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Performance Calibration for a Preheated Hydrous E85 Fueled Gasoline Engine

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Abstract

This work investigates influences of preheating anhydrous E85 and hydrous E85 fuels on engine performance of a 4-cylinder port fuel injection and ignition timing calibrated Toyota engine. The engine was loaded by dynamometer at low and medium load conditions over the speed of 2,500 rpm. The intake fuel temperatures were varied between 30°C and 70°C. The tested engine was calibrated to maximum brake torque timing and theoretical fuel-to-air equivalence ratio. For both loads when the intake fuel temperature was climbed up, the engine tended to deliver an increased brake torque and power for both fuels while, at elevated intake fuel temperatures, a pronounce increment in brake torque and power is observed. For all fuel temperatures, the brake torque and power from hydrous E85 combustion were higher as the engine can ignite earlier. The engine consumes more hydrous E85 at the low fuel temperatures while the adverse outcomes were attained at the higher temperature of 70°C. It is suggested that possibility to calibrate for combustion generated emissions can be achieved by adjusting ignition timing and injection duration, or adding fuel preheating element.

Keywords: anhydrous E85; engine performance; ethanol; fuel consumption; hydrous E85

1. Introduction

Fuel crisis within the last few decades encourages the use of alternative fuels available in Thailand such ethanol. This is due to fuel availability for continuously increasing consumption and price [1]. In addition, an incomplete combustion in internal combustion engines causes harmful substances to humans [2]. As a result, a quest for renewable energy to minimize such problems has been sorted [3].

Anhydrous ethanol (99.5% purity) is blended with gasoline in different amounts to fuel current gasoline engines, for instance E85 (volumetric blend of 15% gasoline and 85% anhydrous ethanol). However, the anhydrous ethanol production requires water removal from hydrous ethanol (95% ethanol and 5% water) at a cost [4]. In addition, water in hydrous ethanol can be affected by ambient heat prior to entering fuel injection system, and hence the combustion chamber of an engine.

Ethanol is a fuel derived from agriculture and industry, considered to be a renewable energy and a clean fuel. In the distillation process to separate ethanol, a 95 percent purity ethanol with 5 percent water has to be dehydrated to obtain 99.5 percent purity ethanol. The subsequence will be mixed with gasoline to make anhydrous E85 as previously described [5].

By the use of pure ethanol, there were several experiments on both original and calibrated gasoline engines. Costa and Sodré (2010) [6] comparative studied between ethanol mixture (6.8% water content) with 78% gasoline mixed with 22% ethanol fueling in four-cylinder four-stroke gasoline engine. It was found that the brake torque from the combustion of hydrous ethanol was greater. The hydrous ethanol mixture was used with engine at high revolution, resulting in greater thermal efficiency and brake specific fuel consumption than those of gasoline-ethanol blends.

Gupta *et al*. (2011) [7] studied the effects of hydrous ethanol (10% and 20% water contents) fueling to a single cylinder 4-stroke 125 cc gasoline engine. The results have been revealed that the hydrous ethanol enhanced thermal efficiency and brake specific fuel consumption respectively with the increasing amount of water in the fuel mixtures.

Recently, in other aspects, Ketprakong and Chuepeng (2015) [8] explored the effects of preheating hydrous E85 fuel on combustion characteristics and exhaust gas emissions. Its experiment was accomplished using calibrated Toyota gasoline engine running at 2,500 rpm with light and medium load (25% and 50% pedal positions). The fuel temperatures prior to intake to the engine were set at 30, 50 and 70°C. The preheating hydrous E85 caused a higher peak cylinder pressure, faster combustion rate and longer combustion duration. This was as the ignition timings were advanced to the earlier crank positions. Meanwhile, the combustion of the preheating hydrous E85 emitted lower nitrogen oxides but higher carbon dioxide compared to other fuels tested.

Subsequently, there are some other aspects that have not been yet studied concerning preheating hydrous E85. The main aim of this work is to experimental study the power output and specific consumption of a gasoline engine, calibrated for performance as a result of preheating hydrous E85. By this manner, the engine's ignition timing and fuel injection systems have to be adjusted for stoichiometric combustion. At the end of this paper, the conclusion can suggest how to cope with this fueling condition changed.

2. Experimental apparatus

2.1 Engine and dynamometer

The engine used in this experimental study is a 4A-FE Toyota gasoline engine with concise specifications as shown in Table 1.

Table. 1 Engine specification

Engine type	4 cylinder - 4 stroke
Bore	81.03 mm
Stroke	76.92 mm
Displaced volume	1,587 cc
Maximum torque	147 N·m at 5,200 rpm
Maximum power	86 kW at 6,000 rpm
Compression ratio	9.5:1
Cooling system	Water cooled

The engine is loaded by a water-brake engine dynamometer Schenck model D-400-1e with the maximum capacity of 400 kW power and 1,300 N·m torque. The standard load cell for the engine dynamometer is from Wagezelle type U2A standing with the maximum capacity of 1 Mg load. Its sensitivity is 2 mV/V and its non-linearity is less than ±0.05%. The reading for revolution speed from the engine dynamometer in speed control mode is bounded for ±10 rpm accuracy. The engine test rig is depicted in Fig. 1.

Fig. 1 Engine test rig

2.2 Engine indicating system

Cylinder pressure data corresponding to the position of the crankshaft is used to study the characteristics of combustion; detail analysis was presented elsewhere in [8]. The combustion pressures in the fourth cylinder are measured by a pressure transducer from Kistler Model 6052C with -19.90 pC/bar sensitivity at 200°C and ± 0.4% FSO linearity. The subsequent signal is sent to a charge amplifier from Dewetron DEWE-30-4 model to be used to expand and refine the efficacy signal from the pressure at the same time. The measured position of the engine crankshaft rotation is determined by an incremental shaft encoder from TR Electronic GmbH. Operating rates of the encoder is set to measure the position of the shaft 360 pulses per revolution (360 ppr) of the crankshaft. All signals are sent to a data acquisition system model Dewetron DEWE-ORION-0816-100x with the sampling rate of 1 MS/s and DEWEsoft V.6.6.8 software for data analysis. Cylinder pressure traces of 100 consecutive cycles are averaged as statistical values and are then used to represent the values of the combustion characteristics in each test condition.

The engine indicating system is used for monitoring knock event when calibrating for the ignition timing. However, the ignition knock was not observed throughout the test as it may be due to an inherently higher octane number for ethanol, the main constituent in E85 fuel.

2.3 Air and fuel flow metering

The rate of air flowing into the engine is measured by a Testo Model 435 with the accuracy within ± 0.3% of reading. The rate of fuel flow in mass basis is measured by CST balance model CDR-6 and the fuel mass can be measured with accuracy within ± 0.05 g. The temperature of the fuel flow is controlled within the range ± 1°C of the set temperatures.

2.4 Ignition timing and fuel controllers, and ignition timing measurement

A standard electronic control unit (ECU) is additionally equipped with ATD E85 ECU and ATD TAP controllers for adjusting fuel quantity and ignition timing, respectively. In this circumstance, some OEM signals have to be tapped as input signals to the additional controllers, e.g. speed, throttle position sensor, manifold absolute pressure, and so on. After calibration, the actual ignition timing events are determined using an Inductive Advance Timing Analyzer from Trisco model DA-3100. Its capability is to sense the ignition timing of advancement to 60 crank angle degree before top dead center in between 200 and 10,000 rpm interval using engine 12 VDC.

2.5 Fuel

There are two types of fuel used in the test: anhydrous E85 with 0.43% water in ethanol (called AE85) and hydrous E85 with 5% water in ethanol (called HE85). General properties of both fuels are listed in Table 2.

Table. 2 Fuel properties

Selected properties	AE85	HE85
Empirical formula	C_2H_6O	$C_{1.91}H_{5.82}O$
Theoretical air-to-fuel ratio	9.765	9.33
Water content in ethanol	0.43%	5%

3. Experimental methodology

3.1 Test condition

The engine tests are conducted under steady-state conditions using the anhydrous E85 and hydrous E85 (5% water content in ethanol), hereinafter referred to AE85 and HE85, respectively.

Table. 3 Engine test conditions at 2,500 rpm

Condition	Fuel	Fuel temperature (°C)	Load (% pedal)
1	AE85	30	25
2		30	50
3		50	25
4		50	50
5		70	25
6		70	50
7	HE85	30	25
8		30	50
9		50	25
10		50	50
11		70	25
12		70	50

The engine is loaded by the dynamometer in constant speed mode at 2,500 rpm, representative as usual speed often being used in real driving condition. Meanwhile, the engine loads are set as low to moderate at the throttle position of 25% and 50% as indicated by linear variable differential transformer voltage. The temperature of the intake fuel to the engine is controlled to be constant at 30, 50 and 70°C. A summary of all test conditions is illustrated in Table 3. In each test condition, the values of torque, speed and fuel consumption are recorded as well as ambient conditions. The latter is intentionally used for calculating correction factor, discussed in the next section.

As aforementioned, the engine's fuel injection system can be adjusted for the ignition timing and the fuel quantity. In this study, the engine is calibrated for the ignition timing to approach the engine's maximum brake torque (MBT). In addition, the amount of fuel is adjusted to approach a theoretical fuel-to-air equivalence ratio measuring using lambda sensor, dependent on fuel type used.

3.2 Studied parameters

In each test condition, the engine brake torque is measured as previously mentioned in Section 2.1 corresponding to the engine revolution speed. Subsequently, the engine brake power (P_b) can be calculated using Eq. (1) [9].

$$P_b = 2\pi N T_b \tag{1}$$

where N is engine revolution speed and T_b is engine brake torque.

Specific fuel consumption is determined by calculation from fuel flow rate in mass basis divided by power output using Eq. (2) [9].

$$\mathrm{bsfc} = \frac{\dot{m}_f}{P_b} \tag{2}$$

where bsfc is the brake specific fuel consumption, $\dot{m}_f$ is the fuel consumption, and P_b is the brake power.

The engine fuel conversion efficiency (η_f) is given by [10]:

$$\eta_f = \frac{1}{\mathrm{bsfc} \times Q_{LHV}} \tag{3}$$

where $\dot{m}_f$ is the fuel mass flow rate per cycle, P_b is the brake power and Q_{LHV} is the lower heating value of the fuel used. However, in this circumstance of the blended fuels, their Q_{LHV} values have to be weighed by mole fraction of each fuel. In this case, the Q_{LHV} values of the gasoline and neat ethanol are of 44.0 MJ/kg and 26.9 MJ/kg, respectively.

Ambient air temperature, pressure, and humidity can affect the power output and air mass flow rate inducted into the engine. The brake power obtained from the dynamometer reading has to be adjusted by a correction factor (C_F) given in [10] to convert it to standard atmospheric condition for comparison between engines.

$$C_F = \left(\frac{p_s}{p_m - p_{v,m}}\right) \times \left(\frac{T_m}{T_s}\right)^{0.5} \tag{4}$$

where p_s and T_s are standard air absolute pressure and temperature, respectively; p_m is measured ambient air absolute pressure; $p_{v,m}$ is ambient water vapor partial pressure; and T_m is measured ambient temperature. Therefore, all the torque and power values presented here in the result section are already corrected.

4. Results and discussion

4.1 Ignition timing

Initially, the engine running on AE85 fuel at 2,500 rpm ignites at the baseline timing of 20 crank angle degrees (CAD) before top dead center. However, when the engine has been calibrated for MBT while keeping the fuel-to-air equivalence ratio to be of unity with fuel type and temperature variation, the ignition timing has to be adjusted to meet these requirements.

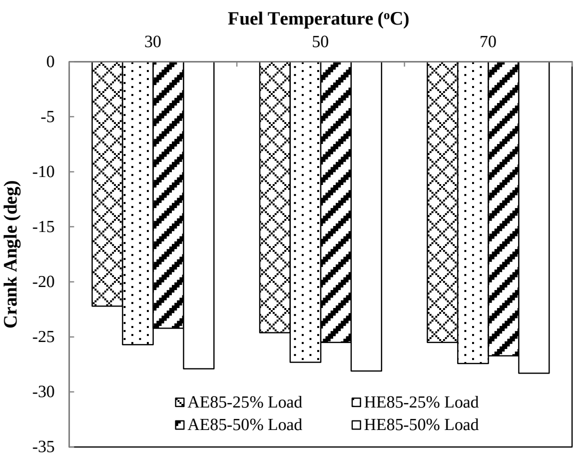

Fig. 2 Ignition timing

Fig. 2 shows the ignition timing after the calibration for MBT that varied with fuel type (AE85 and HE85), fuel temperatures, and engine load at the constant speed of 2,500 rpm.

The use of HE85 can advance the ignition timing to earlier CADs compared to those of AE85 for both loads at all intake fuel temperatures. At higher loads, the engine exhibits to advance its ignition timing itself. When the intake fuel temperatures were elevated, the engine can even be advanced to earlier CAD to get for MBT. However, the difference in advancement of the ignition timing at lower fuel temperature is shown to be more pronounce.

At 25% load, the maximum advancement of the ignition timing occurred at 30°C was by 3.5 CAD when using HE85 instead of AE85, 5.7 CAD more advancement compared to those running AE85 prior to calibration. Meanwhile at the same engine load, the minimum advancement of the ignition timing occurred at 70°C was by 1.9 CAD when using HE85 instead of AE85, 7.4 CAD more advancement compared to those running AE85 prior to calibration.

At 50% load, the maximum and minimum advancements of the ignition timing occurred at 30°C and 70°C were by 3.7 CAD and 1.6 CAD, respectively when using HE85 instead of AE85, 7.9 CAD and 8.3 CAD more advancement compared to those running AE85 prior to calibration.

Water content in ethanol when HE85 is in use can decelerate the combustion rate that involves to a knock resistance [10]. This brings about a capability of HE85 to be ignited in to earlier advance crank angles.

At higher loads, the engine doses more fuel in to the combustion chamber [10]. This causes a longer time for fuel to combust within the chamber, longer combustion duration [8]. The engine requires advanced ignition timing to get the combustion process accomplished with in a limited combustion time. In the same manner, when the combustion mixture undergoes beneath environment at high fuel and combustion temperatures, the engine exhibits likewise. Therefore, both higher loads and high fuel temperatures can advance the ignition timing as obviously seen in Fig. 2.

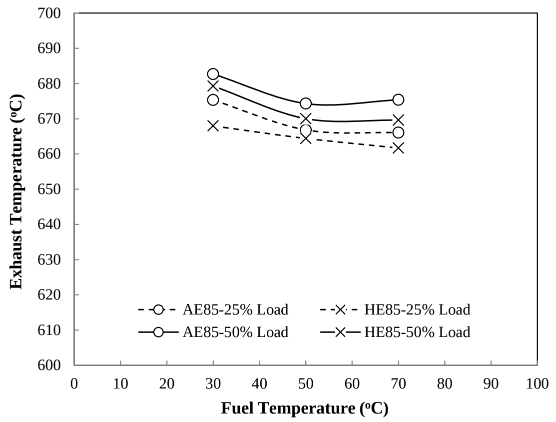

Fig. 3 Exhaust temperature

Fig. 3 shows the exhaust gas temperatures averaged from four cylinders in each test condition after the calibration for MBT that varied with fuel type (AE85 and HE85), fuel temperatures, and engine load at the constant speed of 2,500 rpm.

The average exhaust temperatures were ranging from 662°C to 683°C for all conditions. The elevated fuel temperatures lower the exhaust temperatures for all the tests. The preheated fuel intakes to the combustion process that can fire combustible mixture in some greater extent regardless of fuel type used, resulting in lowered exhaust temperatures. Meanwhile, the HE85 combustion brought cooler exhaust gases than those of the AE85. The advanced ignition timing for the HE85 fuel is accounted for the combustion to be ended prior to that of the AE85. Obviously, the lower engine load generates cooler exhaust gases [10].

4.2 Engine brake torque and power

Primarily, the engine running on AE85 fuel at 2,500 rpm delivered brake torque as a baseline by 16.1 N·m and 26.8 N·m, respectively for low and high loads. However, when the engine was calibrated for MBT while keeping the fuel-to-air equivalence ratio to be of unity with fuel type and temperature variation, the brake torque changed, and hence the brake power.

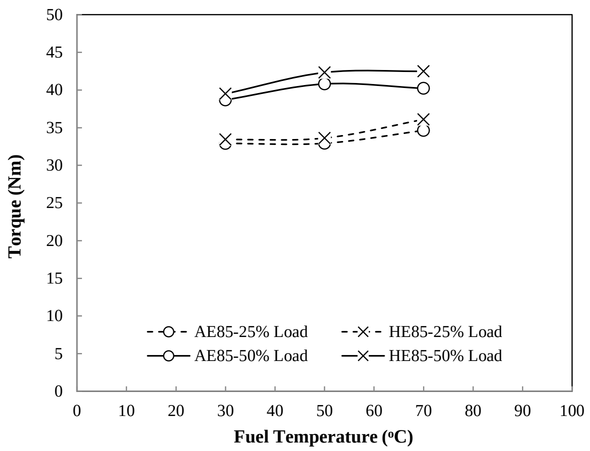

Fig. 4 Brake torque

Fig. 4 shows the brake torque after the calibration for MBT that varied with fuel type (AE85 and HE85), fuel temperatures, and engine load at the constant speed of 2,500 rpm.

The use of HE85 as a fuel for the engine with calibration enhances the brake torque compared to those of AE85 for both loads at all intake fuel temperatures. However, the higher brake torque for HE85 was more evident at the higher load. At the higher load, the brake torque from AE85 combustion was down when the intake fuel temperatures were elevated. Meanwhile, at the lower load, it was contrast of the trend compared to that at the higher load.

At 25% load, the maximum different brake torque occurred at 70°C was by 1.5 N·m when using HE85 instead of AE85, counting for 20.0 N·m greater torque compared to those running on AE85 prior to calibration. Meanwhile at the same engine load, the minimum different brake torque occurred at 30°C was by 0.5 N·m when using HE85 instead of AE85, counting for 17.3 N·m compared to those running on AE85 prior to calibration.

At 50% load, the maximum and minimum different brake torque occurred at 70°C and 30°C were by 2.3 N·m and 0.8 N·m, respectively when using HE85 instead of AE85, counting for 15.7 N·m and 12.7 N·m greater brake torque compared to those running AE85 prior to calibration.

Note that, for both engine loads, the brake torque between both HE85 and AE85 were linearly different at all intake fuel temperatures. These trends of the torques project the same pattern to the derivative brake power as shown in Fig. 5.

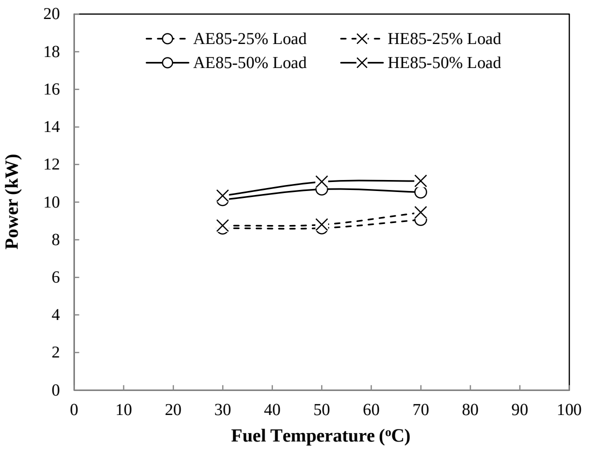

Fig. 5 Brake power

Initially, the engine running on AE85 fuel at 2,500 rpm delivered brake power as a baseline by 4.2 kW and 7.0 kW, respectively for low and high loads.

Fig. 5 shows the brake power after the calibration for MBT that varied with fuel type (AE85 and HE85), fuel temperatures, and engine load at the constant speed of 2,500 rpm.

As a result of the brake torque, the use of HE85 as a fuel for the engine with calibration enhances the brake power compared to those of AE85 for both loads at all intake fuel temperatures. However, the higher brake power for HE85 was more evident at the higher load. At the higher load, the brake powers were stable for HE85 but they were enhanced for AE85, when the intake fuel temperatures were elevated.

At 25% load, the maximum and minimum different brake powers occurred at 70°C and 30°C were by 0.3 kW and 0.1 kW, respectively when using HE85 instead of AE85, counting for 2.4 kW and 4.9 kW greater brake powers compared to those running AE85 prior to calibration.

At 50% load, the maximum and minimum different brake powers occurred at 70°C and 30°C were by 0.6 kW and 0.2 kW, respectively when using HE85 instead of AE85, counting for 4.1 kW and 3.3 kW greater brake powers compared to those running AE85 prior to calibration.

The greater brake torques and powers from HE85 combustion compared to that of AE85 were mainly due to capability to advance ignition timing [10] even though HE85 fuel contains a trace of water that lowers its heating value. Even when the intake fuel temperatures were elevated, the engine can be advanced to the earlier crank angle position to get MBT.

4.3 Specific fuel consumption and fuel conversion efficiency

The brake specific fuel consumption is shown in Fig. 6 after the calibration for MBT that varied with fuel type, fuel temperatures, and engine load at the constant speed of 2,500 rpm.

Apparently at 30°C, the case of HE85 consumed more fuel than the case of baseline AE85 due to its reduced heating value by water component in HE85. This has caused the slight decrease in engine fuel conversion efficiency (Fig. 7) compared to the case of baseline AE85 by -2.4% and -5.1% at higher and lower loads, respectively.

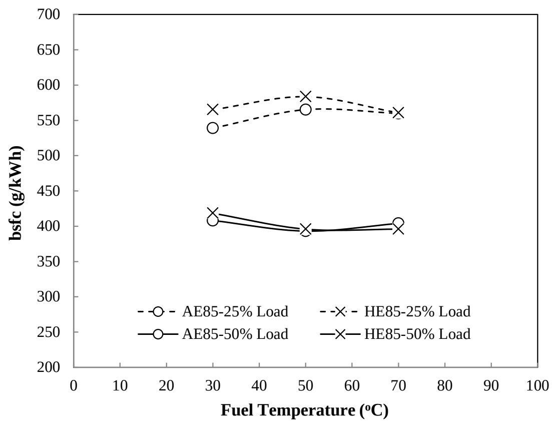

Fig. 6 Brake specific fuel consumption

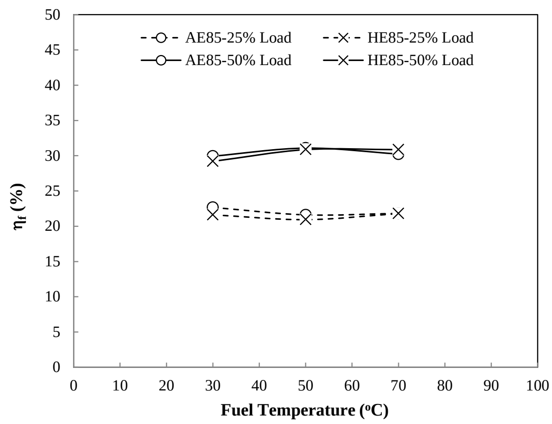

Fig. 7 Fuel conversion efficiency

In Fig. 6 when the fuel temperature was heightened up to 50°C, the fuel consumption behavior for two loads was deviate, consuming more fuel at higher load but consuming less fuel at lower load. However, when the fuel temperature was set to 70°C with engine MBT calibration, the fuel consumption for the HE85 case has shown to be equivalent to that of the AE85. This brought the fuel conversion efficiency for both fuel at 70°C to be comparable as seen in Fig. 7.

By calibrating the engine, it is confirmed by experiment that the engine can be operated by retrofitting with alternative fuel such HE85. To earn the same benefit such fuel conversion efficiency in the case of using HE85 that contains trace of water, it is recommended to preheat the fuel prior to intake to the engine. However, care should be taken in the case of fire issue that may occur during operation.

5. Conclusion and recommendation

This research work studies the effects of preheating AE85 and HE85 fuels on engine performance, fuel consumption, and efficiency of a 4-cylinder Toyota gasoline engine calibrated for MBT and stoichiometric combustion at 2,500 rpm with low and medium load condition. The fuel temperature is constantly controlled at 30, 50 and 70°C.

When feeding the HE85 fuel with elevated temperatures, the engine can ignite earlier. At normal fuel intake temperature of 30°C, the engine with HE85 fueling at the lower and medium loads can advance the ignition timing by 3.5 CAD and 3.7 CAD, respectively compared to the AE85 after calibration (5.7 CAD and 7.9 CAD compared to AE85 prior to calibration). At the fuel intake temperature of 70°C, the ignition advancement can be even extended. These causes the engine can deliver more torque and power for HE85 combustion compared to those of the AE85. With the set calibration, the engine consumed HE85 in equivalence to that of AE85 at 70°C; this gained the fuel conversion efficiency to be comparable between the two fuels.

This shows a positive impact of the application for the engine that has to be calibrated for ignition timing and fuel injection to gain for MBT while the preheating HE85 is required up on the extent under safety limit.

6. Acknowledgement

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7. References

[1] Sangeeta, Moka, S., Pande, M., Rani, M., Gakhar, R., Sharma, M., Rani, J., and Bhaskarwar, A.N. (2014). Alternative fuels: an overview of current trends and scope for future, *Renewable and Sustainable Energy Reviews*, vol.32, pp. 697-714.

[2] Leong, S.T., Muttamara, S., and Laortanakul, P. (2002). Influence of benzene emission from motorcycle on Bangkok air quality, *Atmospheric Environment*, vol.36, pp. 3495-3503.

[3] Zhuang, Y. and Hong, G. (2013). Primary investigation to leveraging effect of using ethanol fuel on reducing gasoline fuel consumption, *Fuel*, vol.105, pp. 425-431.

[4] Kaminski, W., Marszalek, J., and Ciolkowska, A. (2008). Renewable energy source—dehydrated ethanol, *Chemical Engineering Journal*, vol.135, pp. 95-102.

[5] Kumar, S., Singh, N., and Prasad, R. (2010). Anhydrous ethanol: a renewable source of energy, *Renewable and Sustainable Energy Reviews*, vol.14, pp. 1830-1844.

[6] Costa, R.C. and Sodré, J.R. (2010). Hydrous ethanol vs gasoline-ethanol blend: engine performance and emission, *Fuel*, vol.89, pp. 287-293.

[7] Gupta, P. et al. (2011). Impact of water contents blended with ethanol on SI engine performance and emissions, *Journal of Research and Applications in Mechanical Engineering*, vol.1, pp. 7-11.

[8] Ketprakong, O. and Chuepeng, S. (2015). Effects of Preheating Hydrous E85 Fuel on Combustion Characteristics and Exhaust Gas Emissions (in Thai), paper presented in *the 29th Conference of Mechanical Engineering Network of Thailand*, Nakornratchasrima, Thailand.

[9] Stone, R. (1999). Introduction to Internal Combustion Engines, Macmillan Press, China.

[10] Heywood, J.B. (1988) Internal Combustion Engine Fundamentals, McGraw-Hill, Singapore.