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Study of Agricultural Engine for Predicting Engine Performance by Thermodynamics Model

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Abstract

The KUBOTA RT140, the engine used widely in Thailand's agricultural field, was selected as the test engine for this investigation. To develop the engine performance, a number of time consuming and costly tests were necessary to optimize engine performance. Thus, thermodynamics model (AVL Boost) was applied to predict engine performance in order to reduce costs and time. For the test, the KUBOTA RT140 was modified by the installation of measurement equipment i.e. in-cylinder pressure transducer, fuel flow. Following the experiment, the engine was tested at 50% load while the ambient temperature and pressure, in-cylinder temperature and pressure, heat release rate and lift valve curve were recorded. Experimental data and the obtained results were used for calculation of the thermodynamics model. Thermodynamics calculation results were calibrated and engine performance i.e. in-cylinder pressure and brake specific fuel consumption were closely matched to the experimental data. The calibrated thermodynamics model of adjustment of advance and retard timing found the same trends as experimental data for a diesel engine. Moreover, the Diesel Dual Fuel (DDF) engine model studied by varying of the mixing ratio (Z) was investigated. The results show that the obtained thermodynamics model is verified to predict engine performance for both diesel and DDF engine.

***Keywords**:* Engine performance, Thermodynamics model, KUBOTA RT140

1. Introduction

Due to current demands such as shortage of non-renewable energy resources and emission reduction, the engine and automotive industries are striving to enhance existing engine and vehicle concepts [1]. In an agricultural country, many small diesel engines are used for durable and tough tasks such as the Kubota RT140 engine. This is the agricultural engine widely used due to its powerful response and fuel efficiency. According to the theoretical engine cycle, the diesel cycle (constant pressure) has greater thermal efficiency at the end of combustion than the Sabathe cycle during constant maximum pressure conditions. Therefore, when the diesel cycle is applied as a design concept for any diesel engine, it can benefit engine efficiency and reduce NOx emission, with no distinction between engine size and speed. This is a principle of emission control technology which is described in [2]. However, it has been noted that excessive time and high costs were necessary for engine development. As shown in previous studies [3] and [4], in order to develop the diesel engine, the AVL Boost program was studied using engine performance simulations. Comparisons were done between engine performance and engine experiments pertaining to the combination of various parameters to achieve the NOx emission target. In this study of the Kubota RT140, the engine performance simulation is calculated by AVL Boost program and confirmed through engine experiments. This study also changes the injection timings of matched model to examine and compare the results between experiment and simulation.

2. Procedure

2.1 Experimental apparatus

This experiment was performed on a single cylinder, 4 stroke diesel engine Kubota RT140. The engine specifications are described in Table 1. The power output and torque were controlled using an eddy current dynamometer NISHISHIBA NEDZ-113. The in-cylinder pressure was measured using the Kistler 6061B as the pressure transducer. The data obtained from the transducer was passed through a charge amplifier and sent to a data acquisition system for further processing [5]. The amount of fuel consumed was measured by weight and was converted into an electrical signal by the Load Cell. Moreover, the intake and exhaust valve lift curve were measured by strain gauges. The signals were then sent for further analysis to the data acquisition system. The schematic of the experimental apparatus can be seen in Fig. 1

Table. 1 Specifications of the diesel engine

ENGINE SPECIFICATIONS	
MODEL	KUBOTA RT140
Engine Type	1 cylinder
Combustion Chamber Type	Direct Injection
Bore x Stroke	97 x 96 mm
Displacement Volume	709 cc
Compression Ratio	18:01
Maximum power output	14 hp at 2400 rpm
Continuous rated power output	12 hp at 2400 rpm
Maximum Torque	5 kgf-m at 1600 rpm
Injection pump type	In-line pump
Governor Type	Mechanical
Injection Timing	18 deg BTDC

2.2 Methodology

In this experiment, the torque and engine speed are kept constant at 24.5 N-m and 2000 rpm, respectively. First, the engine is under full diesel operation at 50% load to obtain the engine performance , i.e. in-cylinder pressure, heat release rate, intake and exhaust valve lift curve and brake specific fuel consumption. Heat release rate and brake specific fuel consumption were calculated using the eqs. (1) and (2), repectively[3].

$$\frac{dQ_{net}}{d\theta} = \left(\frac{\gamma}{\gamma-1}\right) P \frac{dV}{d\theta} + \left(\frac{1}{\gamma-1}\right) V \frac{dP}{d\theta} \qquad (1)$$

$$\text{bsfc} = \frac{\dot{m}}{P} \qquad (2)$$

The lower heating value for diesel was taken to be 45.56 MJ/Kg. Moreover, the engine was under full diesel operation and gas level was varied by increasing the mixing ratio (Z). Z was increased until it reached to a point where further improvement was not possible under normal engine operation. The mixing ratio Z can be calculated by using Eq.(3).

$$Z\% = \frac{m^{\bullet}_{CNG}}{m^{\bullet}_{D} + m^{\bullet}_{CNG}} \times 100 \qquad (3)$$

Typical natural gas composition for Bangkok and the East region was used in this study. Energy component of natural gas with respect to total energy can be found using Eq.(4).

$$CNG\% = \frac{m^{\bullet}_{CNG} \times LHV_{CNG}}{m^{\bullet}_{D} \times LHV_{D} + m^{\bullet}_{CNG} \times LHV_{CNG}} \qquad (4)$$

Mass flow rates and energy components of diesel, the gaseous fuel, fueling mass and the total lower heating value for diesel and natural gas are described in the Table 2.

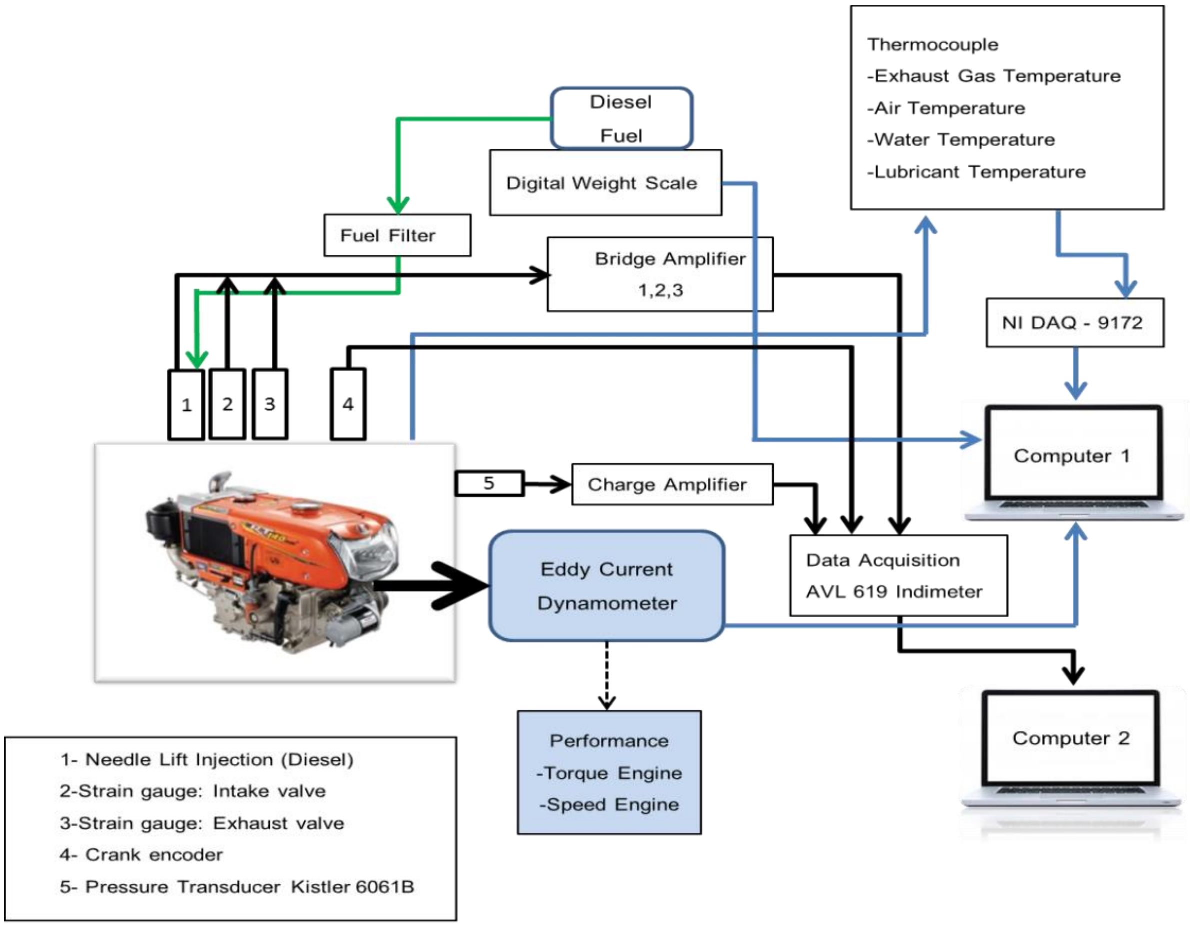

Fig. 1 Schematic diagram of the experimental apparatus

Table. 2 Details of the energy component for diesel and CNG.

Z	Mass flow rate		LHV		LHV (total)	Fueling mass
[%]	CNG	Diesel	CNG	Diesel		[g/cycle]
	[kg/hr]	[kg/hr]	[MJ/kg]	[MJ/kg]	[MJ/kg]	
0	0	1.47	0	45.6	45.6	0.0245
70	1.81	0.76	27.5	12.7	40.1	0.0428
82	1.84	0.38	32.4	7.3	39.7	0.0370

2.3 Simulation procedure

The thermodynamic engine cycle simulation tool (AVL Boost v.2011) was used to calculate the effects of engine configuration on engine performance. The calculation model is shown in Fig. 2. In this simulation model, intake and exhaust valve curves, intake and exhaust temperature, intake and exhaust pressure, heat release rate obtained from experiment and engine specifications are used as engine model input data.

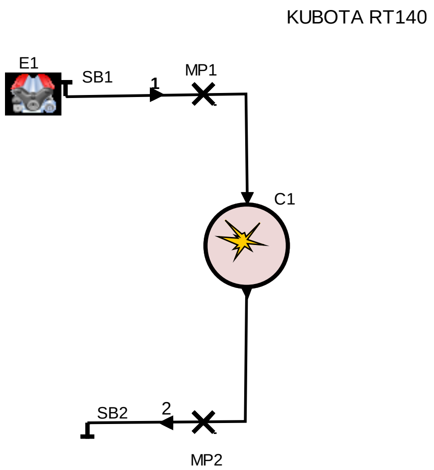

Fig. 2 Calculation model

When the model is calculated by 1D thermodynamics, the results of in-cylinder pressure, brake specific fuel consumption and exhaust temperature are compared and confirmed with experimental results.

3. Results and Discussion

3.1 Experimental results

Fig. 3 shows intake and exhaust valve curves measured by strain gauge through bridge amplifier and data acquisition. Moreover, in-cylinder pressure and heat release rate calculated from eq. (1) for base, retard and advance injection timing are illustrated in Figs.4-5. The results show the in-cylinder pressure increases(decreases) when the injection timing advances(retards) due to the combustion period shifted forward(backward). For 50% load in this study, the peak firing pressure is equal to or less than the in-cylinder pressure at the end of the combustion stroke. The timing at peak firing pressure is also retarded remarkably, and fuel consumption is significantly deteriorated (Table. 3). These valve curves, valve specifications and heat release rate were used as simulation input data. Moreover, the rate of heat release and in-cylinder pressure between diesel and DDF engine, are compared. Fig. 6 shows that the maximum cylinder pressure for diesel is higher than the dual fuel operation. This is due to lower heating value for of natural gas as compared to diesel. However, the varying of mixing ratio(Z) at 70% and 82% was investigated to study in-cylinder pressure and BSFC for DDF engine model (Figs.6-7).

Table. 3 Experimental results at base, retard and advance injection timing 13,18, 23 degBTDC)

Case	Injection timing [degCA]	Pmax [MPa]	BSFC [g/kWh]	Texh [degC]
Retard	-13	75.7	263.9	478.9
Base	-18	74.1	287.6	490.0
Advance	-23	70.4	316.0	501.0

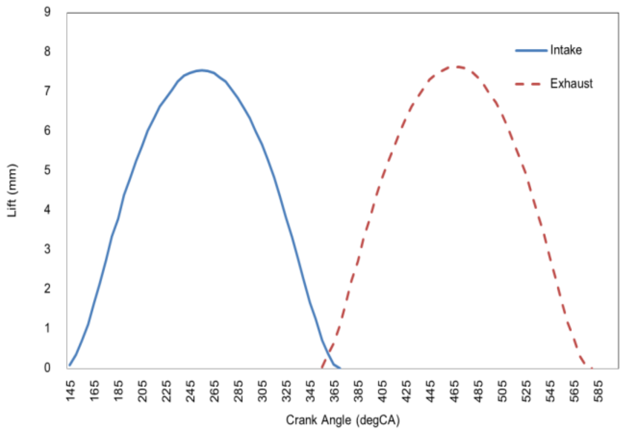

Fig. 3 Intake and exhaust valve curve for KUBOTA RT140

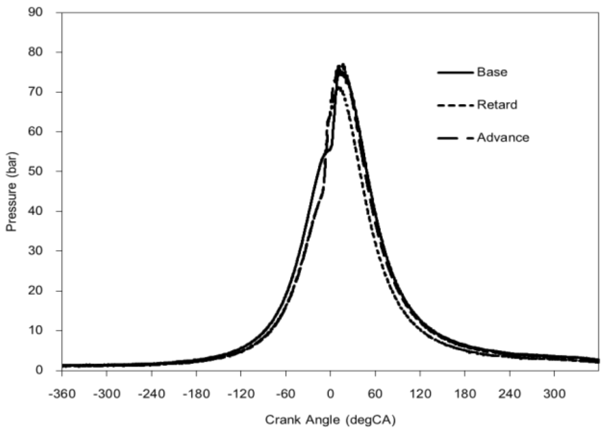

Fig. 4 In-cylinder pressure at injection timing 13,18 and 23 deg BTDC

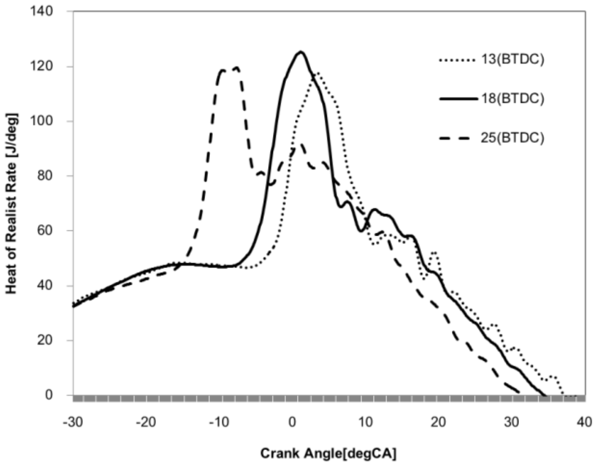

Fig. 5 Rate of heat release at injection timing 13,18 and 23 deg BTDC

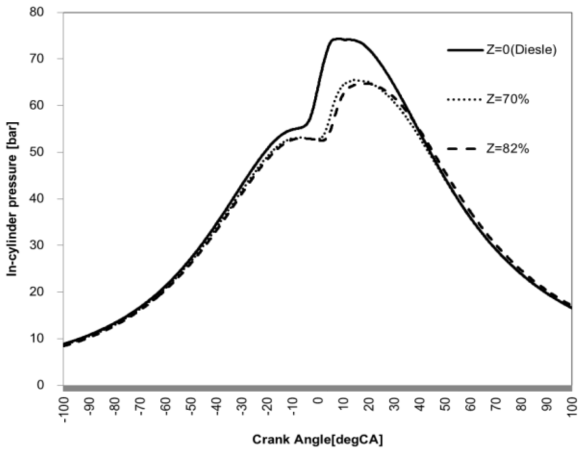

Fig. 6 Comparison of in-cylinder pressure between diesel and DDF at Z=0%, 70%and82%

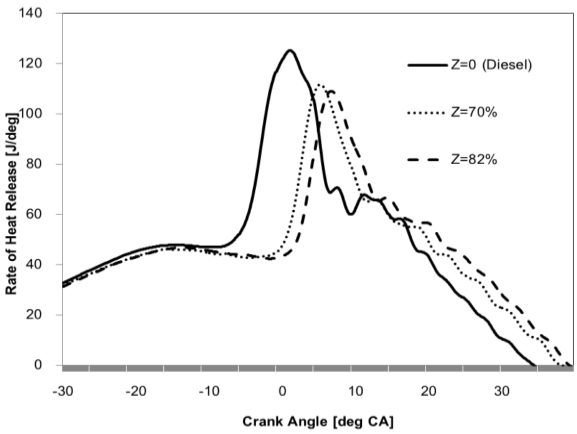

Fig. 7 Comparison of heat release rate between diesel and DDF at Z=0%, 70% and 82%

3.2 Comparison between experimental and simulation results

When the experimental data was input in the engine model (KUBOTA RT140), the results of engine performance compared between experiment and simulation were obtained. In the condition of injection timing 18 degCA BTDC, the simulation of brake specific fuel consumption differs from the experiment by less than 0.07%(Table. 4). Using the same model, when the injection timing was adjusted to 15 (advance) and 23 (retard) degCA, the same trends of brake specific fuel consumptions at 15, 18 and 23 degCA BTDC between simulation and experimental results are obtained (Table. 5).

Table. 4 Comparison between experimental and simulation results at injection timing 18 degCA BTDC

Case	Injection timing	Pmax	BSFC
	[degCA]	[MPa]	[g/kWh]
Experiment	-18.0	74.1	287.6
Simulation	-18.0	74.4	287.8

Table. 5 Comparison of fuel consumption between experimental and simulation results at injection timing 13, 18 and 23 degCA BTDC

Case	Injection timing	BSFC(exp)	BSFC(sim)
	[degCA]	[g/kWh]	[g/kWh]
Retard	-13	263.9	261.6
Base	-18	287.6	287.8
Advance	-23	316.0	319.8

Moreover, this matched model was applied to DDF engine to study the effects of mixing ratio(Z) variation. Table 6 shows the comparison of engine performance between experimental and simulation for DDF at Z=0%, 70% and 82%. It can be seen that the maximum cylinder pressure for pure diesel was higher than for dual fuel operation. When Z increases, not only the lower heating value changes but also inlet pressure slightly increases and injection timing retards. This is due to the lower heating value of natural gas compared to diesel. For the DDF engine, the simulation results of in-cylinder pressure and brake specific fuel consumption varied by Z (0%, 70% and 82%) are almost the same as experimental results. The brake specific fuel consumption and in-cylinder pressure of the simulation differs from the experiment by less than 0.04 and 1.04%, respectively (Table. 4).

Table. 6 Comparison of engine performance between experiment and simulation for DDF at Z=0%, 70% and 82%

Z	Intake pressure	LHV (total)	Pmax (exp)	Pmax (sim)	BSFC (exp)	BSFC (sim)
[%]	[bar]	[MJ/kg]	[bar]	[bar]	[g/kwh]	[g/kwh]
0	1.45	45.6	74.1	74.4	287.6	287.8
70	1.28	71.3	65.5	65.3	501.0	500.6
82	1.27	83.3	64.8	64.9	432.7	437.2

4. Conclusions

The KUBOTA RT140 engine was tested with 50% load while the ambient temperature and pressure, in-cylinder temperature and pressure, heat release rate and lift valve curve were recorded as engine data. Experimental data and obtained results were used to input to calculate the thermodynamics model. Thermodynamics calculation results were calibrated and engine performance i.e. in-cylinder pressure, brake specific fuel consumption, and heat release rate almost matched with the experimental data.

Moreover, the calibrated thermodynamics model of adjustment of advance and retard timing found the same trends as experiment data. In the DDF engine, simulation results of in-cylinder pressure and brake specific fuel consumption varied by Z (0%, 70% and 82%) are almost the same as the experimental results. The results show the obtained thermodynamics model is verified to predict engine performance for both diesel and DDF engines.

5. Acknowledgement

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