

BSFC Improvement by Diesel-Rankine Combined Cycle in the High EGR Rate and High Boosted Diesel Engine

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ABSTRACT

In heavy duty diesel engines, waste heat recovery systems are remarkable means for fuel consumption improvement. In this paper, Diesel-Rankine combined cycle which is combined diesel cycle with Rankine cycle is studied to clarify the quantitative potential of fuel consumption improvement with a high EGR rate and high boosted diesel engine.

The high EGR rate and high boosted diesel engine of a single cylinder research engine was used and it reaches brake specific fuel consumption (BSFC) of 193.3 g/kWh at full load (BMEP=2.0MPa). And its exhaust temperature reaches 370 C. The exhaust gas temperature does not exceed 400 C in high boosted diesel engine even at full load operating condition because of a high excess air ratio. On the other hand, exhaust gas quantity is larger due to a high boosting. So, it is estimated that the thermal energy of exhaust gas is enough for recovery in the high boosted diesel engine, although exhaust gas temperature is not so higher than that of an ordinary diesel engine.

In the heat balance of the high boosted research diesel engine at medium engine speed, the exhaust loss is 38 % at full load. From this result, it is possible to recover the exhaust gas energy, when engine is operated above medium load condition. In this predictive study, water, methanol, toluene, HCFC-123, R134a and R245fa are compared as working fluid in Rankine cycle with superheating. As a result of this study, it is found that Diesel-Rankine combined cycle has a potential to improve BSFC for 2.6 - 3.0 % at full load condition.

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INTRODUCTION

The prevention of global warming is unanimously considered to be of great importance. As a countermeasure, diesel engines have attracted much attention because of their low fuel consumption and reduced CO₂ emission. Research and development for improvement in its fuel consumption have been conducted, in addition to reduction of exhaust emissions such as NO_x and PM [1, 2, 3]. The development of waste heat recovery technology is required to obtain the further improvement in fuel consumption in the future, together with the improvement in engine thermal efficiency, such as engine downsizing.

Waste heat recovery technologies include a turbo-compound system, a thermoelectric generator and a combined cycle with a Rankine cycle. In this study, the combined cycle of a diesel cycle and Rankine cycle is focused as the waste heat recovery technology of a diesel engine for heavy-duty

commercial vehicles. A combined cycle system combines conversions in a high-temperature cycle and a low-temperature cycle in an energy conversion process in series with respect to temperature, to use heat energy effectively. It has been put into practical use to realize high thermal efficiency in a heavy-duty thermal power generation plant using the Brayton cycle of a gas turbine as a high-temperature cycle and the Rankine cycle for a low-temperature cycle [4].

Waste heat recovery by the combined cycle with Rankine cycle in a gasoline engine and diesel engine were reported as application of a combined cycle for automobile engines [5, 6, 7, 8, 9, 10, 11]. However, there are few reports regarding the effect of combined cycle on fuel economy of high-boosted diesel engine for heavy-duty commercial vehicle. This study examines the performance of combined cycle with Rankine cycle which is based on the experimentally obtained result of a high-boosted and high rate EGR heavy-duty single-cylinder

diesel engine. It is a predictive study of fuel consumption improvement by a high-boosted diesel combined cycle.

EXPERIMENTAL SET-UP

Single Cylyndr Test Engine

The experiment is conducted using a four-cycle direct injection single-cylinder diesel engine, of which specifications are presented in [Table 1](#). This experimental engine, presumed as a high-boosted engine for heavy-duty commercial vehicle application, is a single-cylinder engine having a potential to operate with the maximum cylinder pressure $P_{\max} = 30$ MPa with a common rail fuel injection system.

Table 1. Engine specifications

Item	Specifications
Engine Type	DI single cyl. 4 valve
Displacement L	2.004
Bore × Stroke mm	135×140
Max. Engine Speed rpm	2000
Injection System	Common Rail System (Max. Pinj=220MPa)
Nozzle mm	Minisac 0.177×8-150°
Piston Material	Steel (Monotherm)
Comb.Chamber	Shallow dish φ 98
Compression Ratio	16.0
Swirl Ratio	0.87
Aspiration	Externally Supercharging
EGR System	High Pressure Loop

Instrumentation for Measurements

Engine performance was measured using dynamometer (Meidensya Corporation) and MEIDACS-DY6100P. The fuel flow rate was measured using a volumetric fuel flow meter (FP-2000/20; Ono Sokki Company Ltd.). The gaseous emissions were analyzed by an exhaust gas analyzer (MEXA-9100DEGR; Horiba Ltd). The smoke level was measured with a paper filter smoke meter (AVL415S) and was presented as the filter smoke number (FSN).

Fuel and Lubricant

Commercially available low sulfur diesel engine (sulfur content of 10 ppm or less standard) was used. Fuel properties are shown in [Table 2](#).

Table 2. Fuel properties for test

Category	Properties	Category	Properties
Density 15 deg.C	g/cm ³ 0.8279	Elements	C 86.0
Kinematic viscosity 30 C mm ² /s	4.147	mass %	H 13.9
Flash point	C 76.0		O -
Cetane index (JIS K2280)	61.3		N < 0.1
Cetane number	57.4	Components	Saturates 82.9
Distillation	IBP 180.5	Vol. %	Olefins 0
C	5% 212.5		Aromatics 17.1
	10% 229.0		Mono- 15.8
	50% 287.5		Di- 1.0
	90% 337.5		Tri- 0.3
	EP 362.0	Gross calorific value kJ/kg	45890
Sulfur	mass ppm 5	Lower calorific value (Calculated) kJ/kg	43020

EXPERIMENTAL CONDITIONS

The experiments are conducted to ascertain the exhaust heat energies of a high-boosted and high rate EGR research diesel engine. The engine speed is set to $N_e = 1,200$ rpm, which is frequently used for practical automotive engines. [Table 3](#) presents the experimental conditions. The EGR rate was set not less than 45% for NOx reduction, and smoke was set to 0.3 or less in FSN. Intake air temperature was fixed at 50 C. Fuel injection timing was determined in real time by monitoring the rate of heat release so that ignition takes place at the top dead center (TDC), irrespective of either load or the EGR rate. Moreover, this study adopted high-pressure loop EGR system. Experiments were conducted with the same exhaust back pressure as intake manifold pressure because an external supercharger was used. This setup made the engine pump work negative, so that the engine torque could be measured directly using a DC dynamometer. Consequently, BMEP in this experiment included the pumping loss of air intake and exhaust.

Table 3. Engine operating conditions

N_e rpm	BMEP MPa	EGR rate %	Pb kPa	Pinj MPa
1200	2.0	45	401.3	200
	1.6	50	376.3	200
	1.2	55	351.3	200
	0.8	57	276.3	200
	0.4	59	176.3	180
	0.2	66	126.3	150

EXPERIMENTAL RESULTS

Exhaust Emissions and Fuel Consumption

Figure 1 shows exhaust emissions with BMEP at engine speed $N_e = 1200\text{rpm}$. BSNOx was a low level as 0.3-0.6 g/kWh from a light to full load because of high EGR rate. BSHC, BSCO, and smoke were also in low level. Figure 2 indicates BSFC, brake thermal efficiency η_e , exhaust temperature and exhaust mass flow in high-boosted and high rate EGR conditions at $N_e = 1200\text{rpm}$. BSFC improved with the load increase, and BSFC at full load was 193.3 g/kWh, namely brake thermal efficiency η_e was 43.4% at the point. The exhaust gas temperature rises according to the load increase, and reached 369.4 C at full load. Excess air ratio λ was $\lambda = 1.5$ even at full-load operation because of a high boost. Consequently, the maximum exhaust gas temperature was not so high. Whereas the mass flow rate of exhaust gas was large quantity because the boost pressure rises with the load increases. So that it is expected that the thermal energy of exhaust gas is enough for recovery in the high boosted diesel engine, although exhaust gas temperature was not so high than the conventional diesel engine. The exhaust gas temperature of this single cylinder engine was around 400 C at maximum, although the exhaust gas temperature decreases by the turbine work in the multi-cylinder diesel engine with turbo-charger. It is assumed for this study that the exhaust temperature of this single-cylinder engine is equivalent to the exhaust gas turbine outlet temperature of the multi-cylinder diesel engine with turbo-charger. This assumption is adopted for predictive calculations.

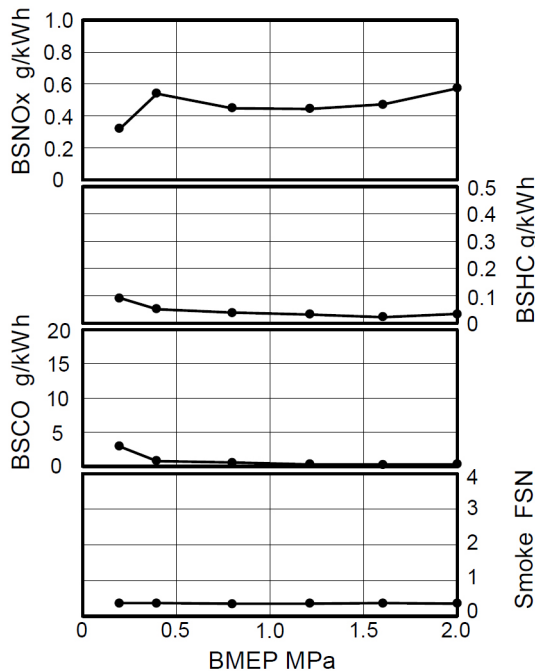


Figure 1. Exhaust emissions in base single cylinder engine

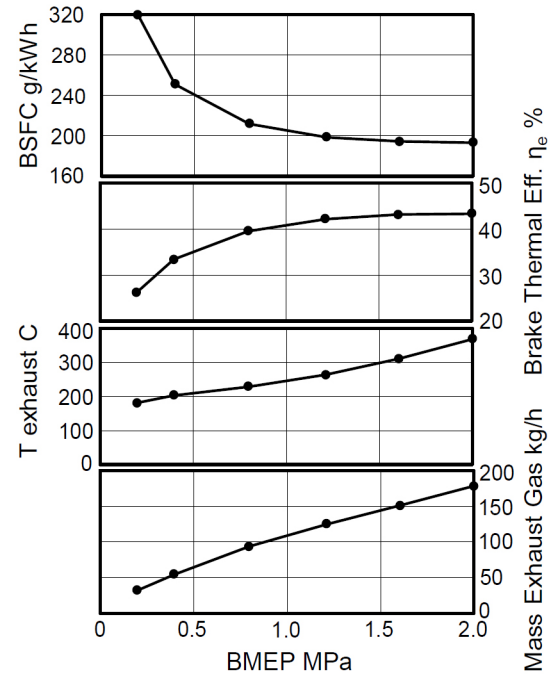


Figure 2. BSFC and exhaust gas temperature in base single cylinder engine

Heat Balance of High-Boosted Diesel Engine

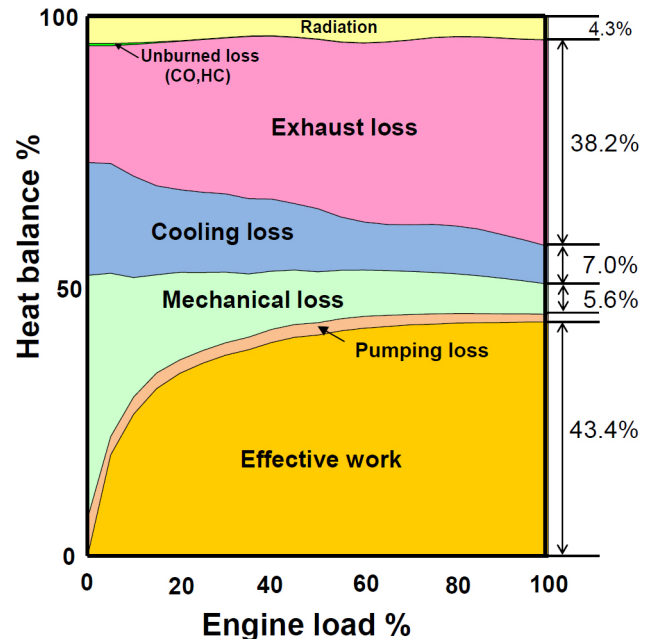


Figure 3. Heat balance of high boosted diesel engine at $N_e=1200\text{rpm}$

Figure 3 shows the heat balance of this high-boosted diesel engine at $N_e = 1,200\text{ rpm}$ with each engine load. The exhaust loss was almost same level compare to the cooling

loss and mechanical loss in the operating conditions of light load up to around 10% load. The exhaust loss increased relatively in operating conditions from middle to high load to cooling loss and mechanical loss, and it was 38.2% at full load. From this result, it is desirable that waste heat recovery implements from middle to high load operation. Waste heat recovery is suitable for highway running conditions in which operating conditions are nearly constant and an engine is mostly operated at middle-high load, rather than urban driving with frequent stop-and-go driving conducted at a light load.

DIESEL-RANKINE COMBINED CYCLE (DRCC)

System Configuration of DRCC

The Diesel-Rankine combined cycle (DRCC) is a combined cycle with a diesel cycle and a Rankine cycle with superheating, as shown in the T-S diagram in Figure 4. Figure 5 presents the DRCC system schema. This study is conducted to clarify the maximum potential of waste heat recovery only from the exhaust gas by DRCC. For this purpose, a heat exchanger is mounted just behind the exhaust manifold of the high-boosted diesel engine, where heat exchange is carried out between high-temperature exhaust gas and a working fluid, and its superheated vapor is generated. It is assumed that the exhaust gas is returned from outlet of the heat exchanger to intake manifold in the high-pressure loop EGR system. High-temperature and high-pressure superheated vapor generated by the heat exchanger is led to an expander, where it is expanded to obtain output power. Then the vapor is introduced to a steam condenser, pressurized with a compressor to return to saturated water, after which it is again led to the heat exchanger.

The cycle-efficiency of a Rankine cycle with superheating η_c is derived from Eqs. (1) and (2) as shown below.

$$\eta_c = \frac{l_{net}}{h_1 - h_4} \quad (1)$$

$$l_{net} = l_{out} - l_{in} \quad (2)$$

In these equations, h_1 is the specific enthalpy of superheated vapor at heat exchanger outlet, h_4 is the specific enthalpy of the working fluid at compressor outlet, l_{net} is specific net work, l_{out} is the specific work of the expander and l_{in} is the specific work of the compressor.

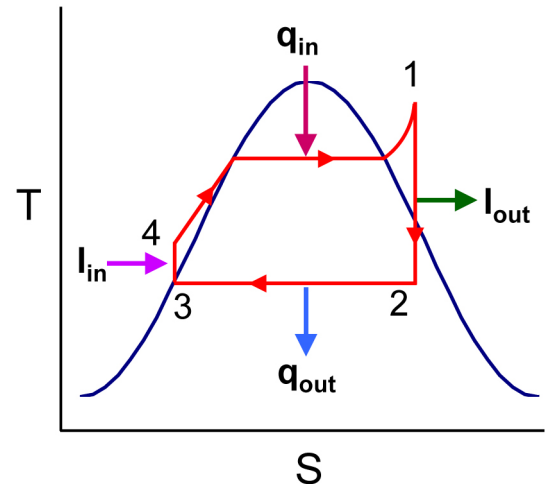


Figure 4. T-S diagram of Rankine cycle with superheating

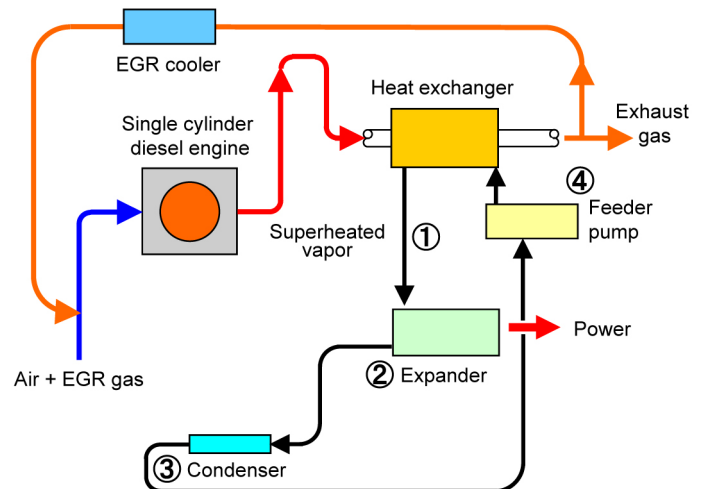


Figure 5. Diesel-Rankine combined cycle system

Operating Condition of DRCC

Basic operating conditions of DRCC are assumed as following.

1. All exhaust gas from a single cylinder engine is introduced into a heat exchanger for heat exchange with a working fluid.
2. Choice of a working fluid is an important in DRCC. This study evaluates water, methanol, toluene, HCFC-123, R134a and R245fa as a working fluid [12, 13] and selects a suitable working fluid for DRCC.
3. The atmospheric pressure is $P_0 = 101.3$ kPa and the atmospheric temperature is $T_0 = 25$ C (298.15 K).
4. Table 4 presents the maximum pressure of superheated vapor P_1 , the maximum temperature of superheated vapor T_1 , outlet pressure of steam condenser P_3 and outlet temperature