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Parameters Affecting Heat Transfer Rate in the Hydrogen Fueled Engine

Dr Kailash Mohapatra[1], Abhaya Kumar Baliarsingh[2],Bighnesh Sahu[3]
[1] *Professor, Department of Mechanical Engineering, Gandhi Institute For Technology (GIFT), Bhubaneswar*
[2]*Assistant Professor, Department of Mechanical Engineering, Gandhi Institute For Technology (GIFT), Bhubaneswar*
[3] *Assistant Professor, Department of Mechanical Engineering, Gandhi Engineering College, Bhubaneswar*

Abstract: *In this work, effect of the inlet conditions for the intake charge on the in-cylinder heat transfer characteristics for port injection Hydrogen Fueled Engine H_2ICE are investigated through steady state simulation. One-dimensional gas dynamics was used to describe the flow and heat transfer in the components of the engine model. Firstly a brief demonstration for the model description was inserted; followed by the model governing equations. The engine model is simulated with variable engine speed and AFR with influence of the variation of intake charge conditions (pressure and temperature). Engine speed varied from 2000 rpm to 5000 rpm with increments equal to 1000 rpm, while AFR changed from stoichiometric to lean limit. The inlet pressure is varied from 0.95 bar to 1.05 bar with 0.05 interval and the inlet temperature varied from 290 to 310 with 10 interval. The combined effects for the intake charge conditions with variation of AFR and the engine speed on the in-cylinder heat transfer characteristics for port injection H_2ICE are presented in this paper. The baseline engine model is verified with existing previous published result. The results show that the heat transfer characteristics to be more affected by changes in the intake pressure than in the temperature. It is also found that the effect of change for the intake charge pressure disappeared for lean mixture. Beside that the acquired results are presented by examining the dependency of in-cylinder heat transfer rate on the engine speed and AFR.*
Keywords: *heat transfer, hydrogen fueled engine, intake conditions, port injection.*

I. Introduction

As a result of the developments in the modern era, where new technologies are introduced every day, transportation’s energy use increases rapidly. Fossil fuel particularly petroleum fuel is the major contributor to energy production and the prime fuel for transportation. Rapidly depleting reserves of petroleum and decreasing air quality raise questions about the future. Due to limited reserves of crude oil, development of alternative fuel engines has attracted more and more concern in the engine community. The introduction of alternative fuels is beneficial to help alleviate the fuel shortage and reduce engine exhaust emissions (Huang et al. 2006; Saravanan et al. 2007). One of the alternative energy is hydrogen. Hydrogen, as alternative fuel, has unique properties of significant advantage over other types of fuel. 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. Extensive studies were investigated on hydrogen fueled internal combustion engines (Kahraman, et al. 2007; Rahman et al. 2009; Stockhausen et al. 2000). With increasing concern about the energy shortage and environmental protection, research on improving engine fuel economy, hydrogen fueled engine is being developed into a hydrogen fueled engine with different type of fuel supply method (Eichlseder, et al. 2003; Kim, et al. 2005; Ganesh,, et al. 2008).

It is well known that the performance of an engine is influenced by the intake charge conditions. The most important intake conditions affecting gas engine performance are the intake pressure and temperature (Soares and Sodre, 2002; Sodre and Soares, 2003). But the effect of the conditions for the intake charge on the in- cylinder heat transfer is not well recognized. The aim of the research work presented in this paper is to assess the potential of inlet charge conditions (temperature and pressure) for in-cylinder heat transfer reduction of port injection H_2ICE.

II. Materials And Methods

A single cylinder port injection hydrogen fuel model was developed utilizing the GT- suite software. The injection of hydrogen was studied in the midway of the intake port. The computational model of single cylinder hydrogen fueled engine is shown in Fig. 1. The engine specifications used to make the model (A) are listed in Table 1. The intake and exhaust ports of the engine cylinder are modeled geometrically with pipes and 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 smooth transition. The intake runners were linked to the intake ports with 0.04 m diameter and

0.08 m length. The temperature of the piston is higher than the cylinder head and cylinder block wall temperature. 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 are included in the discharge coefficients calculated for the valves but no additional pressure losses were used for wall roughness. The exhaust port was modeled as rounded pipe with 0.04 m inlet diameter and 0.8 m length. Exhaust wall temperature was calculated using a model embodied in each pipe.

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 adopting the Woschni's correlation (Woschni, 1967) for the in-cylinder heat transfer calculation. 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 (Aceves and Smith, 1997). The authors found during the analysis that the heat transfer correlation under predicts heat transfer loss.

Table 1: Engine specifications for model A.

Parameter	Value	Unit
Bore	100	mm
Stroke	100	mm
Connecting rod length	220	mm
Compression ratio	9.5	-
Inlet valve open	9	CA(BTDC)
Exhaust valve open	55	CA(BBDC)
Inlet valve close	84	CA(ABDC)
Exhaust valve close	38	CA(ATDC)
No. of cylinder	1	-

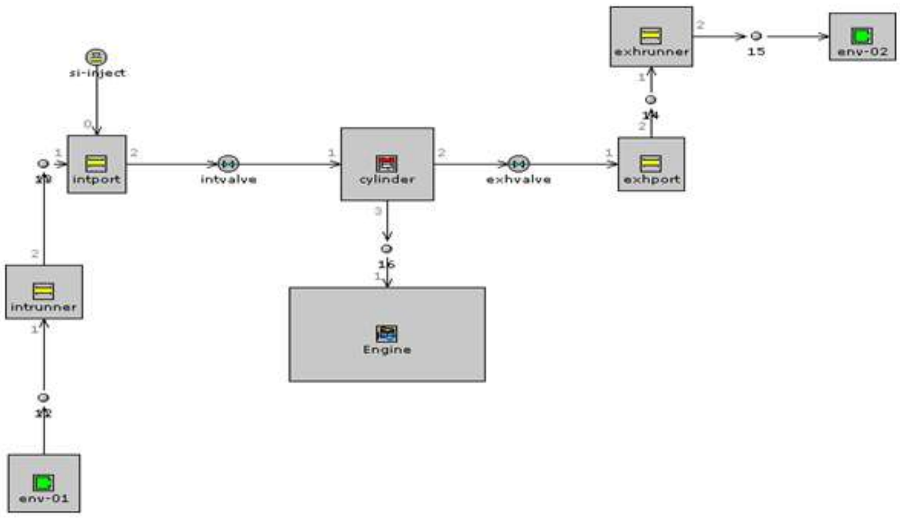

Figure 1: Model of single cylinder four stroke,port injection hydrogen fueled engine

Heat Transfer Modeling Equations

One-dimensional gas dynamics model is used 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. To complete the simulation model, other additional formulas beside of the main governing equations are used for calculations of the pressure loss coefficient, friction coefficient, and heat transfer.
The pressure loss coefficient is defined by:

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

where p_1 and p_2 are the inlet and outlet pressure respectively, ρ charge density and u_1 the inlet velocity.

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

$$C_f = \frac{16}{\mathrm{Re}_D} \qquad \mathrm{Re}_D < 2000; \;\; \mathrm{Re}_D = \frac{vD}{\nu}$$
$$C_f = \frac{0.08}{\mathrm{Re}_D^{0.25}} \qquad \mathrm{Re}_D > 4000 \tag{2}$$

$$C_{f(\text{rough})} = \frac{0.25}{\left(2\log\left(\frac{D}{z}\right) + 1.74\right)^2} \tag{3}$$

where Re_D, D and z are Reynolds number, pipe diameter and roughness height respectively.

The amount of heat rate which is transferred from the in-cylinder hot gases to its walls calculates according to the formula of Newton's law of cooling:

$$Q = hA(T_g - T_w) \tag{4}$$

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 (Eichelberg 1939), Woschni's equation (Woschni 1967) and Annand's equation (Annand 1963). 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 (5):

$$h = 3.26 D^{-0.2} P^{0.8} T_g^{-0.55} w^{0.8}$$
$$w = 2.28 C_m + 0.00324 \frac{(P - P_m) V_h T_r}{P_r V_r} \tag{5}$$

where D, P, P_m, T_g, V, C_m, V_h 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. This empirical equation was derived for hydrocarbon combustion engines and it was based on observations using the turbulent heat transfer equation for tubes. A realistic simulation of the wall heat transfer is an imperative condition for the accurate analysis of the working process of ICE. So, for the hydrogen fuel engines should be correct choice for the formula which gives the best guess for the amount of heat transfer from the combustion chamber gas to its walls. The engine model for (Aceves and Smith 1997) estimate engine heat transfer by using Woschni's correlation (Woschni 1967). It was found during the analysis that the heat transfer correlation under predicts heat transfer loss. Therefore, for the present model 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 according to (Aceves and Smith 1997).

III. Results And Discussion

Steady state gas flow and heat transfer simulations for the in-cylinder of four stroke port injection spark ignition hydrogen fueled engine model is running for two operation parameters namely Air-Fuel Ratio (AFR) and engine speed with influence of the variation of inlet conditions (pressure and temperature). The Air-Fuel Ratio (AFR) was varied from stoichiometric limit (AFR = 34.33:1 based on mass) to very lean limit (AFR=171.65) and engine speed was varied 2000-5000 rpm with 1000 rpm interval. As well as the inlet pressure varied from 0.95 bar to 1.05 bar with 0.05 interval and the inlet temperature varied from 290 to 310 with 10 interval.

Model Validation

The experimental results obtained from (Lee et al., 1995) were used for purpose of initial validation of this study. Engine specifications of (Lee et al., 1995) and present single cylinder port injection engine model (B) are listed in Table 2. The same engine model which described in Figure 1 was used for the purpose of this validation (taking into account the difference in the engines dimensions). Engine speed and AFR were fixed at 1500 rpm and 68.66 respectively in this comparison to coincide with Lee et al., results.

The in-cylinder pressure traces for the baseline model (B) and experimental published results (Lee et al., 1995) are shown in Figure 2. It can be seen that in-cylinder pressure trace have a good match during compression stoke and acceptable trends during expansion stroke while large deviation observed during combustion period due to the delay in the combustion as claimed by Lee et al., beside the difference between the some engine configuration conditions that is not mentioned in (Lee et al., 1995). However, considerable coincident between the present model (B) and experimental results can be recognized in spite of the mentioned model differences. To demonstrate the effectiveness of the adopted model for the present study model (A), direct comparison with model (B) in term of in-cylinder pressure traces was done as shown in
Figure 3. The difference between the models is due to the difference in dimensions and compression ratio.

Table 2: Specifications of the engines models.

Engine Parameter	Lee et al., (1995)	Present Model B	Unit
Bore	85	85	mm
Stroke	86	86	mm
TDC clearance height	NA*	3	mm
Piston pin offset	NA	1	mm
Connecting rod length	NA	150	mm
Compression ratio	8.5	8.5	-
Inlet valve open	16	16	^{0}CA(BTDC)
Exhaust valve open	52	52	^{0}CA(BBDC)
Inlet valve close	54	54	^{0}CA(ABDC)
Exhaust valve close	12	12	^{0}CA(ATDC)

* NA=not available.

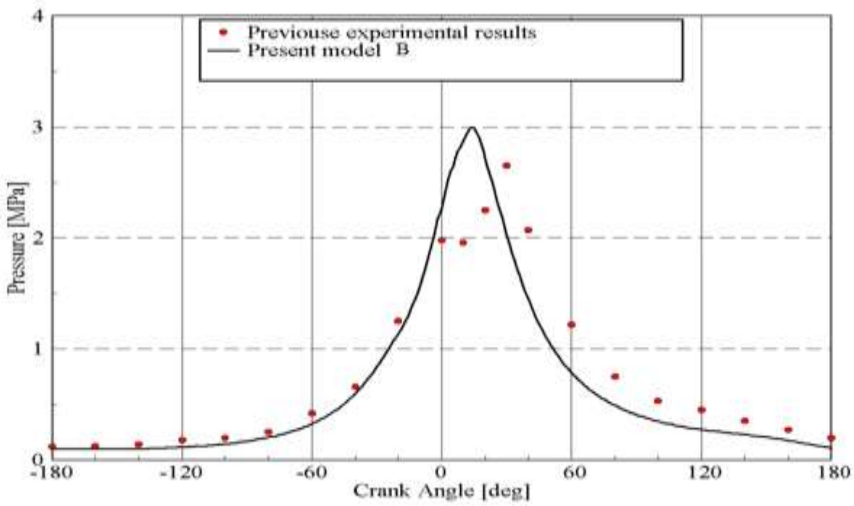

Figure 2: Comparison between published experimental results Lee et al. (1995) and present single cylinder port injection engine model based on in-cylinder pressure traces.

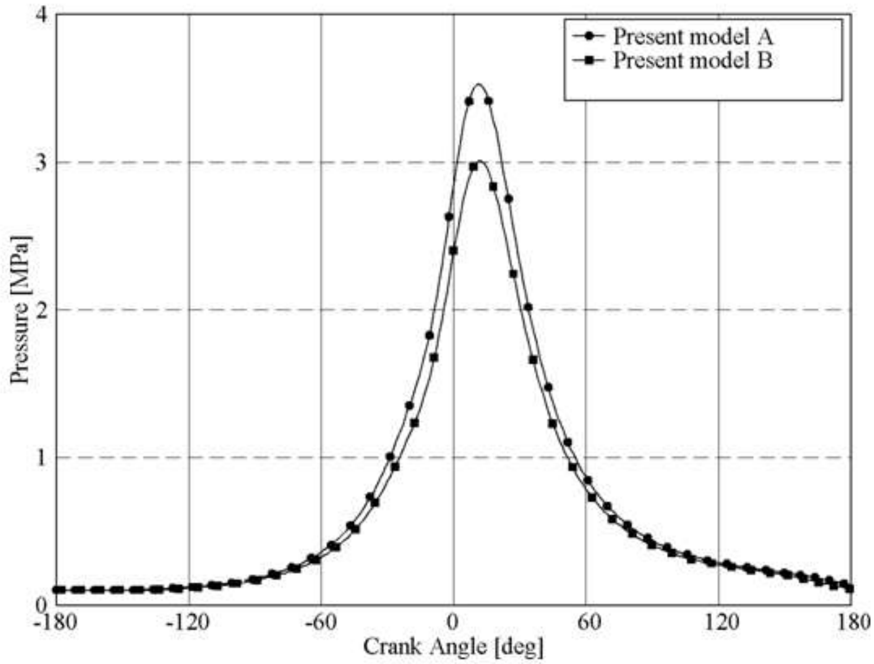

Figure 3: Comparison between models (A and B) based on in-cylinder pressure traces.

Effect of Intake Charge Pressure on the Heat Transfer Rate

Effect of the inlet pressure on the in-cylinder heat transfer rate at different engine speed is shown in Figure 4. It can be seen that the in-cylinder heat transfer rate increases with increases of inlet pressure for the intake charge for all engine speed. At high engine speed the effects of pressure for the intake charge are more pronounced, where the increment in heat transfer rate is increasing with increase of the engine speed for all inlet pressure values. Figure 5 shows the effect of intake charge pressure variation on the in-cylinder heat transfer rate with different AFR values. As intake charge pressure increase the in-cylinder heat transfer increases for all AFR values. However, it also showed decrease in increment trend by moving from the stoichiometric to lean limits.

It can be seen that the heat transfer rate has increased with increasing engine speed due to increasing the driving force (forced convection) for the heat transfer inside the cylinder. While decreased by increasing AFR because of decreasing in the energy content for the inlet charge to the cylinder. The observation of the heat transfer rate behavior through the cylinder to ambient revealed that in case of hydrogen fuel gives higher values than that of hydrocarbon fuels due to the higher heating value, faster flame speed and small quenching distance. This can be used as an indicator for clarifying that the hydrogen fuel gives more heat loss compared to hydrocarbon fuel.

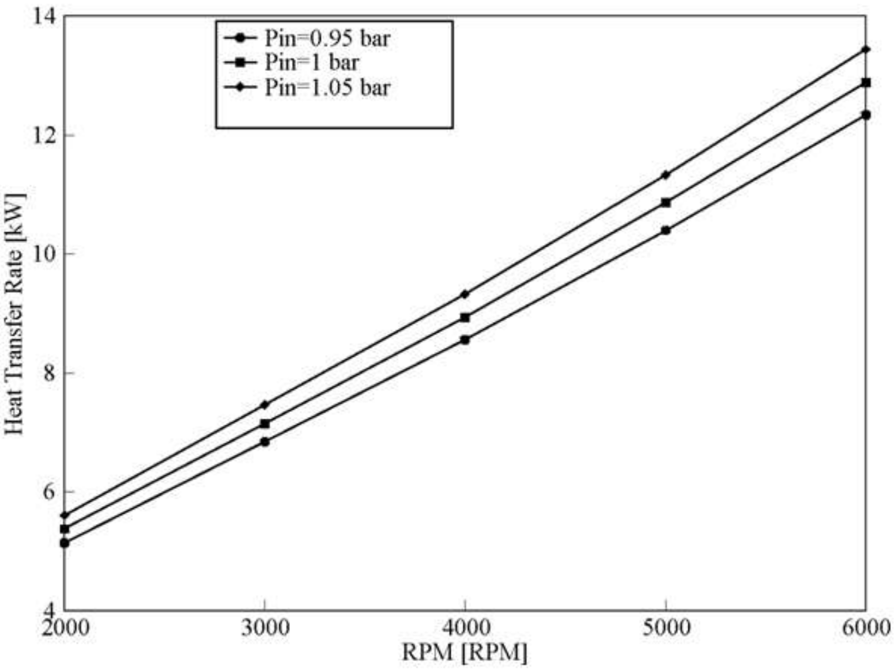

Figure 4: Variation of in-cylinder heat transfer rate with engine speed and variable intake charge pressure.

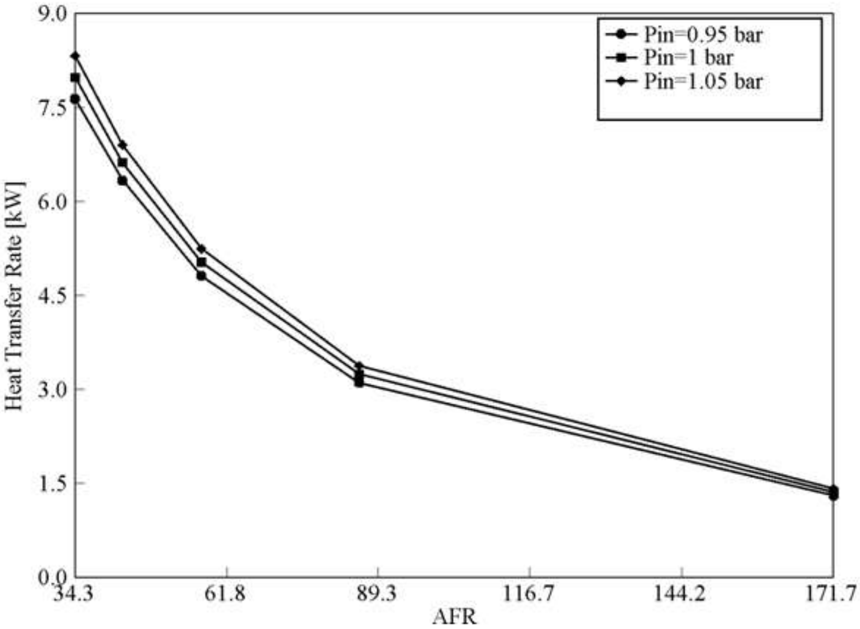

Figure 5: Variation of in-cylinder heat transfer rate with AFR and variable intake charge pressure.

Effect of Intake Charge Temperature on the Heat Transfer Rate

Variation of in-cylinder heat transfer rate with engine speed for different intake charge temperature is revealed in Figure 6. It appeared that negligible effect for the intake charge temperature on the in-cylinder heat transfer rate, especially at lower engine speed values. The combined effect for AFR and intake charge temperature on the in-cylinder heat transfer rate is demonstrated in Figure 7. There is no impact of the intake charge temperature on the behavior of in-cylinder heat transfer rate with AFR variation.

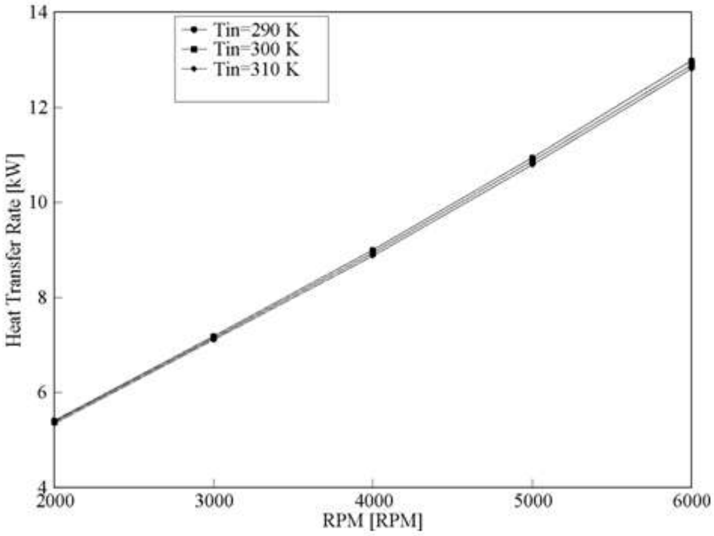

Figure 6: Variation of in-cylinder heat transfer rate with engine speed and variable intake charge temperature.

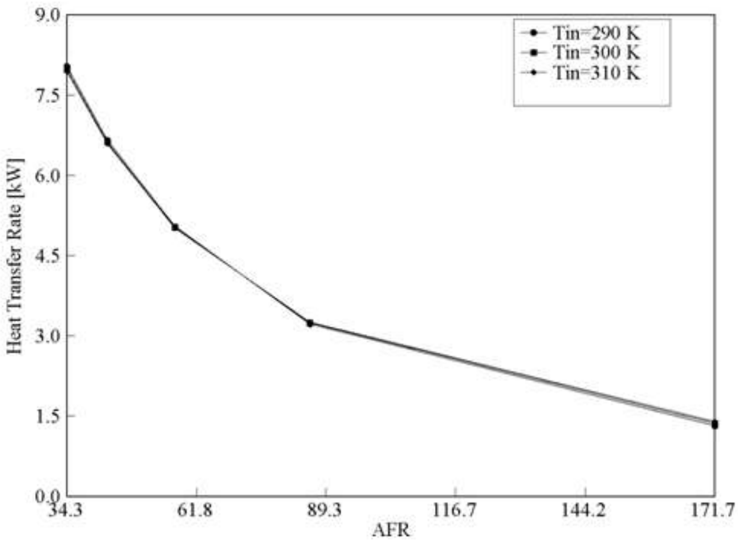

Figure 7: Variation of in-cylinder heat transfer rate with AFR and variable intake charge temperature.

IV. Conclusion

The combined influence of the engine speed and AFR with intake charge conditions (pressure and temperature) on the in-cylinder heat transfer characteristics for port injection H_2ICE have been investigated and quantified. The results show that increasing the pressure for intake charge gives negative impact on the in-cylinder heat transfer rate with engine speed and AFR variation. There is no impact of the intake charge temperature on the behavior of in-cylinder heat transfer rate with AFR variation and negligible effect with engine speed variation. Beside that the acquired results are presented by examining the dependency of in-cylinder heat transfer rate on the engine speed and AFR.

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