur at intake valve during engine full combustion cycle at 6000 rpm where the major VE improvement occurred.

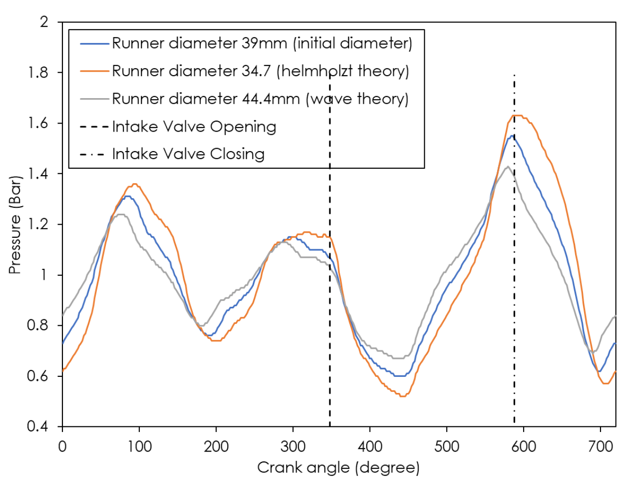


Figure 12 Pressure at inlet valve between runner diameter 39mm (initial runner), runner diameter 34.7 mm (Helmholtz theory) and runner diameter 44.4 mm (Wave theory) at 6000 rpm

Based on Figure 12, both initial runner length and optimized runner diameter helmholtz have high pressure wave that reaches too early before intake valve opening compared to optimized runner diameter based on wave theory. Thus, the wave theory runner diameter has the best VE improvement through delayed high-pressure wave reaching to intake valve opening. These are the reasons that optimized runner diameter by wave theory have improved VE at rpm range between 5500 rpm to 6500 rpm. By comparing both Figure 11 & 12, it shows that low and mid-range rpm of engine speed need early timing and high maximum pressure of wave for VE improvement while high-range rpm of engine speed need delay timing and low maximum pressure of wave for VE improvement.

3.4.2 Effect of Pressure Wave Timing on Mass Flow Rate

Figure 13 shows the impact of runner diameter and its pressure wave timing on mass flow rate of air entering engine cylinder at 3500 rpm. Based on Figure 12, the high-pressure wave of all runner diameter reaches slightly early before intake valve opening but with higher maximum pressure of wave compared to other two runner diameter which cause a minimum reverse air flow between 550 and 600 crank angle degrees. This is the reason for its higher VE as more air flows into the cylinder while less air flows out back from cylinder compared to the other two runners.

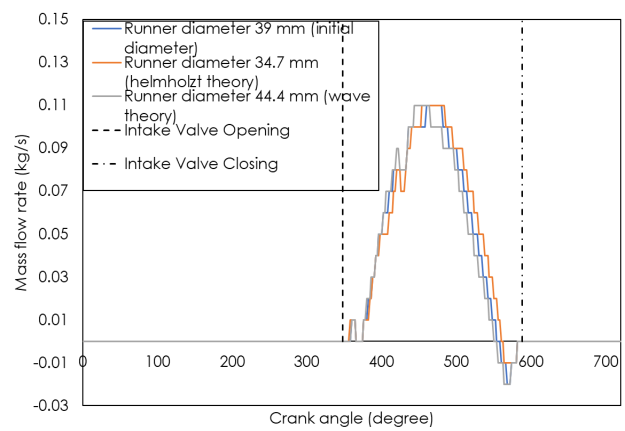


Figure 13 Mass flow rate at inlet valve between runner diameter 39mm (initial runner), runner diameter 34.7 mm (Helmholtz theory) and runner diameter 44.4 mm (Wave theory) at 3500 rpm

Figure 14 shows the impact of runner diameter and its pressure wave timing on mass flow rate of air entering engine cylinder at 6000 rpm. Based on Figure 14, all three runners have no reverse air flow from the cylinder. This is due to high engine speed which prevents air from reverse flow. The reverse flow condition tends to occur on low and mid-range rpm which is 4500 rpm and below. Although both initial and helmholtz theory runners have higher peak pressure of wave and reach almost same as wave theory runner, but it showed that lower peak pressure and slightly delayed wave of wave theory runner has more air entering the cylinder. Thus, it has better VE at range 5500 rpm to 6500 rpm. This was because of the advantage by initial runner which had bigger diameter of runner. Bigger diameter of air flow provides higher velocity of air and increases the momentum [16].

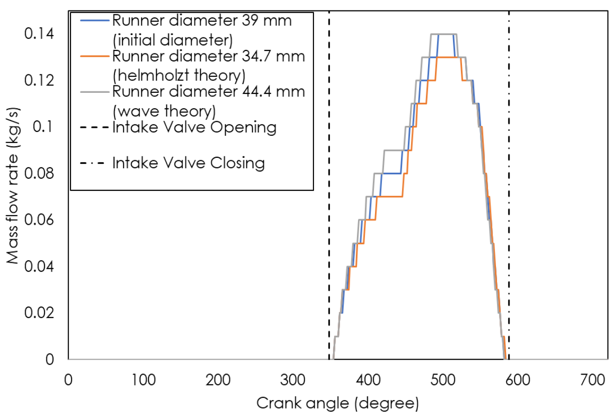


Figure 14 Mass flow rate at inlet valve between runner diameter 39mm (initial runner), runner diameter 34.7 mm (Helmholtz theory) and runner diameter 44.4 mm (Wave theory) at 6000 rpm

4.0 CONCLUSIONS

This research conducted a simulation study on the impact of intake parameters such intake runner length and intake runner diameter with helmholtz theory and wave theory approaches on the engine performance characteristics of 4-cylinder SI engine. It was conducted on 1D simulation using Lotus Engine Simulation software and the engine specification is based Proton Campro 1.6 L engine. The major findings include for a range from 2500 to 4500 rpm, the volumetric efficiency can be improved up to 8.17% with the increase of runner length to 528 mm based on wave theory. For the same range of engine, the volumetric efficiency also can be improved up to 5.34% with the decrease of runner diameter to 34.7 mm based on helmholtz theory. For a range from 5500 to 6500 rpm, both longer runner lengths based on helmholtz and wave theory showed decline in VE performance while short initial runner length has higher VE. For the same range of engine, the VE can be improved up to 4.19% with the increase of runner diameter to 44.4 mm based on wave theory. Combining these findings, it can be concluded that the optimum runner for VE at mid-range engine speed are 528mm length and 34.7mm diameter runner while at high range engine speed are 400mm length and 44.4mm diameter runner.

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Conflicts of Interest

The authors declare that there is no conflict of interest regarding the publication of this paper.

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