

## INFLUENCE OF ENGINE SPEED ON HEAT TRANSFER CHARACTERISTICS OF PORT INJECTION HYDROGEN FUELED ENGINE

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### ABSTRACT

This paper presents the numerical investigation of the in-cylinder heat transfer characteristics of port injection hydrogen fueled internal combustion engine. One dimensional gas dynamics was described the flow and heat transfer in the components of the engine model. Special attention is paid to selection and correction of heat transfer which describe of in-cylinder heat transfer to coincide with the practical observations. Engine speed varied from 2000 rpm to 5000 rpm with increment of 500 rpm. The difference between hydrogen and methane revealed in terms of heat transfer rate and the percentage of heat transfer from the total fuel energy are investigated. The acquired results show that hydrogen fueled engine has a higher heat transfer rate with methane because of hydrogen fuel has higher heating value, faster flame speed and small quenching distance. Instantaneous results reveal that the normalized apparent and cumulative heat release is affected by engine speed. The heat transfer rate and coefficient are also affected with engine speed. Beside that the steady state results are presented by examining the dependency of average heat transfer rate and the percentage ratio of heat transfer to the total fuel energy on the engine speed.

**Keywords:** Hydrogen Fuel, Heat Transfer, In-cylinder, Port Injection, Engine Speed.

### 1. INTRODUCTION

In today's modern world, where new technologies are introduced every day, transportation's energy use is increasing rapidly. Fossil fuel is the major contributor to produce energy and the prime fuel for transportation. Rapidly depleting reserves of petroleum and decreasing air quality raise questions about the future. Therefore, alarms about climate contamination and decrease petroleum resources continue to activate research on alternative fuels have been carried out to substitute fossil fuels. With this tendency into account, the use of alternative, renewable fuels and modern vehicle designs has been introduced solution to mitigate these problems. With increasing concern about energy shortage and environmental protection, research on improving engine fuel economy and reducing exhaust emissions has become the major researching aspect in combustion and engine development. Due to limited reserves of crude oil, development of alternative fuel engines has attracted more and more concern in the engine community. Alternative fuels, usually belong to clean fuels compared to diesel fuel and gasoline fuel in the combustion process of engines. The introduction of these alternative fuels is beneficial to slowing down the fuel shortage and reducing engine exhaust emissions. Hydrogen fuel is regarded as one of the most promising alternative fuels for automobiles in future. It is combustion produce no

greenhouse gases, no ozone layer depleting chemicals, and little or no acid rain ingredients and pollution [1]. Hydrogen, produced from renewable energy (solar, wind, biomass, tidal etc.) sources, would result in a permanent energy system which would never have to be changed [2]. Hydrogen Internal Combustion Engines (H<sub>2</sub>ICE) is a technology available today and economically viable in the near-term. This technology demonstrated efficiencies in excess of today's gasoline engines and operate relatively cleanly (NO<sub>x</sub> is the only emission pollutant) [3]. So far, extensive studies were investigated hydrogen fueled internal combustion engines [4-11]. However, most of the previous study regarding the engine performance and combustion process were carried out. Increases efficiencies, high power density and reduce emissions are the main objectives for internal combustion engines (ICE) development [12]. One of the major parameter that effective in the improvement of performance and emission regulation in the ICE is the amount of heat loss from the total heat release during combustion process. Hydrogen can be readily used in spark ignition engines as a clean alternative to fossil fuels. However, the higher burning velocity, higher coefficient of diffusivity, higher thermal conductivity and shorter quenching distance of hydrogen compared with hydrocarbons cause a larger heat transfer from the burning gas to the combustion chamber walls. Because

of this cooling loss, the thermal efficiency of H<sub>2</sub>ICE is sometimes lower than that of conventionally fueled engines. Therefore, reducing the cooling loss is a crucial element in improving the thermal efficiency of hydrogen combustion engines [13]. Based on the analysis of losses that is done by Wimmer et. al. [14]. The authors found that the wall heat transfer has a major influence on overall efficiency in H<sub>2</sub>ICE operation. The significant difference in the physical properties of hydrogen fuel and fossil fuels is the driving force to recognize every behavior for H<sub>2</sub>ICE. A good knowledge about heat transfer characteristics inside the cylinder is very important to accurate descriptions for the heat transfer phenomena inside H<sub>2</sub>ICE. From the above point of view, need to more attention about this matter. The objectives of the present study are to investigate the influence of engine speed on in-cylinder overall heat transfer characteristics for port injection H<sub>2</sub>ICE.

## 2. MATERIALS AND METHODS

### 2.1 Engine Model

A single cylinder, four stroke spark ignition port injection hydrogen fuel was developed in this study. GT-Suite was utilizing the development of engine model. The injection of hydrogen was located in the midway of the intake port. The computational and schematic models of the single cylinder four stroke port injection hydrogen fueled engine are shown in Figures 1 and 2 respectively. The engine specification is listed in Table 1. It is important to indicate that the intake and exhaust ports of the engine cylinder are modeled geometrically with pipes. The air enters through a bell-mouth orifice to the pipe. The discharge coefficients of the bell-mouth orifice were set to 1 to ensure the smooth transition as in the real engine. The pipe of bell-mouth orifice with 0.07 m of diameter and 0.1 m of length are used in this model. The pipe connects in the intake to the air cleaner with 0.16 m of diameter and 0.25 m of length was modeled. The air cleaner pipe identical to the bell-mouth orifice connects to the manifold. A log style manifold was developed from a series of pipes and flow-splits. The total volume for each flow-split was 256 cm<sup>3</sup>. The flow-splits compose from an intake and two discharges. The intake draws air from the preceding flow-split. One discharge supplies air to adjacent intake runner and the other supplies air to the next flow-split. The last discharge pipe was closed with a cup to prevent any flow through it because there is no more flow-split. The flow-splits are connected with each other via pipes with 0.09 m diameter and 0.92 m length. The junctions between the flow-splits and the intake runners were modeled with bell-mouth orifices. The discharge coefficients were also set to 1 to assure smooth transition, because in most manifolds the transition from the manifold to the runners is very smooth. The intake runners for the four cylinders were modeled as four identical pipes with .04 m diameter and 0.1 m length. Finally the intake runners were linked to the intake ports which were modeled as pipes with 0.04 m diameter and 0.08 m length. The overall temperature of the head, piston and cylinder for the engine parts are listed in Table 2. The temperature of the

piston is higher than the cylinder head and cylinder block wall temperature because it is not directly cooled by the cooling liquid. Heat transfer multiplier is used to take into account for bends, additional surface area and turbulence caused by the valve and stem. The pressure losses in these ports are included in the discharge coefficients calculated for the valves. No additional pressure losses due to wall roughness were used.

The exhaust runners were modeled as rounded pipes with 0.03 m inlet diameter, and 80° bending angle for runners 1 and 4; and 40° bending angle of runners 2 and 3. Runners 1 and 4, and runners 2 and 3 are connected before enter in a flow-split with 169.646 cm<sup>3</sup> volume. Conservation of momentum is solved in 3-dimensional flow-splits even though the flow in GT-Power is otherwise based on a one-dimensional version of the Navier-Stokes equation. Finally a pipe with 0.06 m diameter and 0.15 m length connects the last flow-split to the environment. Exhaust system walls temperature was calculated using a model embodied in each pipe and flow-split. Table 3 are listed the parameters used in the exhaust environment of the model.

A simulation of the wall heat transfer is an imperative condition for the accurate analysis of the working process of ICE. The engine model is to estimate the engine heat transfer using Woschni's correlation [15]. The original values of the constant in the correlation were multiplied by factor equal to 1.8, resulting in a better match with the experimental data [16-17]. The authors were found during the analysis that the heat transfer correlation under predicts heat transfer loss.

Table1: Engine specification

Engine Parameter	Value	Unit
Bore	100	mm
Stroke	100	mm
Total displacement	3142	(cm <sup>3</sup> )
Connecting rod length	220	mm
Compression ratio	9.5	
Intake valve close, IVC	-96	°CA
Exhaust valve open, EVO	125	°CA
Inlet valve open, IVO	351	°CA
Exhaust valve close, EVC	398	°CA

Table 2: Engine components temperature

Components	Temperature (K)
Cylinder head	550
Cylinder block wall	450
Piston	590

Table 3: Parameters used in the exhaust environment

Parameters	Value	Unit
External environment temperature	320	K
Heat transfer coefficient	15	W/m <sup>2</sup> K
e temperatureRadiativ	320	K
Wall layer material	Steel	
Layer thickness	3	mm
Emissivity	0.8	

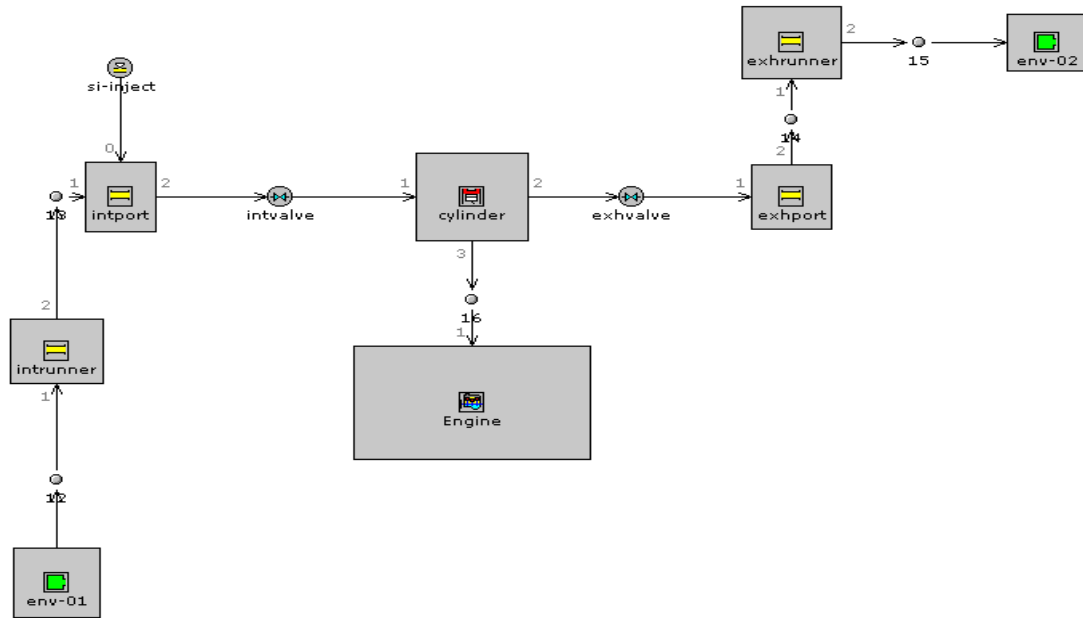


Fig 1. Model of single cylinder, four stroke port injection hydrogen fueled engine

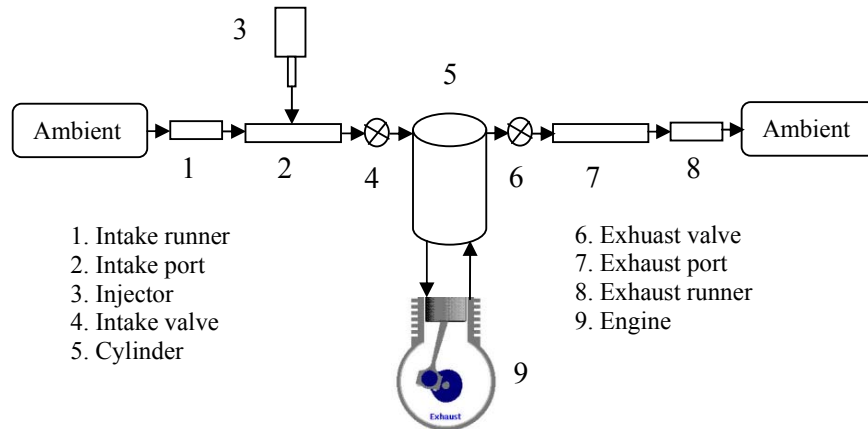


Fig 2. Schematic model of port injection hydrogen fueled engine.

## 2.2 Heat Transfer Governing Equations

One dimensional gas dynamics to represent 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.

The pressure loss coefficient is defined by Equation (1):

$$C_{pl} = \frac{p_1 - p_2}{\frac{1}{2} \rho u_1^2} \quad (1)$$

where  $p_1$  and  $p_2$  are the inlet and outlet pressure

respectively,  $\rho$  the density and  $u_1$  the inlet velocity.

The friction coefficient can be expressed for smooth and rough walls as Equation (2) and (3) respectively:

$$C_f = \frac{16}{Re_D} \quad Re_D < 2000 \quad (2)$$

$$C_f = \frac{0.08}{Re_D^{0.25}} \quad Re_D > 2000$$

$$C_{f(rough)} = \frac{0.25}{\left[ 2 \log \left( \frac{D}{2h} \right) + 1.74 \right]^2} \quad (3)$$

where  $Re$ ,  $D$  and  $h$  are Reynolds number, pipe diameter and roughness height respectively.

The heat transfer from the internal fluids to the pipe and flow split walls is dependent on the heat transfer coefficient, the predicted fluid temperature and the internal wall temperature. The heat transfer coefficient is calculated every time step and it is a function of fluid velocity, thermo-physical properties and wall surface roughness.

The internal wall temperature is defined as Equation (4).

$$h_g = \frac{1}{2} C_f \rho U_{eff} C_p Pr^{-\frac{2}{3}} \quad (4)$$

where  $U_{eff}$ ,  $C_p$  and  $Pr$  are the effective speed outside boundary layer, specific heat and Prandtl number respectively. The Prandtl number can be expressed as Equation (5)

$$Pr = \frac{\mu C_p}{k} = \frac{\nu}{\alpha} \quad (5)$$

where  $\mu$ ,  $k$ ,  $\nu$  and  $\alpha$  are the dynamic viscosity, conduction heat transfer coefficient, kinematic viscosity and thermal diffusivity respectively.

The heat transfer coefficient depends on characteristic length, transport properties, pressure, temperature and characteristic velocity. There is a wealth of heat transfer correlations for describing heat transfer process inside combustion chamber such as Eichelberg's equation [18], Woschni's equation [15] and Annand's equation [19]. The in-cylinder heat transfer is calculated by a formula which closely emulates the classical Woschni correlation. A unique feature of Woschni correlation is the gas velocity term while most of the other correlations use a time averaged gas velocity proportional to the mean piston speed, Woschni separated the gas velocity into two parts: the unfired gas velocity that is proportional to the mean piston speed, and the time-dependent, combustion induced gas velocity that is a function of the difference between the motoring and firing pressures. The heat transfer coefficient can be expressed as Equation (6):

$$h = 3.26 D^{-0.2} P^{0.8} T_g^{0.55} w^{0.8} \quad (6)$$

$$w = 2.28 C_m + 0.00324 \frac{(P - P_m) V_h T_r}{Pr V_r}$$

where  $D$ ,  $P$ ,  $P_m$ ,  $T_g$ ,  $V_h$ ,  $C_m$ ,  $V$  and  $r$  are the bore diameter, pressure, motored pressure, gas temperature, volume, mean piston speed, swept volume and a reference crank angle respectively.

This approach keeps the velocity constant during the unfired period of the cycle and then imposes a steep velocity rise once combustion pressure departs from motoring pressure.

### 3. RESULTS AND DISCUSSION

It has been adequately emphasized that hydrogen fuel possesses some properties which are uniquely different

from the corresponding properties of conventional hydrocarbon fuels. This was primarily the reason why initially the research and development work on. It has usually been found that the first two conditions occur under heavy load conditions. These often cause the engine to stop. The symptoms of unsteady combustion the most pronounced effect in an internal combustion engine. Hydrogen has long since been attempted as a fuel for the internal combustion engine.

#### 3.1 Steady Heat Transfer Characteristics

Steady state heat transfer analysis was carried out for in-cylinder of four stroke port injection spark ignition hydrogen fueled engine model. The engine speed varied from 2000 rpm to 5000 rpm with change step equal to 500 rpm and crank angle varied from  $-40^\circ$  to  $100^\circ$ . The stoichiometric limit (AFR = 34.33) based on mass where the equivalence ratio ( $\phi = 1.0$ ) was considered the air fuel mixture throughout the study. In order to recognize the difference between the in-cylinder heat transfers characteristics for ICE with hydrogen and hydrocarbon fuels, a direct comparison was verify the engine model in terms of heat transfer rate and ratio of heat transfer to total fuel energy which are as shown in Figures 3 and 4 respectively. It can be seen that the increase of heat transfer rate with increases of engine speed for both fuel (Figure 3) and ratio of heat transfer to total fuel energy in percentage decreases with increases of engine speed (Figure 4). It is also clearly seen that the heat transfer rate through the cylinder to the ambient for hydrogen fuel higher than methane for both due to the higher heating value, faster flame speed and small quenching distance for hydrogen. The ratio of heat transfer rate to the total fuel energy is used as another indicator for clarifying that hydrogen fuel gives amount of heat loss more than methane even as a percentage from the total energy supplied by the fuel as shown in Figure 4.

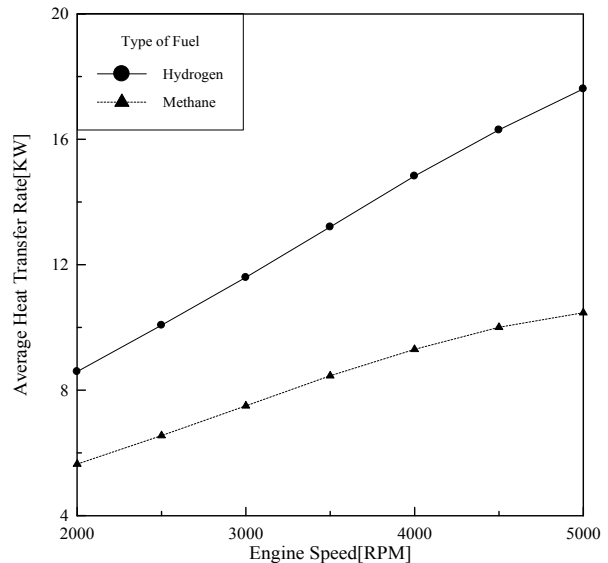


Fig 3. Comparison between hydrogen and methane in term of heat transfer rate for equivalence ratio  $\phi=1.0$ .

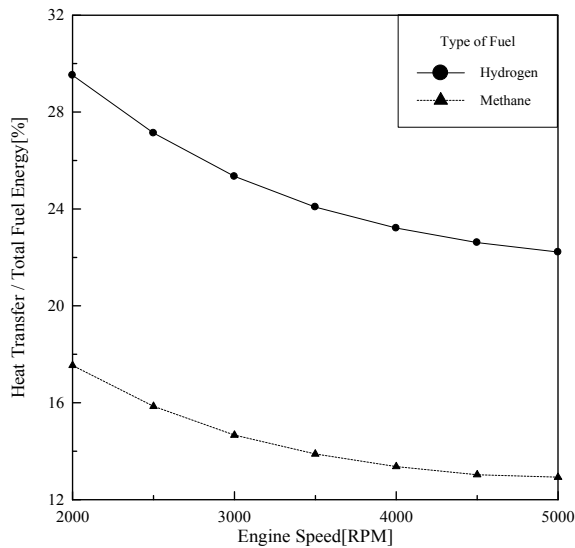


Fig 4. Comparison between hydrogen and methane in term of percentage ratio of heat transfer to total fuel energy

Both of the normalized apparent and cumulative heat release rates are used as indicators for recognizing the in-cylinder heat transfer characteristics for port injection H<sub>2</sub>ICE. Effects of normalized apparent and cumulative heat release rate on engine speed are presented in Figures 5 and 6 respectively. These figures reveal that the dependency of the normalized heat release rate on engine speed are negligible effects because of its related to the amount of energy content in the inlet charge to combustion chamber. The maximum heat release rate decreases with increases of engine speed.

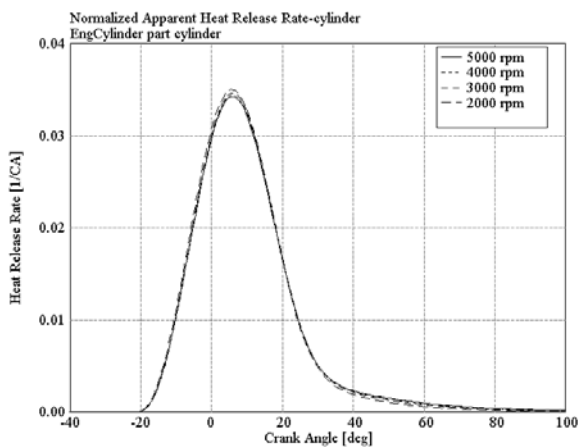


Fig 5. Variation of normalized apparent heat release rate with engine speed

### 3.2 Transient Heat Transfer Characteristics

Transient heat transfer analysis was carried out for in-cylinder of four stroke port injection spark ignition hydrogen fueled engine model. The engine speed varied from 2000 rpm to 5000 rpm with change step equal to 500 rpm and crank angle varied from -180° to 540°. The

air-fuel ratio was varied from rich (stoichiometric) limit (AFR = 34.11:1 based on mass where the equivalence ratio,  $\phi = 1.0$ ) to a very lean limit (AFR =171.65 where ( $\phi = 0.2$ ). The instantaneous behaviors for heat transfer rate and coefficient are employed the in-cylinder heat transfer characteristics for port injection H<sub>2</sub>ICE. Figure 7 shows the instantaneous heat transfer rate from the in-cylinder gases to cylinder walls with various engine speed conditions at AFR=34.33:1 (or equivalence ratio  $\phi=1.0$ ). The obtained results can be seen that the heat transfer rate increases with increases of the engine speed during part of compression stroke, power stroke and most of the exhaust stroke due to increase the turbulence intensity inside the cylinder and lead to increase the forced convection effect which is represent as the main driving force for the heat transfer process between the in-cylinder gases and walls surface. Instantaneous variation of heat transfer coefficient with engine speed at AFR=34.33:1 (Equivalence ratio  $\phi = 1.0$ ) is illustrated in Figure 8. As expected, heat transfer coefficient increased by increasing of engine speed for all cycle strokes where the heat transfer coefficient depends on the forced convection phenomenon. Thus as engine speed increase the effect of forced convection due to increase the in-cylinder gas velocity.

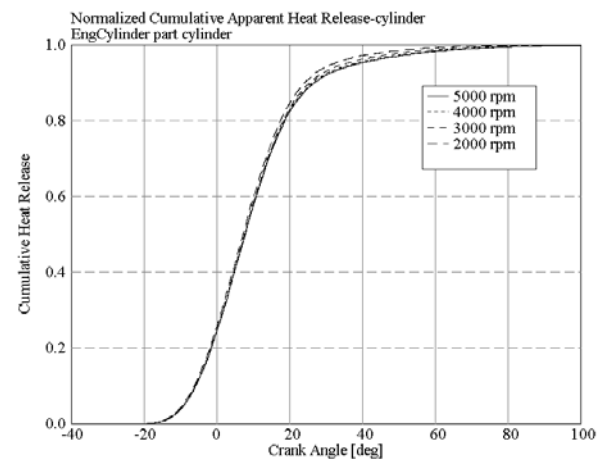


Fig 6. Variation of normalized cumulative heat release rate with engine speed for equivalence ratio ( $\phi$ ) = 1.0

Beside the instantaneous behavior the average heat transfer rate and the percentage ratio for heat transfer to the total fuel energy are used as indicators for the in-cylinder heat transfer characteristics for port injection H<sub>2</sub>ICE. Variation of average in-cylinder heat transfer rate with engine speed is clarified in Figure 9. The average value of heat transfer rate increases as engine speed increases for all AFR values due to increase the driving force (forced convection) for the heat transfer inside the cylinder. While the average value of heat transfer rate increase as AFR decrease for all engine speed and increment in the average heat transfer rate with increasing of the engine speed because of decreasing the energy content for the inlet charge to the cylinder. Variation of the percentage ratio of heat transfer to the total fuel energy with engine speed is presented in figure 10. The ratio of heat transfer rate to total fuel energy

decreases with increases of both engine speed and AFR except AFR=171.65 which has different behavior. For AFR of 171.65, the percentage ratio decreases as engine speed increases until 3500 rpm after that the ratio increases due to the limitation in the energy content for charge interred to the cylinder and continuous increment in the heat transfer rate from the in-cylinder gas to the walls.

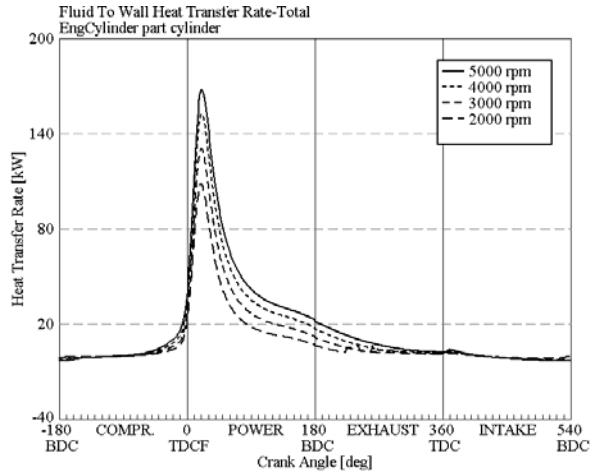


Fig 7. Variation of heat transfer rate with engine speed for equivalence ratio  $\phi=1.0$ .

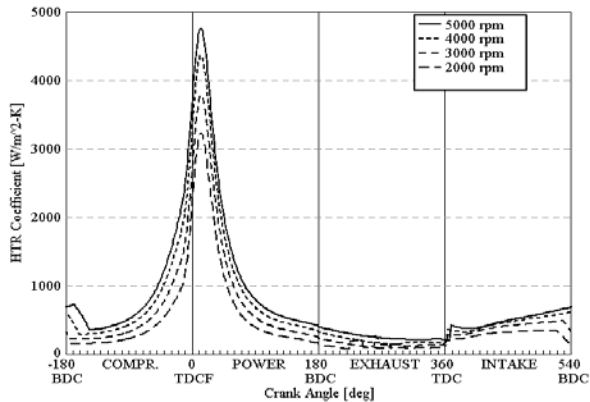


Fig 8. Effect of heat transfer coefficient with engine speed for Equivalence ratio ( $\phi$ )=1.0

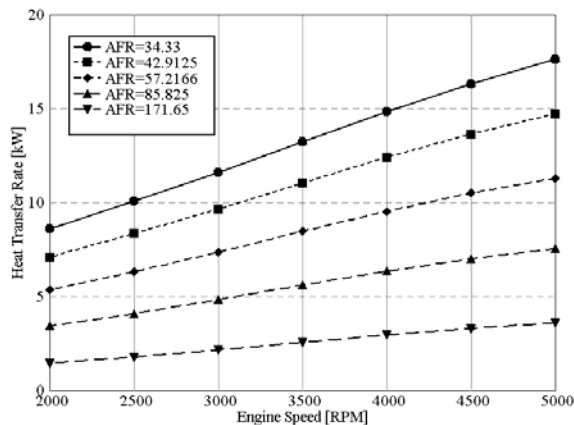


Fig 9. Variation of the average heat transfer rate against engine speed.

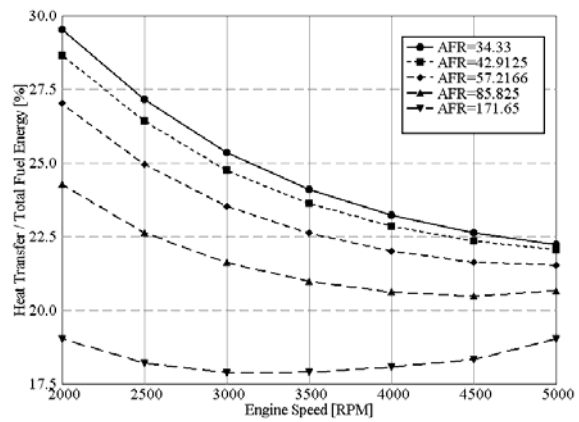


Fig 10. Variation of ratio of heat transfer to total fuel energy with engine speed.

#### 4. CONCLUSIONS

The in-cylinder characteristics of transient and steady states heat transfer process for port injection H<sub>2</sub>ICE have been investigated and quantified. Compression for in-cylinder characteristics between hydrogen and methane fuel in term of heat transfer rate and the percentage ratio for the heat transfer rate to the total fuel energy is clarified. Hydrogen has higher values for the heat transfer rate and the percentage ratio for the heat transfer rate to the total fuel energy due to the difference in physical properties and combustion behavior between of hydrogen and methane. The foregoing results indicate that the dependency both of the normalized apparent and cumulative heat release on engine speed. While for the heat transfer rate and heat transfer coefficient dependency on both of the AFR and engine speed is found. The results show that the average heat transfer rate increases as engine speed increases. As well as the percentage ratio of heat transfer to the total fuel energy variation indicate that decrease with increase of the engine speed.

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