Investigation of Intake Boost Pressure Effects on Performance and Exhaust Exergy of a Natural Gas Fueled HCCI Combustion Engine

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***Abstract* – To increase thermal efficiency of Internal Combustion (IC) engines and follow emissions regulations, new combustion strategies should be used. Homogeneous Charged Compression Ignition (HCCI) engine is one of the advanced combustion strategies which can reduce NOx significantly while it has a high thermal efficiency like diesel engines. In this study, a single cylinder HCCI engine was considered and the effects of change of boost pressure on engine performance were reported. Also, the effects of boost pressure on exhaust exergy components were studied. It is found that with increasing the boost pressure the specific fuel consumption per cycle reduced significantly.**

***Keywords* – HCCI Engine, Intake Boost Pressure, Parametric Study, Exhaust Exergy.**

I. Introduction

Internal Combustion (IC) engines are studied for decades. Increasing the thermal efficiency of IC engines as well as reduction of pollutants are main goals which scientists and engineers deal with. Two typical types of IC engines are Spark Ignition (SI) and Compression Ignition (CI) engines. In SI engines a homogenous mixture of fuel and air is compressed till the mixtures is ignited by introduction of spark in the mixture. On the other and in CI engine, the air is compressed inside the mixture and a fuel with high reactivity is injected to air. Due to high temperature and pressure of air, auto ignition of fuel occurs inside the engine and combustion happens. Therefore, the CI engines have higher compression ratio which resulted to higher thermal efficiency of this type of engines in comparison with SI engines.

The advantages of CI engines (also called diesel engines) over SI engines have led CI engines to become one of the main sources of power generation, especially in heavy duty engine applications. One of the main advantages is higher compression ratio of diesel engines in comparison with SI engines which leads to higher thermal efficiency as well as other advantages such as lack of throttle valve in the intake manifold of diesel engines, which results in maximum amount of air inside the cylinder in naturally aspirated engines and higher output work. Generally, the load in CI engines is directly controlled by amount of fuel (normally diesel) which is injected into the cylinder, starting from a minimum fueling rate corresponding to engine idling and increases to a maximum fueling rate corresponding to full load engine operation.

CI engines suffer from high amount of particulate matter (PM) and oxides of nitrogen (NOx). Moreover, unburned hydrocarbon (UHC) and carbon monoxide (CO) emissions are also emitted from the diesel engines, but there are not as significant as PM and NOx. It can be mentioned that diesel engines are an important source of PM, which includes small particles with about 0.1 micron diameter and consists of soot (which is mainly carbon) with some additional absorbed hydrocarbon materials. In diesel engines between 0.2 to 0.5 percent of fuel is emitted as PM. NOx is basically Nitric Oxide (NO) and small amount of Nitrogen Dioxide (NO2).

Taghavifar et. al. [1], used a coupled CFD and artificial neural network to investigate the wall heat flux in a direct injected diesel engine fueled with n-heptane. A Ford 1.81 DI diesel engine considered and the input parameters of mass flux, crank angle, liquid mass evaporated, equivalence ratio, turbulence kinetic energy, and pressure were included in the system. The RNG k–e turbulence model was considered to simulate the turbulent combustion in the chamber for unsteady equations of mixtures on closed system were carried out from IVC (52 CAD BTDC) to EVO (110 CAD ATDC). For the combustion process simulation in the combustion chamber the Extend Coherent Flame Model, 3-Zone (ECFM-3Z Model) developed based on the turbulent mixing was selected. In other words, the model was according to a flame surface density transport equation and a mixing model, which describes inhomogeneous turbulent premixed and diffusion combustion phases. They concluded that more wall heat flux was transferred with fuel injection around TDC and along with combustion initiation for 2000 rpm and the higher pressure can be achieved at the same engine speed.

To utilize the advantage of CI engines (i.e., high compression ratio) and reduction of emissions from them, new combustion strategies are introduced in the recent years such as dual fuel combustion engines [2-4], homogenous charge compression ignition (HCCI) engines [5-8], gasoline compression ignition engines [9-10] and other methods [11-12]. In these new combustion strategies, the mixture is ignited in temperatures lower than the NOx production temperature. Thus, the amount of NOx fractions in the exhaust is tremendously lower than typical diesel engines. However, due to high compression ratio in these type of engines, thermal efficiency of them are as high as diesel engines while they do not produce soot which is a challenge in CI engines.

Waste heat recovery system is another method in increasing thermal efficiency of IC engines. In this method, the hot energy in the exhaust will be used to produce power which leads to increase the total work output of an engines. The high energy in the exhaust is due to sudden expansion of high temperature and high pressure combustion gases when exhaust valve opens. Waste energy recovery applications in IC engines have been studied for decades [13-18]. For example, a Rankine bottoming cycle considered by Oomori and Ogino [15] to produce power from waste heat from the engine of a passenger car. In another research effort, Mahabadipour et. al. [18] introduced a method with combining simulation and experimental data to calculate the mechanical and thermal components of the exhaust gas energy. By adopting their method in our study, the mechanical and thermal components of combustion exhaust gas are calculated.

In order to design a suitable combustion system for HCCI engines, it is necessary to understand the effect of changing each operating parameter on combustion behavior. For this end, in this study, a parametric study is performed to determine the effect of intake boost pressure as an important parameter on specific fuel consumption, combustion phasing, heat transfer rate, maximum pressure and temperature inside the cylinder and exhaust characteristics such as NOx emission, and thermal and mechanical components of exhaust exergy at exhaust valve opening (EVO).

II. Methodology

To study the effect of intake boost pressure on combustion phenomena as well as exhaust characteristics, a natural gas fueled HCCI engine is considered. The compression ratio of engine is 17.5 with speed of 800 rpm. Also, the intake manifold temperature was considered as 447 K. In the following table, you can find engine specifications and operating conditions of the engine:

Table1. Engine specifications and operating conditions

Parameter	Specification
Engine type	Single cylinder
Throttle	Fully open
Compression Ratio	17.5
Main fuel	Natural gas
Bore	12 cm
Stroke	14 cm
Connecting rod	20 cm
Intake Temperature	447 K
Boost Pressure	1-1.8 Bar
IVO/IVC	4 /142 CAD
EVO/EVC	502/716 CAD

The closed part of the engine cycle (from IVC to EVO) considered to model HCCI combustion engine. Heat transfer, mass transfer and detailed chemical kinetics were included in the model. For modeling the heat transfer from the cylinder wall Woschni's correlation [19] was adopted. For describing the natural gas oxidation chemistry, the GRI-mech 3.0 [20] chemical kinetic mechanism was used. At exhaust valve opening (EVO), the combustion products encompass a large portion of energy which can be used with a waste energy recovery system. Exergy shows the maximum obtainable energy from a system when it interacts reversibly with environment [21]. Therefore, to study the effect of boost pressure on WER systems, exergy of exhaust gases at EVO should be considered. On the other hand, it should be mentioned that since the amount of unburned fuel is negligible in the combustion products of HCCI engines, therefore, the chemical exergy can be neglected from the exhaust exergy. So, physical exergy is the only component of exhaust exergy flow which can be calculated as follows:

$$\epsilon = (h - h_0) - T_0\,(s - s_0) \tag{1}$$

By considering the ideal gas behavior for combustion gases:

$$(h - h_0) = C_P\,(T - T_0) \tag{2}$$

$$(s - s_0) = C_P\, ln\left(\frac{T}{T_0}\right) - R\, ln\left(\frac{p}{p_0}\right) \tag{3}$$

Therefore, thermal and mechanical components of exhaust exergy will be:

$$\epsilon\_Th = C_P\,(T - T_0) - T_0\, C_P\, ln\left(\frac{T}{T_0}\right) \tag{4}$$

$$\epsilon\_Me = R\, T_0\, ln\left(\frac{p}{p_0}\right) \tag{5}$$

It should be noted that the exergy components will be calculated at EVO. It means that they represent the amount of mechanical and thermal exergy in the beginning of exhaust process not during it.

III. Results and Discussion

In the present work, the intake boost pressure changed from 1 bar (i.e., a naturally aspirated engine) to 1.8 bar (i.e., a turbocharged engine) to investigate the variations of combustion phasing, specific fuel consumption, maximum pressure and temperature inside the cylinder, and heat transfer rate. On the other hand, the exhaust flow characteristics such as emissions and its potential for waste energy recovery were study by reporting the NOx emission, and thermal and mechanical components of exhaust exergy at exhaust valve opening (EVO).

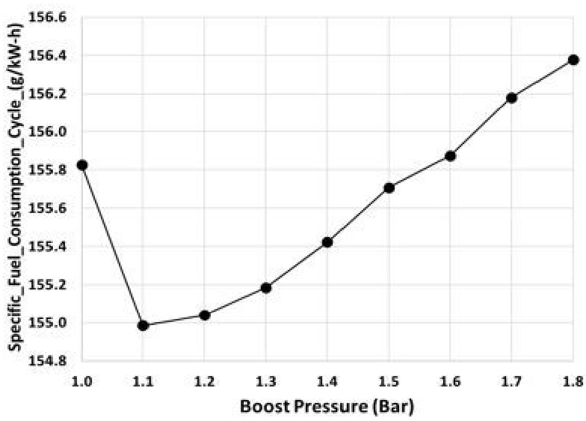

Fig. 1. Specific fuel consumption per cycle at different boost pressures

It is shown in Fig. 1 that with increasing the boost pressure from 1 bar to 1.1 bar, the specific fuel consumption per cycle reduced significantly. However, when boost pressure increased further, specific fuel consumption per cycle started to increase which reached to its maximum for the highest boost pressure. This behavior of specific fuel consumption per cycle will be explained by considering the 50 % combustion phasing.

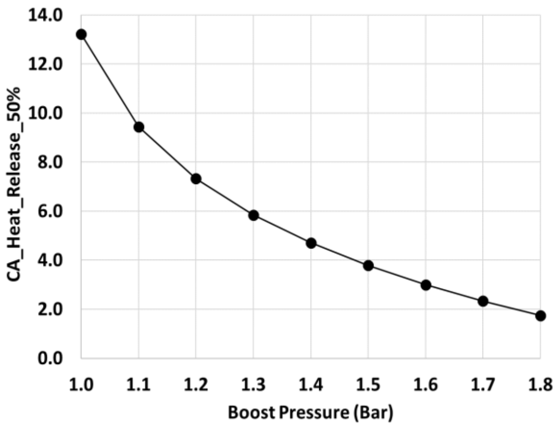

Fig. 2. 50% combustion phasing at different boost pressures

Fig. 2 shows that with increasing the boost pressure from 1 bar to 1.8 bar, 50% combustion phasing retarded from 13 CAD after top dead center to 2 CAD after top dead center. Since the best point for combustion phasing of HCCI engines occurs around 10 CAD, therefore, boost pressure of 1.1 bar had the minimum specific fuel consumption per cycle. In the following figure, the effect of boost pressure on heat transfer rate will be discussed. As it is mentioned before, to calculate the rate of heat transfer from cylinder wall to environments, Woschni's correlation was used.

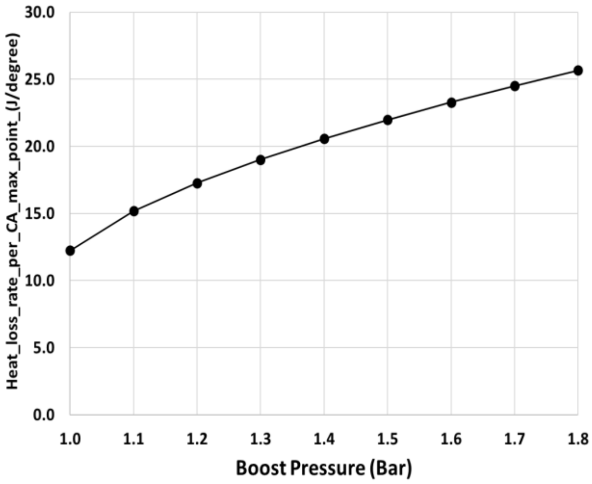

Fig. 3. Rate of heat transfer at different boost pressures

It is evident that with increasing the boost pressure the heat transfer rate from cylinder walls increased uniformly and reached its maximum for maximum boost pressure of 1.8 bar. It can be explained by considering the maximum temperature inside the cylinder.

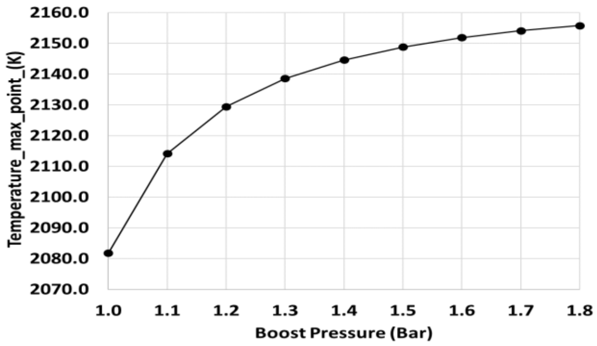

Fig. 4. Maximum temperature inside cylinder at different boost pressures

It is shown in Fig. 4 that with increasing the boost pressure, the maximum temperature inside the cylinder increased uniformly from 2080 K at 1 bar intake boost pressure to more than 2150 K at 1.8 bar intake boost pressure which led to increase of heat transfer rate as shown in Fig. 3.

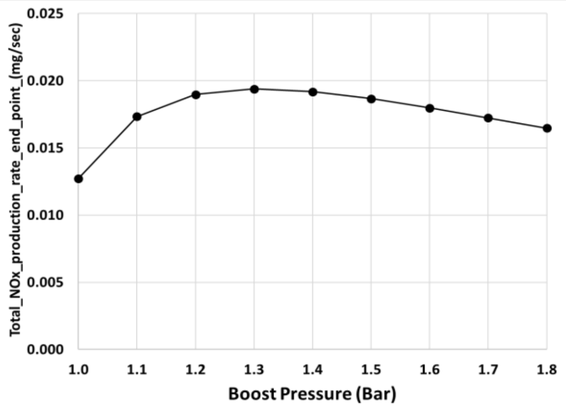

Fig. 5. Rate of total NOx production at different boost pressures

Fig. 5 shows that with increasing the boost pressure, the rate of total NOx production increased and reached to a maximum around 1.3 bar and reduced for higher boost pressures. It is noted that even though the specific fuel consumption was high for the lowest boost pressure of 1 bar, but the rate of total NOx production is minimum which make this operating point appealing based on stringent emissions regulation.

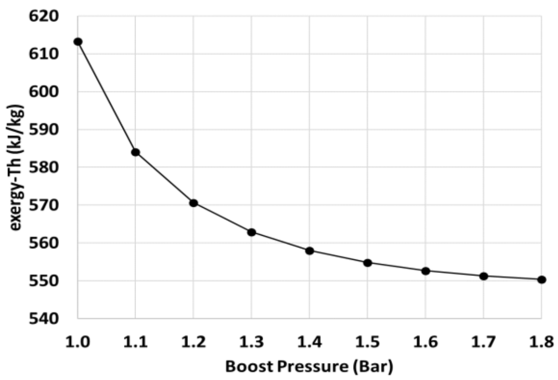

Fig. 6. Rate of thermal exergy at different boost pressures

Fig. 6 shows that with increasing the boost pressure, the specific thermal exergy at EVO reduced monotonically. It can be interpreted by considering the exhaust temperature at higher boost pressure which reduced for higher boost pressure. Therefore, if a WER system is suitable to use thermal exergy, it is better that engine runs at lower boost pressures.

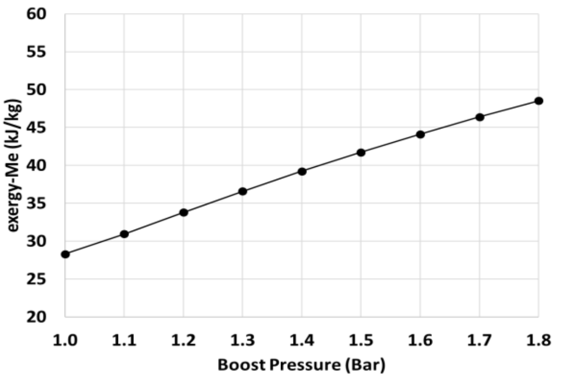

Fig. 7. Rate of mechanical exergy at different boost pressures

It is shown in Fig. 7 that with increasing the boost pressure, the specific mechanical exergy at EVO increased monotonically. It is because of higher exhaust pressure when engine runs at higher boost pressure. Therefore, if a WER system is suitable to use mechanical exergy, it is better that engine runs at higher boost pressures.

IV. Conclusions

In the present work, a single cylinder HCCI engine was considered and effect of change of boost pressure was studied. The intake boost pressure changed from 1 bar to 1.8 bar in steps of 0.1 bar and specific fuel consumption, combustion phasing, heat transfer rate, maximum temperature inside the cylinder and exhaust characteristics such as NOx emission, and thermal and mechanical components of exhaust exergy at exhaust valve opening (EVO) were reported.

It is found that with increasing the boost pressure the specific fuel consumption per cycle reduced significantly, combustion phasing retarded from 13 CAD after TDC to 2 CAD after TDC, the heat transfer rate from cylinder walls increased uniformly, the maximum temperature inside the cylinder increased uniformly, total NOx production increased and reached to a maximum around 1.3 bar and reduced for higher boost pressures, the specific thermal exergy at EVO reduced monotonically and the specific mechanical exergy at EVO increased monotonically.

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