National Conference in Mechanical Engineering Research and Postgraduate Studies (2nd NCMER 2010)
3-4 December 2010, Faculty of Mechanical Engineering, UMP Pekan, Kuantan, Pahang, Malaysia; pp. 183-194
ISBN: 978-967-0120-04-1; Editors: M.M. Rahman, M.Y. Taib, A.R. Ismail, A.R. Yusoff, and M.A.M. Romlay

INFLUENCE OF ENGINE SPEED AND INJECTION TIMINGS ON IN-CYLINDER HEAT TRANSFER FOR PORT INJECTION HYDROGEN FUELED ENGINE

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ABSTRACT

The overall heat transfer process within the in-cylinder for port injection hydrogen fueled internal combustion engine (H2ICE) has been investigated through simulation, while the experimental data has been utilized for validation purpose of the adopted numerical model. One-dimensional gas dynamics was used to describe the flow and heat transfer in the components of the engine model. The engine model is simulated with a variable injection timing, engine speed and equivalence ratio (φ). Simulation has been executed for 60 deg ATDC $\leq \theta_{inj} \leq$ 160 deg ATDC (during the intake stroke), 1000 $\leq$ rpm $\leq$ 6000 and $0.2 \leq \varphi \leq 1.2$. The baseline engine model with gasoline fuel has been verified with experimental data and reasonable agreement has been achieved. The overall results show that there is a combined influence for the engine speed and equivalence ratio (φ) on the overall heat transfer characteristics. The identification for the effect of the injection timing on the overall heat transfer characteristics has been failed because the injection issue is not considered within the combustion approach.

Keywords: Heat transfer, Hydrogen fuel, Port injection, Injection timing, engine speed.

INTRODUCTION

Hydrogen, as alternative fuel, has unique properties give it significant advantage over other types of fuel. In economic terms, it would be pointless material the use of hydrogen in fuel cells compared with using it as a fuel in the operation of internal combustion engines. Therefore; concentration of the researches on the internal combustion engine application for this alternative fuel is logical and more beneficial. Hydrogen can be used as a clean alternative to petroleum fuels and its use as a vehicle fuel is promising in the effects to establish environmentally friendly mobility systems (Rahman et al. 2009a; Verhelst and Sierens, 2007; Bakar et al., 2009). Extensive studies were investigated hydrogen fueled internal combustion engines (Drew et al.,2007; Rahman et al. 2009b,c,d; Ganesh et al., 2008; Mohammadi et al., 2007). Hydrogen fuel with its unique properties needs more studies to gain the essential understanding of to employ it in the practical applications of human life. One of the important issues related to the internal combustion engine is the heat transfer analysis. Within the last decade, several studies have been carried out for treating the heat transfer phenomenon in hydrogen fueled internal combustion engine (Wei et al., 2001; Shudo and Suzuki, 2002a,b; Demuynck et al., 2009; Rahman et al., 2010a,b). Even that, the full understanding of the whole heat transfer issue for internal combustion engine with hydrogen fuel is not clear yet. The useful way to understand the nature of the internal combustion engine is examining the effect of variation the main operation parameters on each physical process occurred within the engine. For that, this work has been detected to clarify the influence of a group of the main operation parameters on the heat transfer issue for hydrogen fueled engine. Currently, the main operation parameters under taken are: the engine speed, equivalence ratio and injection timing. The engine speed parameter has a direct impact on the heat transfer because it represents the main driving force for the forced convection which is the dominant heat transfer mode for ICE application. While, the influence of the other two parameters on heat transfer does not fully understand, especially in case of hydrogen fuel. The main motivation for current investigation has been come from that gap in description of the influence these parameters on the heat transfer process. Therefore, the authors' effort for the current study has been detected to merge this gap in heat transfer analysis.

MATERIALS AND METHODS

Computer simulations for the engine instead of testing every point on a dynamometer represent the alternative methodology to analyze the internal combustion engine. Engine simulation currently is more able, faster, widely, and yet more precise. One-dimensional modeling represents one of the most widely used simulation approaches for analyzing the internal combustion engine to specify the characteristics. A one-dimensional gas dynamic code was developed utilizing the GT-Power for engine performance prediction. The general characteristics of the code are described in details (Ciesla et al., 2000; Morel et al., 2003), while the model set up process is abridged outlined in the following section.

Computational Model

The real details for the engine components are introduced to the GT-Power model. The approach of building a GT-Power model starts with segmenting the power-train into its components. The main components of the power-train are the air cleaner, throttle, intake manifold, engine and the exhaust system. All of these engine components connected by exploiting the orifice connection object. The engine specifications are listed in Table 1. A four stroke, single cylinder, spark ignition, port injection and hydrogen fuel with dual valves for the intake and exhaust systems is developed as shown in Figure 1.

The injection of gas hydrogen fuel was located in the midway of the flow split before the intake port by utilizing a single sequential pulse fuel injector. Fuel injector is specified by an equivalence ratio, a fuel delivery rate and injection timing. The equivalence ratio was set for a wide range, from very lean to rich mixture. Adjusting of the injection timing has been settled consisting with the opening period for intake valves. Several considerations for heat transfer and pressure losses calculations were made the model more realistic. Firstly, the heat transfer multiplier is used to account for bends, additional surface area and turbulence caused by the valve and stem. Secondly, the pressure losses in these ports are included in the discharge coefficients calculated for the valves. Finally, the additional pressure losses due to wall roughness was accounted during all computation by introducing the equivalence values of the surface roughness for all pipe segments according to type of material (Rahman et al., 2009c). The intake and exhaust manifolds specifications are presented in Table 2. One-dimensional gas dynamics model is used for representation the flow and heat transfer in the components of the engine model. Engine performance can be studied by analyzing the mass, momentum and energy flows between individual engine components and the heat and work transfers within each component.

Table 1: Engine specifications

Parameter	Value	Unit
Bore	57	mm
Stroke	58.7	mm
Connecting rod length	100	mm
Compression ratio	10.4	-
Inlet valve open	29	CA(BTDC)
Exhaust valve open	59	CA(BBDC)
Inlet valve close	59	CA(ABDC)
Exhaust valve close	29	CA(ATDC)
No. of cylinder	1	-

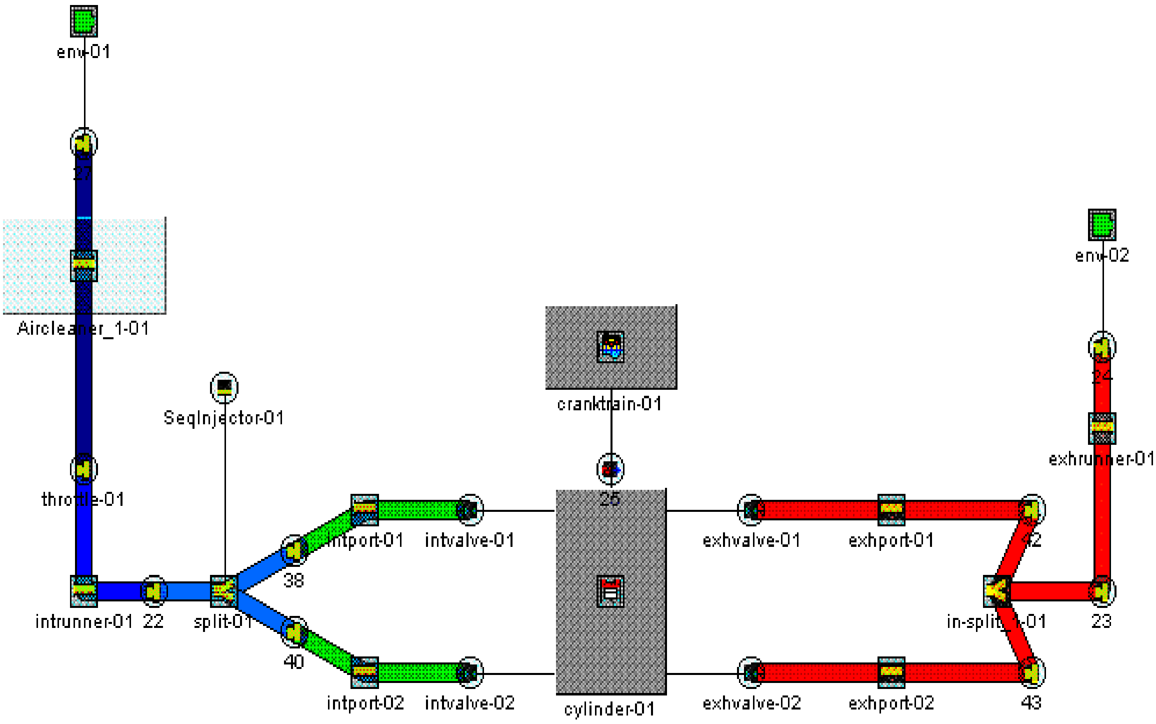


Figure 1: Computational model for single cylinder four strokes, port injection hydrogen fueled engine

Table 2: Intake and exhaust manifolds specifications

Part Name	Diameter(mm) Inlet	Diameter(mm) Outlet	Length (mm)	Volume (mm^3)	Surface roughness (mm)
Air cleaner	20	20	150	-	0.0025
Intake runner	22	22	75	-	0.25
Intake Flow split	-	-	-	62800	0.25
Intake port	21	20	40	-	0.2
Exhaust runner	17	17	180	-	0.25
Exhaust flow split	-	-	-	62800	0.25
Exhaust port	18	17	40	-	0.2

Simulation of one-deminsion flow involves the solution of the conservation equations; mass, momentum, and energy in the direction of the mean flow as discribed (1)-(3).

The mass conservetion equation is defined as Eq. (1).

$$\frac{dm}{dt} = \sum_{boundaries} m_{flux} \tag{1}$$

The momentum conservetion equation is expressed as Eq. (2).

$$\frac{d(m_{flux})}{dt} = \frac{dp.A + \sum_{boundaries}(m_{flux}.u) - 4.C_f.\frac{\rho.u^2.A.dx}{2D}}{dx} - \frac{C_{pl}.(\frac{1}{2}.\rho.u^2).A}{dx} \tag{2}$$

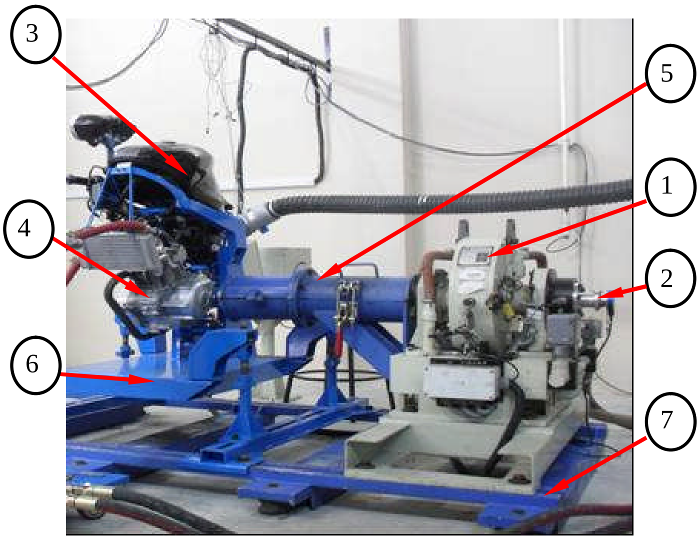
The energy conservetion equation is calculated by Eq. (3).

$$\frac{d(m_{flux}.e)}{dt} = p\frac{dV}{dt} + \sum_{boundaries} m_{flux}.H - \alpha.A(T_g - T_w) \tag{3}$$

EXPERIMENTAL SETUP

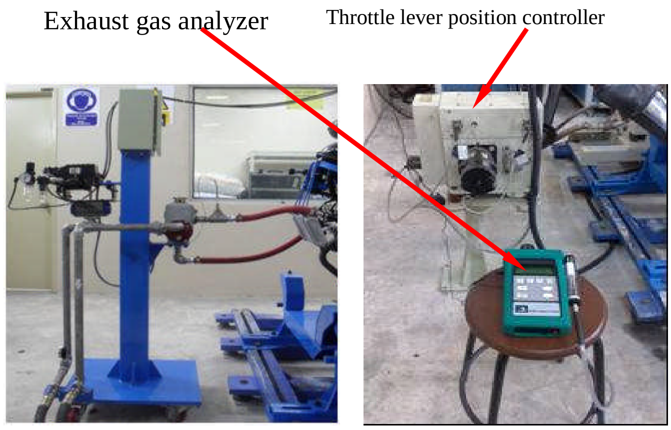
A YAMAHA FZ150i single cylinder, four stroke, water cooled motorcycle engine has been utilized for experimental tests. It is naturally aspirated and comes with port injector. The aluminum cylinder head has a pent-roof configuration for the combustion chamber with a center mounted sparkplug. The cylinder head has dual intake and exhaust valves actuated by a single overhead cam (SOHV). The engine is representative of the small engine group. The engine specifications also are listed in Table 1. The engine with eddy current dynamometer engine test-rig is shown in Figure 2. The engine cooling system was modified in order to carry out the experiment in a sufficient way. The heat liberated by the engine combustion is transferred to the closed cooling water circuits through a specified heat exchanger as shown in Figure 3(a). The test cell also includes an eddy current dynamometer for loading the engine. A Dynalec Controls, Model ECB-15 kW, eddy current dynamometer was utilized for the power absorption and engine speed regulation. This dynamometer was selected after bring down the values of the engine specifications (maximum torque and maximum power with corresponding engine speed values) on the dynamometer performance curve for calibration purposes. The dynamometer electromechanically absorbs the power delivered by the engine. The heat generated by the applied torque is removed by utilizing a closed cooling circuit. The cooling circuit for dynamometer is composed of an electrical centrifugal pump, cooling fan, pipes for connection the cooling circuit and special water reservoir filled with tap water.

Throttle control is one of the basic controls required for the testing of an engine. All tests have been done under a wide open throttle condition. A Dynalec Controls, type TLPC/612A for control unit and type TM/612B for throttle actuator, has been utilized in this work as shown in Figure 3(b). The acquired database included in-cylinder pressure against crank angle within a combustion analyzer to obtained the indicated mean effective pressure (IMEP) under multi rated engine speed. The in-cylinder pressure trace was instantaneously measured by using FGP transducer. The FGP pressure sensor type XPM5-100 bar is installed on the cylinder head to measure the engine cylinder pressure. A special threaded hole was made in the cylinder head to accept the pressure sensor. Thread size of the FGP sensor type XPM5-100 bar is M5x0.8 mm. A Kistler Type 2613B crank angle encoder, mounted at the end of the dynamometer's shaft, provides the clock signal for measurement. The crank angle encoder has been mounted by fitting an adapter to the free end of the dynamometer's shaft. The crank angle encoder and pressure sensor are connected with DEWE5000, a computerized based combustion analyzer, completed with a data acquisition system. Combustion analyzer unit, DEWE5000 has been employed for pressure data collection in crank angle domain. One of the input parameters for the simulation model is the equivalence ratio. Currently, all exhaust air/fuel ration measurements have been conducted using a portable KANE (Auto 4-1/Auto 5-1) hand held exhaust gas emission analyzer as shown in Figure 3(b).



1. Dynamometer, 2. Crank angle encoder, 3. Fuel tank, 4. Tested engine
5. Metal cover for coupling assembly, 6. Engine stand, 7. Dynamometer structure bed

Figure 2: The eddy current dynamometer engine test-rig



(a) (b)

Figure 3: (a) Engine heat exchanger with its accessories (b) Throttle lever position controller and exhaust gas analyzer

RESULTS AND DISCUSSION

Overall heat transfer simulations inside the cylinder of four stroke, port injection, spark ignition, hydrogen fueled engine model is performed for the three operation parameters namely injection timing, equivalence ratio (φ) and engine speed. The equivalence ratio was varied from rich (φ = 1.2) to a very lean (φ = 0.2) with 0.1 interval, and the engine speed varied from 1000 rpm to 6000 rpm with 1000 rpm interval. The injection timing has been settled within the intake valves opening period according to the engine valve timing as well as to be consisting with benchmarks stated by previous studies on the occurrence of the backfire phenomenon (Sierens and Verhelst, 2003; Liu et al., 2008).

Model Validation

The experimental test rig was used in order to provide validation data for the baseline engine numerical model. Therefore, for the verification purpose the baseline engine simulation model has been carried out in a test rig of a single cylinder research engine. The baseline engine model (with gasoline fuel) has been verified with experimental results. The in-cylinder pressure data represents the most crucial indicator for all engine modeling aspects. Thereby, currently the indicated mean effective pressure (IMEP) for the baseline engine model with gasoline fuel has been utilized as a proof. Besides that, the brake torque for the engine has been utilized to assess the numerical results. The results of the computational model are verified against the experimental results of the gasoline fueled engine used during the experimental work as shown in Figure 4 and 5.

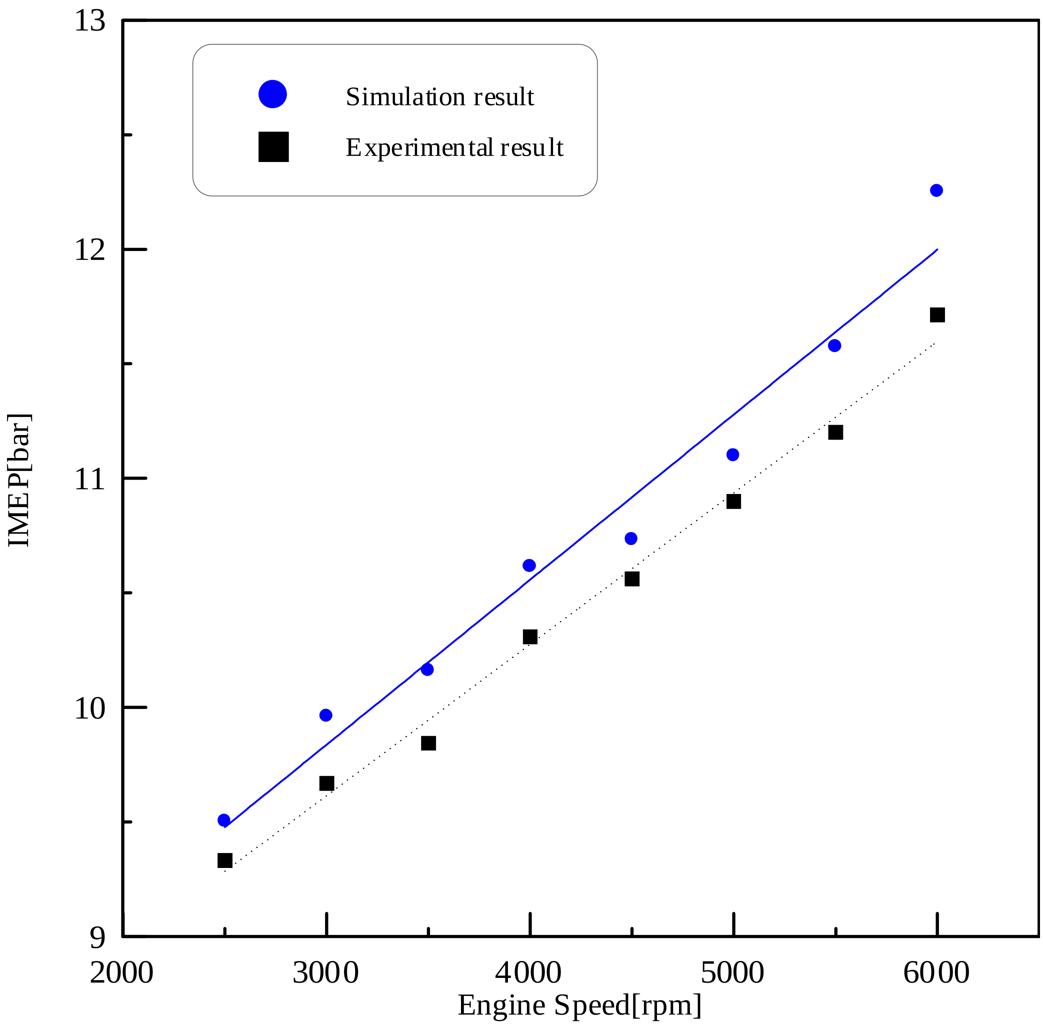


Figure 4: Comparison between experimental and simulated cylinder IMEP for baseline gasoline engine model

In Figure 4, the experimental indicated mean effective pressure IMEP against engine speed are compared with computational results. It can be seen that the computational results are closely follows the experimental results. The predicted results are reasonably in good agreement with the experimental results (within maximum relative error 4% for IMEP). Figure 5 shows that the experimental brake torque for the gasoline fueled engine against engine speed is compared with the computational results. It is shown that the computational results have the same trends with the experimental results. In spite of the large deviation between the computational and experimental results (with maximum relative error 7.5 % for the engine brake torque), the adopted model is still capable of describing the engine performance with acceptable coincidence. It can be seen obviously from this comparison that the present simulation model is capable to predict with the sufficient accuracy the engine performance of SI engine using gasoline, and then it can extend to be used for hydrogen fuel.

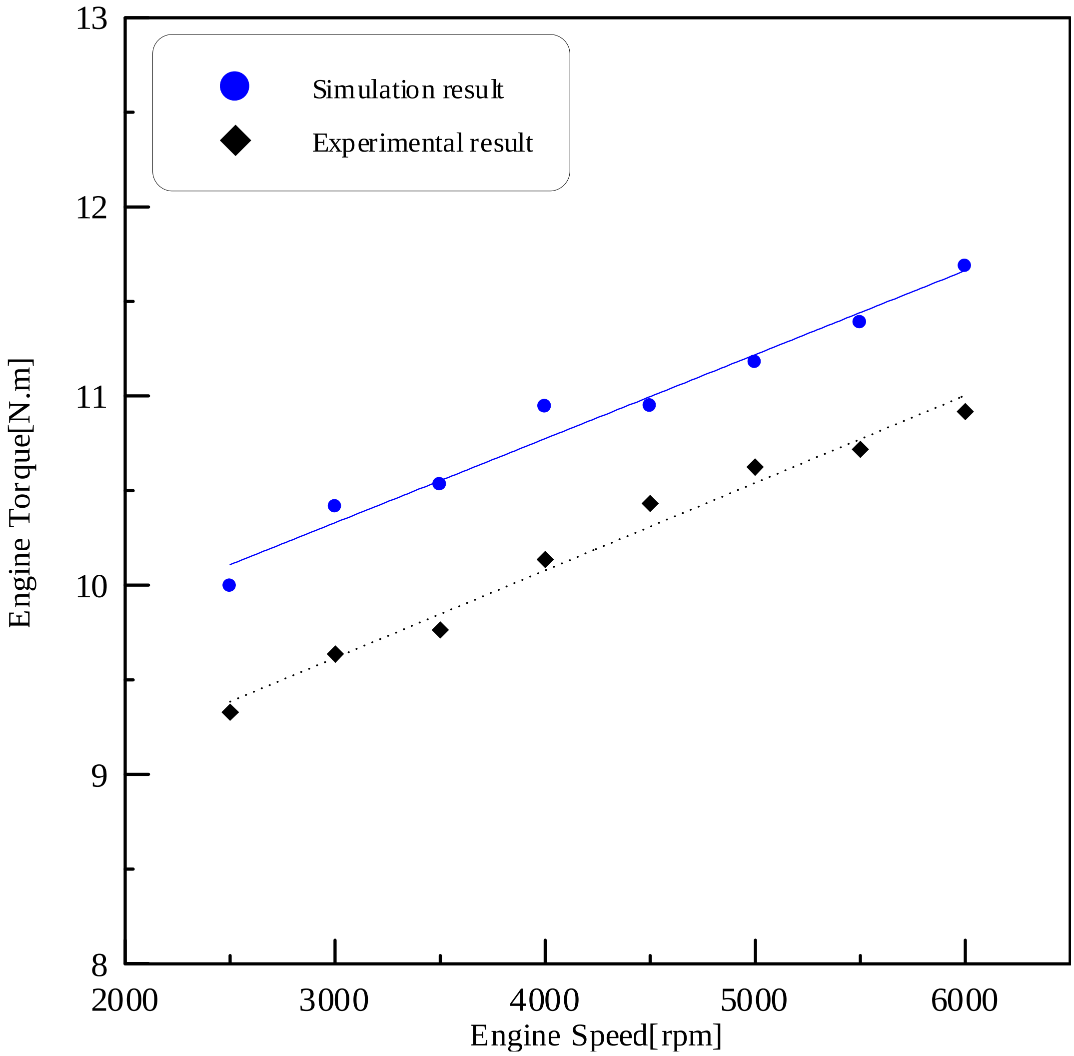


Figure 5: Comparison between experimental and simulated engine torque for the baseline gasoline engine model

To indicate the combined influence of the operation parameters on the overall heat transfer characteristics namely the overall heat transfer rate and the percentage ratio of heat transfer to the total fuel energy. The overall heat transfer rate trends in terms of the engine speed, injection timing and equivalence ratio are represented in Figure 6, 7 and 8 respectively. By observing this action for the heat transfer rate can be knowledge characteristic of the overall heat transfer process inside the cylinder for port injection H_2ICE. A combination effect of the engine speed and injection timing on a heat transfer rate is revealed in Figure 6. The heat transfer rate increase linearly with inclease of engine speed for all injection timing values. This behavior is expected because of strengthening of the forced convection as engine speed has been increased. The combined influence for the equivalent ratio and injection timing on the rate of heat transfer rate can be recognized in Figure 7. The heat transfer rate increases as increases of the equivalent ratio from the very lean limit until the stoichiometric limit for all injection timing values. Beyond the stoichiometric limit, the heat transfer rate is degraded for all injection timing due to the insufficient oxygen amount for completing the combustion process for all charge mixture.

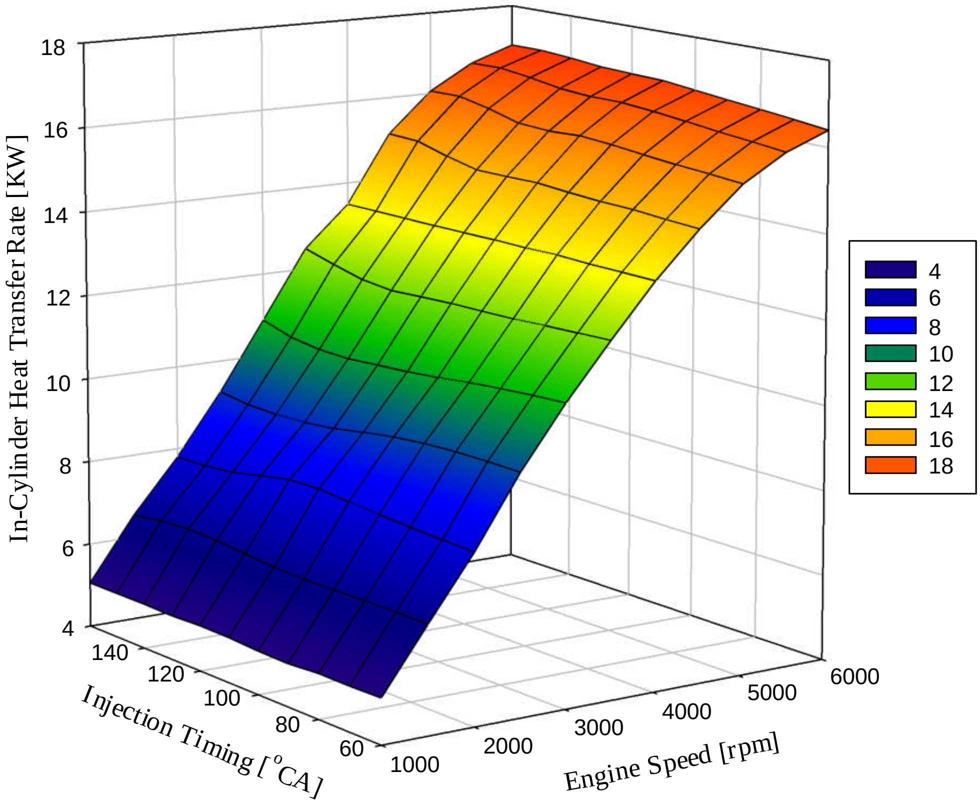


Figure 6: Variation of overall heat transfer rate with injection timing and engine speed

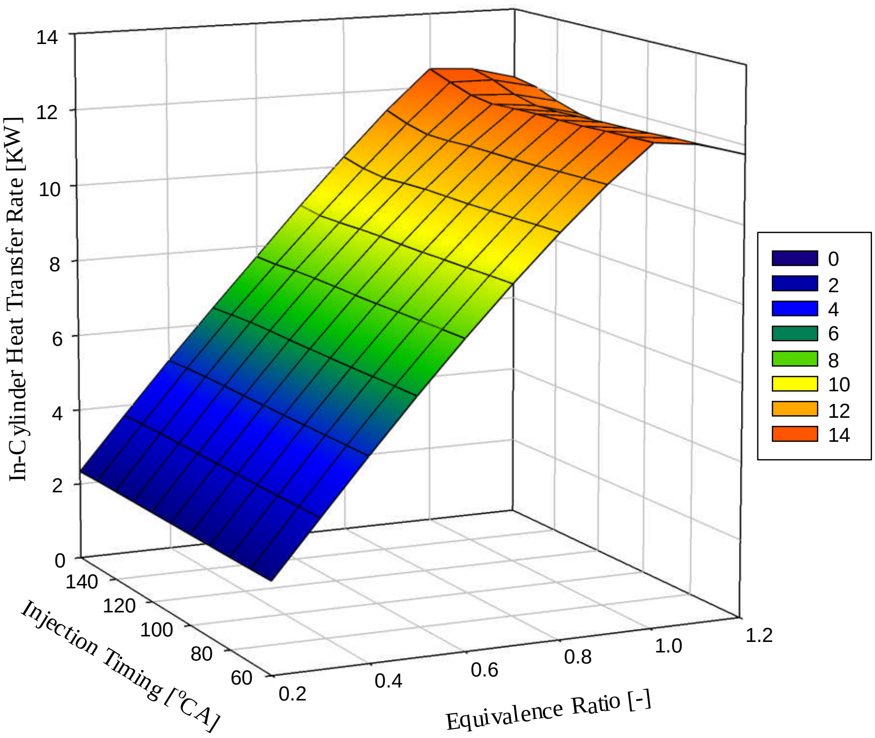


Figure 7: Variation of average heat transfer rate with injection timing and equivalence ratio

On the other hand, the effect of the injection timing combined with engine speed and equivalence ratio on the heat transfer rate could not be captured with current one-dimension gas dynamic mode. The model weakness in capturing of the influence for the injection timing on the heat transfer rate comes from the excluding of the injection information within the combustion approach. Variation of the in-cylinder overall heat transfer rate with an engine speed and equivalence ratio is clarified in Figure 8. The overall heat transfer rate increase as the engine speed increase for all equivalence ratio values due to increase the driving force (forced convection) for the heat transfer mechanism inside the engine cylinder. While the overall heat transfer rate increase as equivalence ratio decrease for all engine speed values and the increment in the overall heat transfer rate with increasing of the engine speed is decreased as the equivalence ratio increase because of decreasing in the energy content for the inlet charge to the cylinder.

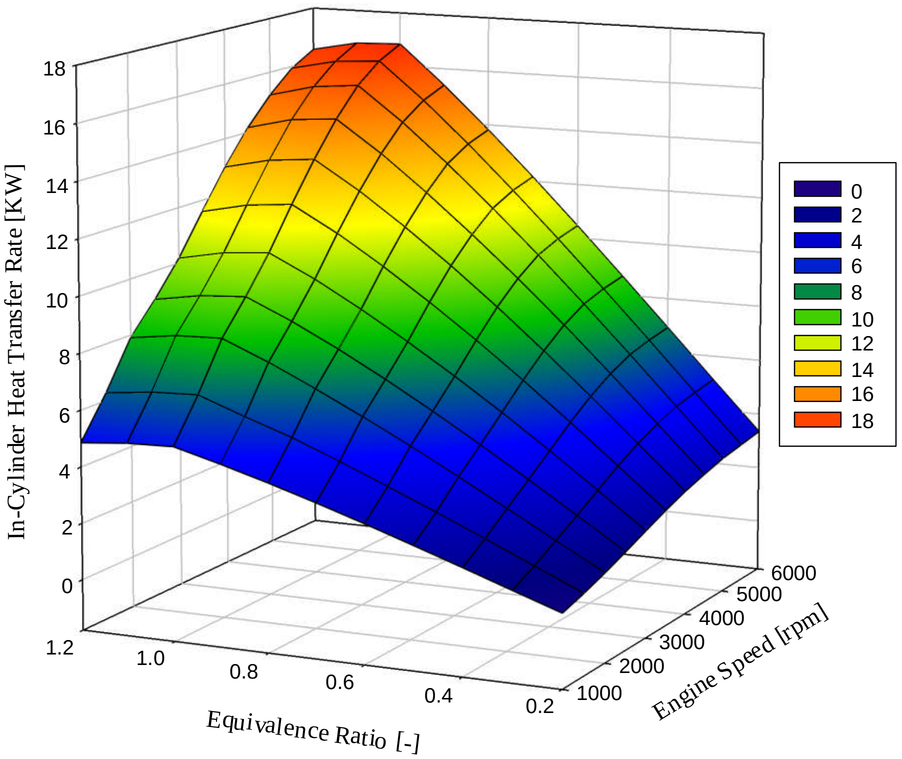


Figure 8: Variation of overall heat transfer rate with engine speed and equivalence ratio

CONCLUSION

The overall characteristics of in-cylinder heat transfer for H_2ICE are numerically carried out by solving the intake, compression, expansion and exhaust strokes. The three operation parameters are considered to evaluate the effect of different parameter combination on the overall heat transfer characteristics in a four-stroke automotive hydrogen fueled engine with a port injection technique. The obtained results show that one-dimensional gas dynamic predictions yield a reasonable understanding for the influence of the engine speed and equivalence ratio on the overall heat transfer characteristics. Due to a wide flammability range for hydrogen fuel, it is unjustifiable to ignore the influence of the equivalence ratio on heat transfer characteristics especially on the heat transfer correlation's form. Therefore the equivalence ratio as a variable in the general correlation form is one of the crucial objectives for the authors' future studies.

ACKNOWLEDGEMENTS

The authors would like to thank Universiti Malaysia Pahang for providing laboratory facilities and financial support under project no. RDU090393 and Doctoral Scholarship scheme (GRS 090121).

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