EFFECT OF ENGINE SPEED ON PERFORMANCE OF FOUR-CYLINDER DIRECT INJECTION HYDROGEN FUELED ENGINE

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ABSTRACT

This paper presents the effect of engine speed on the engine performance of 4-cylinder direct injection (DI) hydrogen fueled engine. The 4-cylinder direct injection hydrogen engine model was developed utilizing the GT-Power commercial software. This model employed one dimensional gas dynamics to represent the flow and heat transfer in the components of engine model. Sequential pulse injectors are adopted to inject hydrogen gas fuel within the compression stroke. Injection timing was varied from 110° before top dead center (BTDC) until top dead center (TDC) timing. Engine speed was varied from 2000 rpm to 6000 rpm. The validation was performed with the existing previous experimental results. The negative effects of the interaction between ignition timing and injection duration was highlighted and clarified. The acquired results show that the engine speeds are strongly influence on the optimum injection timing and engine performance. It can be seen that the indicated efficiency increases with decreases of engine speed; power increases with the decreases of engine speed; indicated specific fuel consumption (ISFC) increases with increases of engine speed. The injection timing of 60° BTDC was the overall optimum injection timing with a compromise.


1. INTRODUCTION

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 manifold injection, direct injection or duel injection according to the fuel supply method [1-3]. Of course, the hydrogen fueled engine with direct injection can fundamentally keep backfires from occurring so it can be utilized as a high powered hydrogen power system if the reliability of high pressure direct injection valve is secured [4]. In today’s modern world, where new technologies are introduced every day, transportation’s energy use is increasing rapidly. Fossil fuel particularly petroleum fuel is the major contributor to energy production and the primary fuel for transportation. Rapidly depleting reserves of petroleum and decreasing air quality raise questions about the future. As world awareness about environment protection increases so does the search for alternative to petroleum fuels. 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. So far, an extensive study investigated hydrogen fueled internal combustion engines (H2ICE) with external mixture formation fuel delivery system [5-6]. However, the operation of these engines subjected to abnormal combustion, such as pre-ignition, backfire and knocking. Moreover, the power outputs of these hydrogen engines are about 30% less than those of gasoline engines [7]. Therefore the premixed-charge spark ignition engines fueled with hydrogen can be used for significantly limited operation range [8].

Injection timing plays a critical role in the phasing of the combustion, and hence the emissions and torque production. Therefore, extensive number of studies indicated the significance of optimization for ignition timing [9-11]. White et al. [10] suggested that late injection can minimize the residence time that
a combustible mixture is exposed to in-cylinder hot spots and allow for improved mixing of the intake air with the residual gases. This selection can control pre-ignition problem. The main challenge for selecting the proper ignition timing that is in-cylinder injection requires hydrogen–air mixing in a very short time. This study attempts to optimize injection timing that gives the best performance of a 4-cylinders direct injection. The 4-cylinder direct injection hydrogen fueled engine model is developed for this purpose. The effects of engine speed on the injection timing and engine performance such as indicated efficiency, indicated specific fuel consumption, power and torque for direct injection hydrogen fueled engine.

2. MODEL DESCRIPTION

The engine model for an in-line 4-cylinder direct injection engine was developed for this study. Engine specifications for the base engine are tabulated in Table 1. The specific values of input parameters including the AFR, engine speed, and injection timing were defined in the model. The boundary condition of the intake air was defined first in the entrance of the engine. 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$^3$. The flow-splits compose from an intake and two discharges. The intake draws air from the preceding 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 intake runners for the four cylinders were modeled as four identical pipes with 0.04 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 length. The air mass flow rate in the intake port was used for hydrogen flow rate based on the imposed AFR.

Table 1. Engine specification

<table>
<thead>
<tr>
<th>Engine Parameter</th>
<th>Value</th>
<th>Unit</th>
<th>Engine Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>100</td>
<td>mm</td>
<td>Compression ratio</td>
<td>9.5</td>
<td></td>
</tr>
<tr>
<td>Stroke</td>
<td>100</td>
<td>mm</td>
<td>Inlet valve close, IVC</td>
<td>-96</td>
<td>°CA</td>
</tr>
<tr>
<td>Connecting rod length</td>
<td>220</td>
<td>mm</td>
<td>Exhaust valve open, EVO</td>
<td>125</td>
<td>°CA</td>
</tr>
<tr>
<td>Piston pin offset</td>
<td>1.00</td>
<td>mm</td>
<td>Inlet valve open, IVO</td>
<td>351</td>
<td>°CA</td>
</tr>
<tr>
<td>Total displacement</td>
<td>3142</td>
<td>(cm$^3$)</td>
<td>Exhaust valve close, EVC</td>
<td>398</td>
<td>°CA</td>
</tr>
</tbody>
</table>

In the powertrain, the induced air passes through the intake cam-driven type valves with 45.5 mm of diameter to the cylinders. The valve lash (mechanical clearance between the cam lobe and the valve stem) was set to 0.1 mm. 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 this part is not directly cooled by the cooling liquid or oil. 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$^3$ volume. Conservation of momentum is solved in 3-dimentional 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. Figure 1 shows the entire model of 4-cylinder direct injection engine.
Table 2. Temperature of the main engine parts

<table>
<thead>
<tr>
<th>Components</th>
<th>Temperature (K)</th>
</tr>
</thead>
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<tr>
<td>Cylinder head</td>
<td>550</td>
</tr>
<tr>
<td>Cylinder block wall</td>
<td>450</td>
</tr>
<tr>
<td>Piston</td>
<td>590</td>
</tr>
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</table>

Table 3. Parameters used in the exhaust environment

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>External environment temperature</td>
<td>320</td>
<td>K</td>
</tr>
<tr>
<td>Heat transfer coefficient</td>
<td>15</td>
<td>W/m²K</td>
</tr>
<tr>
<td>Radiative temperature</td>
<td>320</td>
<td>K</td>
</tr>
<tr>
<td>Wall layer material</td>
<td>Steel</td>
<td></td>
</tr>
<tr>
<td>Layer thickness</td>
<td>3</td>
<td>mm</td>
</tr>
<tr>
<td>Emissivity</td>
<td>0.8</td>
<td></td>
</tr>
</tbody>
</table>

3. RESULTS AND DISCUSSION

The results in the following section show the engine performance behavior with injection timing for each condition under investigation. In order to check the validity and accuracy of the present model, comparison with published experimental results in the literature. The effect of engine speed with injection timing on the engine performance parameters including indicated efficiency, brake specific fuel consumption, power, and torque were discussed. In the present model, hydrogen was injected into the cylinder within a timing range started just before IVC (−96° BTDC) until TDC (0°). Amount of hydrogen injected in one cycle is approximately 22 mg/cycle with injection pulse duration of 4.4 ms. Engine speed was varied from 2000 rpm to 6000 rpm. Stoichiometric condition was fixed throughout the investigation.

The experimental results obtained from Mohammadi et al. [11] were used for the purpose of validation in this study. For the purpose of validation, single cylinder direct injection engine model converted to 4-cylinder direct injection model. The correlation of brake thermal efficiency of the baseline model and experimental results obtained from Mohammadi et al. [11] is shown in Fig. 2. It can be seen that the brake thermal efficiency are good match with the experimental results. Only small deviation was obtained due to the difference between the engine operation conditions that are not mentioned in Mohammadi et al. (2007). However, considerable coincident between the single cylinder model and experimental results can be recognized in spite of the mentioned model differences.
Figure 3 shows the variation of indicated thermal efficiency with the injection timing for the changes of engine speed. It can be seen that the indicated efficiency increases with decreases of engine speed. From the acquired results, indicated efficiency increases slightly with advances of injection timing towards TDC for all engine speed range. It is also seen that the slightly increase of indicated efficiency until about 30° BTDC for 2000 rpm then it drops down. The rate of change in indicated efficiency is higher for higher speed and drop occurs early in higher speeds. For very high speeds, the drop happens earlier due to the early interaction between the injection duration and ignition timing. Optimum injection timing under speeds from 2000 rpm to 5000 rpm was in the range (40°-80°) BTDC while the optimum injection timing for 6000 rpm was 100° BTDC. Obviously, engine speed has a strong contribution in specifying the optimum injection timing. The very limited acceptable injection timing range occurs for high speeds. The selection of the proper injection timing is crucial not only for performance aspects, but also for stable operation. The variation of engine speed on the indicated efficiency is shown in Fig. 4 for stoichiometric operation and injection timing of 100° BTDC. It can be seen that the maximum indicated efficiency is 38.55% corresponding to engine speed 2500 rpm. This variation of indicated efficiency is primarily due to the variation of the volumetric efficiency.

Figure 5 shows the influence of injection timing on ISFC for different engine speeds. Lower engine speeds operation consumes smaller amounts of hydrogen as well as permits wider range for injection timing. The inverse is true for higher speeds where very limited range is available for injection timing. For 2000 rpm, the fuel consumption rates are acceptable throughout the studied range with
injection timing of 60° BTDC being the optimum. At injection timing of 100° BTDC, minimum hydrogen consumed at 6000 rpm. Figure 6 illustrates the variation of power with injection timing with respect to changes the engine speed. It can be seen that the power gained increases with increases of engine speed except 6000 rpm case. However, this happens due to the interaction between injection duration and ignition timing. So, it does not represent the normal situation. This occurs at injection timing in the vicinity of TDC. The maximum power of 123 kW was gained at injection timing of 100° BTDC for 5000 rpm, while the optimum injection timing that gives at 2000 rpm was 40° BTDC and maximum power of 59 kW. The power shows a maximum at engine speed 5000 rpm. It is also observed that the power gained decreases at 6000 rpm due to the increase in the friction losses.

The variation of engine speed on the power gained is shown in Fig. 7 for stoichiometric operation and injection timing of 100° BTDC. From the acquired results, the power increases slightly with advances of injection timing towards TDC for all engine speed range. It is also seen that the slightly increase of power until about 30° BTDC for 2000 rpm then it drops down. The rate of change in power is higher for higher speed and drop occurs early in higher speeds. For very high speeds, the drop happens earlier due to the early interaction between the injection duration and ignition timing and friction losses. Figure 8 shows the trends of torque with injection timing with the interaction of engine speed effect. Higher torques is produced at lower speed with extra advantages of more acceptable operation range of injection timing. The severe drop with high speed introduces a challenge for injection timing optimization. Based on torque measure, the optimum injection timing throughout the studied speeds, ranged from 40° BTDC at 2000 rpm until 100° BTDC at 6000 rpm. This extended range imposes more control difficulties. However, compromise solutions can be applied.
4. CONCLUSIONS

A computational model was developed for four cylinders direct injection hydrogen fueled internal combustion engine. The main results are summarized as follows:

1. The engine performance is strongly depends on the engine speed. The engine speed 2500 rpm gives the maximum indicated efficiency.
2. Optimum injection timing depends also strongly on engine speed. Lower speeds advances optimum injection timing toward TDC timing.
3. As a compromise, injection timing of 60° BTDC can be considered as optimum for the present engine. However, this is for constant injection timing. The recommended operation is with different injection timing bases on engine speed.
4. Interaction between injection duration and spark timing is strongly undesired and can result in unstable operation. This was apparent by the unacceptable performance parameters during interaction period. Avoidance of this interaction should take priority in specifying injection timing.

5. ACKNOWLEDGMENTS

The authors would like to express their deep gratitude to Universiti Malaysia Pahang (UMP) for provided the laboratory facilities and financial support.

6. REFERENCES