

Fundamental Study of Waste Heat Recovery in the High Boosted 6-Cylinder Heavy Duty Diesel Engine

Takuya Yamaguchi
Kurume Institute of Technology

Yuzo Aoyagi, Noboru Uchida, Akira Fukunaga, Masayuki Kobayashi,
Takayuki Adachi, and Munemasa Hashimoto
New Ace Inst Co Ltd

ABSTRACT

In heavy duty diesel engines, the waste heat recovery has attracted much attention as one of the technologies to improve fuel economy further. In this study, the available energy of the waste heat from a high boosted 6-cylinder heavy duty diesel engine which is equipped with a high pressure loop EGR system (HPL-EGR system) and low pressure loop EGR system (LPL-EGR system) was evaluated based on the second law of thermodynamics. The maximum potential of the waste heat recovery for improvement in brake thermal efficiency and the effect of the Rankine combined cycle on fuel economy were estimated for each single-stage turbocharging system (single-stage system) and 2-stage turbocharging system (2-stage system). As a result of estimation, the maximum brake thermal efficiency by the waste heat recovery is theoretically 55.0 % (single-stage turbocharging system) and 53.3% (2-stage turbocharging system) at full load operating condition ($N_e=1600\text{rpm}$), if the available energy of the waste heat in the high boosted diesel engine is converted into the effective work unless considering actual losses. On the other hand, the conversion efficiency from the waste heat to effective work by using the Rankine combined cycle was predicted fairly low. The estimated improvement in fuel consumption by the Rankine combined cycle was 2.7% in single-stage system and 2.9% in 2-stage system, on the condition of the highway cruising at constant speed of 80km/h with a full payload for the heavy duty vehicle (GVW=24980kg).

CITATION: Yamaguchi, T., Aoyagi, Y., Uchida, N., Fukunaga, A. et al., "Fundamental Study of Waste Heat Recovery in the High Boosted 6-cylinder Heavy Duty Diesel Engine," *SAE Int. J. Mater. Manf.* 8(2):2015, doi:10.4271/2015-01-0326.

INTRODUCTION

Much utilization of diesel engines is an effective measure for reducing CO_2 emissions from the transportation because of higher thermal efficiency in a wider operating range compared to other internal combustion engines. Moreover, diesel engines have been strongly required the further improvement in fuel consumption from the cost cut and energy security view point. In recent years, the waste heat recovery technologies such as turbo-compound system, thermoelectric element and the Rankine combined cycle have attracted much attention as the way of improvement in fuel consumption besides the downsizing and down-speeding of engines [1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12].

In recent years, 2-stage turbocharging system was applied to heavy duty diesel engines to achieve higher BMEP (brake mean effective pressure) in the low engine speed region which is low mechanical friction loss. [13, 14] Also, the advanced EGR system which is combined HPL-EGR system with LPL-EGR system was investigated

to obtain lower fuel consumption and lower exhaust emissions (NO_x and PM) simultaneously. [15, 16] The waste heat carried by exhaust gas is predicted to vary depending on such the turbocharging system and EGR system in the high boosted diesel engine, therefore the potential of the waste heat recovery for improvement in fuel consumption should be assumed considering the specific turbocharging system and EGR system. However, there are few reports regarding the effect of turbocharging system and EGR system on the potential of the waste heat recovery in the high boosted diesel engine. In this study, the available energy of the waste heat from a high boosted diesel engine which is equipped with single-stage turbocharging system (single-stage system) and 2-stage turbocharging system (2-stage system) was analyzed based on the second law thermodynamics analysis and the maximum potential of the waste heat recovery for improvement in thermal efficiency was estimated for single stage system and 2-stage system. In addition, the effect of the waste heat recovery using the Rankine combined cycle on fuel consumption was also predicted quantitatively.

EXPERIMENTAL SET-UP

6-Cylinder Test Engine

The experiment was conducted by means of a four stroke direct injection inline 6-cylinder diesel engine having the displacement of 10.5Litres. Table 1 shows the major engine specifications. This engine was equipped with a common rail fuel system (the maximum injection pressure is 220MPa). Figure 1 and Figure 2 show schematic diagram of the single-turbocharging engine system and the 2-stage-turbocharging engine system respectively. Both turbocharging systems were equipped with the advanced EGR system. In the 2stage-turbocharging engine system, two turbochargers were arranged in series. The high pressure stage turbocharger was mounted upstream the exhaust path and the low pressure stage turbocharger was located downstream the exhaust path. In this system the high pressure stage turbocharger and the low pressure stage turbocharger were a different size of variable geometry turbocharger. The compressed gas at downstream of each compressor was cooled by heat exchanger, i.e. intercooler and aftercooler, to reduce intake temperature.

Table 1. Engine specifications

Item	Specifications
Engine type	DI inline 6
Displacement	cm ³ 10520
Bore × Stroke	mm 122×150
Max. engine speed	rpm 2000
Injection system	Common rail system (Max. Pinj=220MPa)
Nozzle	mm Minisac 0.173×8-155°
Piston material	FCD (Ferrum Casting Ductile)
Combustion chamber	Shallow dish
Compression ratio	17.0
Swirl ratio	1.0
EGR system	HPL & LPL EGR system

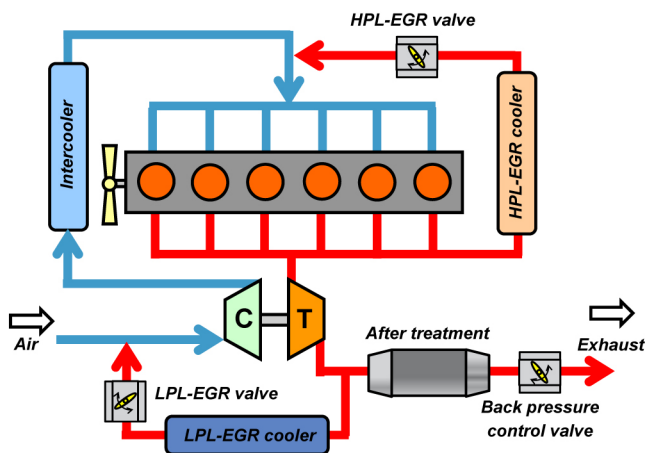


Figure 1. Schematic of engine system (Single-stage system)

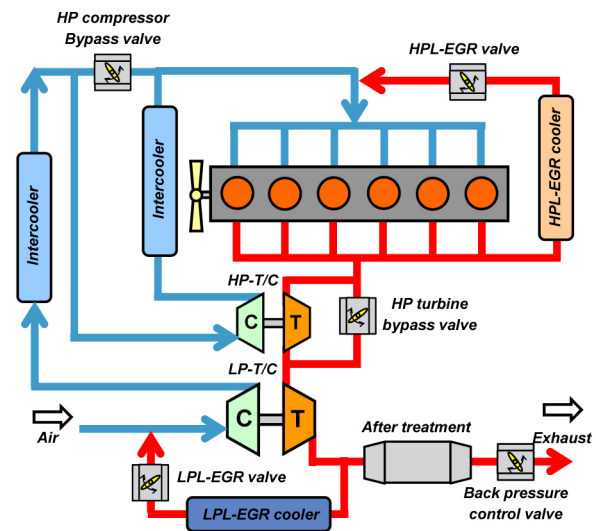


Figure 2. Schematic of engine system (2-stage system)

Instrumentation for Measurements

The engine performance was measured using a low inertia dynamometer (TOYO ELECTRIC MFG. CO. LTD.). The fuel flow rate was measured using a volumetric fuel flow meter (FP-2000/20; Ono Sokki Company Ltd.). The gaseous emissions were analyzed using an exhaust gas analyzer (MEXA-7100 DEGR; Horiba Ltd.). The smoke level was measured by a smoke meter (GSM-10; Tsukasa Sokken Co., Ltd.) Combustion analysis to assess the rate of heat release and others was performed by means of an analyzer (DS-2000 series; Ono Sokki Co. Ltd.) and an in-cylinder pressure transducer (Type 6043A; Kistler Instruments AG).

Fuel and Lubricant

Commercially available low-sulfur (less than 10ppm) diesel fuel in Japan was used. The sulfur content of the used fuel was 5 ppm. And Cetane number was 57.4. Fuel properties are shown in Table 2. Low-sulfur engine oil containing 0.26 mass % of sulfur was used for lubricant oil and its properties were listed in Table 3.

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
	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

Table 3. Lubricant oil properties

Category		Properties
Density 15 deg.C	g/cm ³	0.859
Flash point (COC)	°C	226
Kinematic viscosity	40°C	68.82
	100°C	10.55
Pour point	°C	-35.0
Sulfuric ash content mass	%	1.00
Sulfur mass	%	0.26

EXPERIMENTAL CONDITIONS

The effect of the waste heat recovery on the improvement in fuel consumption of a high boosted diesel engine equipped with both HPL-EGR and LPL-EGR system was evaluated at full load operating condition (Engine speed was from Ne=600 to Ne= 2000rpm). Table 4 shows the experimental conditions of the single-stage system and the 2-stage system at full load operating condition.

Table 4. Experimental conditions at full load operations

Ne rpm	Single-stage turbocharging			2-stage turbocharging		
	BMEP MPa	Pb kPa	EGR rate %	BMEP MPa	Pb kPa	EGR rate %
600	0.85	113.6	19.8	1.42	185.0	20.2
800	1.24	162.6	25.5	1.85	255.6	27.2
1000	2.08	268.3	25.4	2.42	331.5	25.9
1200	2.15	280.1	23.7	2.41	349.9	23.0
1400	2.20	302.0	24.5	2.42	359.4	20.3
1600	2.04	284.0	30.6	2.13	304.1	21.0
1800	1.87	255.7	23.7	1.89	289.7	21.8
2000	1.70	247.4	20.6	1.72	272.3	17.8

EXPERIMENTAL RESULTS IN THE HIGH BOOSTED DIESEL ENGINE

Difference of Boost Pressure and EGR Rate by Turbocharging System

Figure 3 shows the boost pressure (Pb) and BMEP of the single-stage system and the 2-stage system at full load operating conditions. The boost pressure of the 2-stage system was increased remarkably from low engine speed to middle engine speed compared with the single-stage system. BMEP of the 2-stage system was expanded compared to the single-stage system, as a consequence of increase in boost pressure by the 2-stage system. Particularly, the increase in BMEP was remarkable at low to middle engine speed.

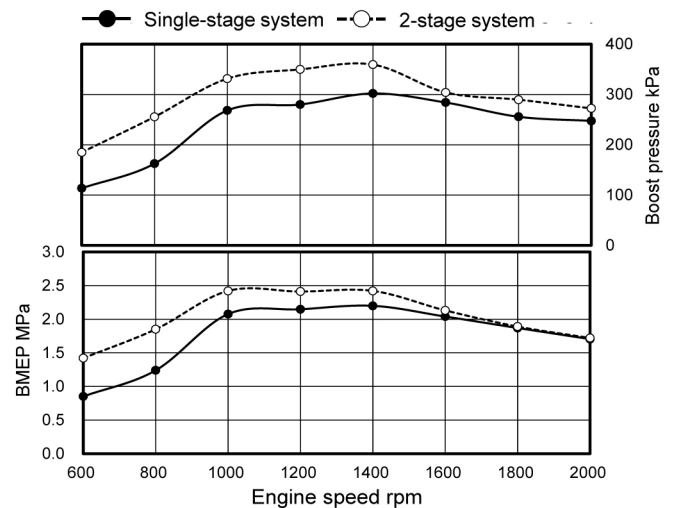


Figure 3. BMEP and boost pressure at full load operating condition

Figure 4 indicates the EGR rate and the volumetric fraction of HPL-EGR with engine speed at full load operating condition. The volumetric fraction of HPR-EGR was defined by Eqs. (1).

$$\text{Vol. frac. of HPL-EGR} = \frac{\text{Vol. of HPL-EGR}}{\text{Vol. of HPL-EGR} + \text{Vol. of LPL-EGR}} \quad (1)$$

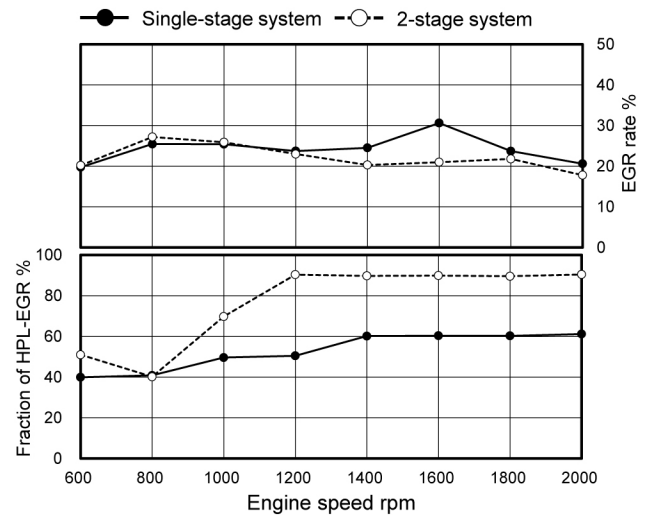


Figure 4. EGR rate and volumetric fraction of HPL-EGR with engine speed

The EGR rate in the 2-stage system was achieved 20-27% from low engine speed to middle engine speed. Although full load BMEP of the 2-stage system was higher than that of the single-stage system around low engine speed condition, EGR rate in the 2-stage system was almost same level as the single-stage system due to high boost pressure. On the other hand, the 2-stage system's EGR rate was lower than that of the single-stage system from middle engine speeds to high engine speeds.

The volumetric fraction of HPL-EGR was different with respect to individual turbocharging systems. The volumetric fraction of HPL-EGR in the 2-stage system was higher than that of the single-stage system except for the measurement at $N_e=800$ rpm. Since the high pressure stage turbocharger was mounted upstream the exhaust path to increase boost pressure around low engine speed region in the 2-stage system, the HPL-EGR system is easier to introduce EGR gas into the cylinder by a higher back pressure.

Gas Temperature at Full Load Operating Condition

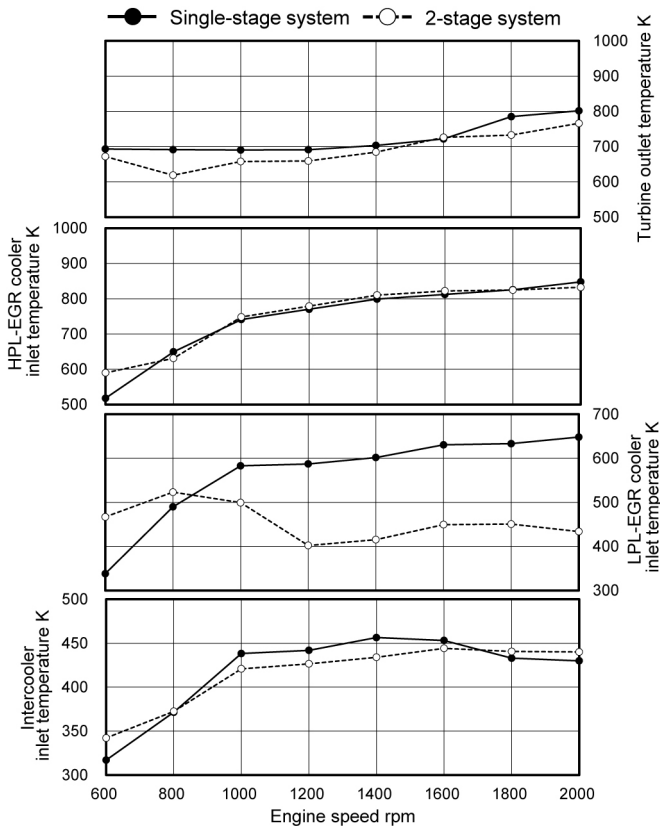


Figure 5. Gas temperature at turbine outlet, HPL-EGR and LPL-EGR cooler

Figure 5 compares variation of the gas temperatures at the turbine outlet, HPL-EGR and LPL EGR cooler and intercooler inlet with engine speed in between the single-stage system and the 2-stage system. At the turbine outlet in the single-stage system, the exhaust gas temperatures were 690–800K for engine speeds over $N_e=1000$ rpm. In the 2-stage system, the exhaust gas temperatures at the turbine outlet were 670–760K at over $N_e=1000$ rpm. The exhaust gas temperatures at turbine outlet in the 2-stage system were 20–70K lower than the exhaust gas temperatures at turbine outlet in single-stage system because more exhaust gas energy was consumed in a 2-stage system compared to single-stage system. The exhaust gas temperatures at the HPL-EGR cooler inlet were 740–850 K at over $N_e=1000$ rpm in both the single-stage system and the 2-stage system. The exhaust gas flow into the HPL-EGR cooler was led from the exhaust manifold to the HPL-EGR cooler directly, so there was no temperature reduction by exhaust turbine work. Hence, exhaust gas temperatures at the HPL-EGR cooler were 50–125K higher than exhaust gas temperatures at the turbine outlet at middle engine speed

($N_e=1000$ –1400rpm). Exhaust gas at the HPL-EGR cooler inlet was the heat source with the highest temperature for the waste heat recovery in the high boosted diesel engine. The exhaust gas temperature at the LPL-EGR cooler inlet in the single-stage system was about 600 K at over $N_e=1000$ rpm. It was 200 K lower than the exhaust gas temperature at the HPL-EGR cooler inlet in the single-stage system. The charge temperatures at intercooler inlet was 420–450 K at over $N_e=1000$ rpm in both turbocharging systems. This was above 150 K lower than the exhaust gas temperature at the HPL-EGR cooler and the LPL-EGR cooler. Therefore, the waste heat at the intercooler was the heat source with the lowest temperature for the waste heat recovery in the high boosted diesel engine.

Exhaust Gas and Charge Mass Flow Rate at Full Load Operating Condition

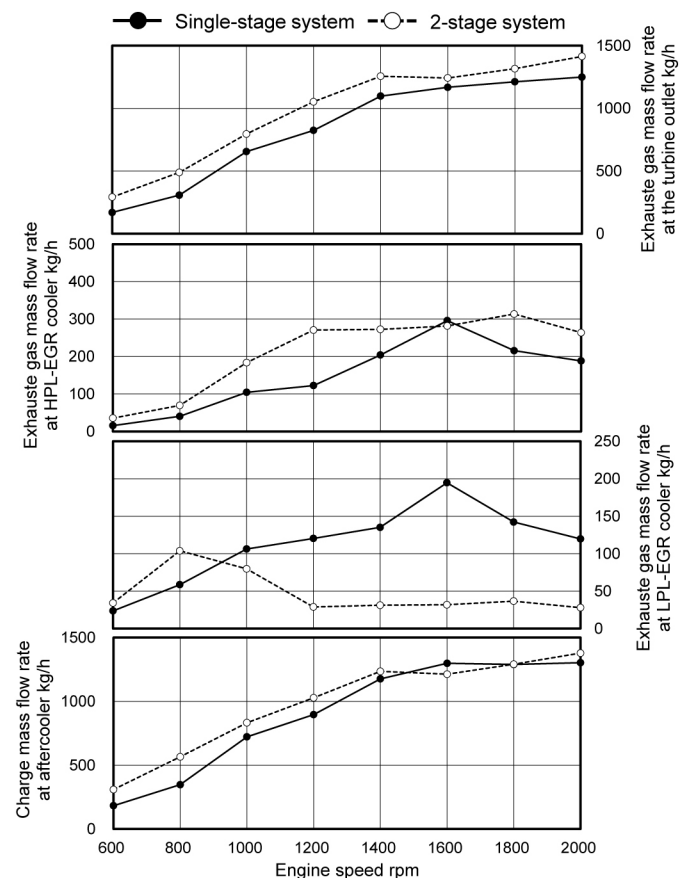


Figure 6. Gas mass flow rates at the turbine outlet, HPL-EGR and LPL EGR cooler and intercooler

Figure 6 compares variation of gas mass flow rates at the turbine outlet, HPL-EGR and LPL EGR cooler and intercooler inlet with engine speed in between the single-stage system and the 2-stage system. The exhaust mass flow rates of the 2-stage system were larger than those of the single-stage system over the all engine speeds. Since the boost pressure of the 2-stage system was higher than that of the single-stage system at full load operating condition, the exhaust gas mass flow rates at the HPL-EGR cooler were larger in the 2-stage system, as the volumetric fraction of the HPL-EGR was higher in the 2-stage system than that of the single stage system. Additionally,

Exhaust gas mass flow rates at the LPL-EGR cooler in the 2-stage system was largely lower than the exhaust mass flow rates at the LPL-EGR cooler in the single-stage system at over Ne=1000rpm. This is because fraction of HPL-EGR in the 2-stage system was higher than that of the single-stage system. Especially, the exhaust gas mass flows rates and the exhaust gas temperature at the LPL-EGR cooler in the 2-stage system was low at over Ne=1200 rpm.

THE AVAILABLE ENERGY ANALYSIS IN THE HIGH BOOSTED DIESEL ENGINE

In the high boosted diesel engine equipped with advanced EGR system, waste heat at the turbine outlet, HPL and LPL-EGR cooler as well as the intercooler can be considered as heat source of the waste heat recovery. In this section, the available energy of these waste heats in the single-stage system and the 2-stage system is analyzed based on the second law thermodynamics analysis. Moreover, the potential of the waste heat recovery for improvement in fuel consumption is evaluated.

The Waste Heat Recovery System after the Exhaust Turbine

Figure 7 shows a schematic of the part of waste heat recovery system after the exhaust turbine. Exhaust gas passed through the exhaust turbine is assumed to flow into the virtual heat exchanger which is mounted just behind the branch point to the LPL-EGR system. In this study, the exhaust gas temperature at the virtual heat exchanger was considered as same as the exhaust gas temperature at the exhaust turbine outlet. Additionally, the exhaust gas temperature at the outlet of the virtual heat exchanger was assumed to be 423.15K (150°C) to evaluate the potential of the waste heat recovery for improvement in thermal efficiency, although higher exhaust gas temperature should be required to convert NOx and PM emissions by using after treatment catalysts in a practical heavy duty diesel engine.

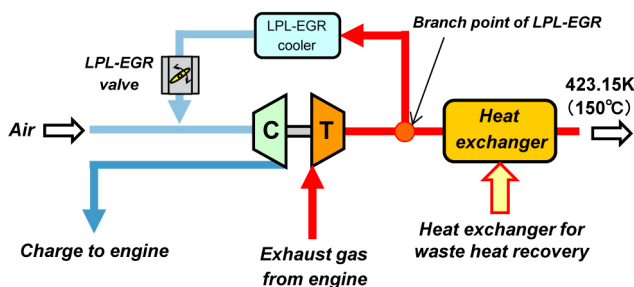


Figure 7. Arrange of the heat exchanger at exhaust process

The Total Available Energy of the Waste Heat in the High Boosted Diesel Engine

Figure 8 and Figure 9 indicate the total available energy of the waste heat from a high boosted diesel engine in the single-stage system and the 2-stage system with engine speed. The available energy at the heat exchanger HPL-EGR and LPL-EGR cooler, intercooler was derived from Eqs. (2), Eqs. (3) and Eqs. (4) as shown below[17]. In

these equations, m is gas mass flow rates at the heat exchanger or each cooler [kg/s], T_0 is standard temperature (298.15 K), T_{in} is the gas temperature at the inlet of the heat exchanger, HPL-EGR cooler, LPL-EGR cooler and intercooler. T_{out} is the gas temperature at outlet of heat exchanger, HPL-EGR cooler, LPL-EGR cooler and intercooler. Specific heat at constant pressure of the gas C_p in the heat exchanger, HPL-EGR cooler, LPL-EGR cooler and intercooler was calculated using gas temperature and gas composition [18]. Additionally, the ratio of available energy ε was defined as Eqs. (5).

$$E_m = m \cdot C_p [(T_m - T_0) - T_0 \cdot \ln(T_m / T_0)] \quad (2)$$

$$E_{out} = m \cdot C_p [(T_{out} - T_0) - T_0 \cdot \ln(T_{out} / T_0)] \quad (3)$$

$$E = E_m - E_{out} \quad (4)$$

$$\varepsilon = \frac{E}{m \cdot C_p \cdot (T_{in} - T_{out})} \quad (5)$$

The total of available energy of the waste heat increased as engine speed increased, and it became the maximum at Ne=2000 rpm. The total available energy of the waste heat was 108.0 kW in the single-stage system and 108.4 kW in the 2-stage system at Ne=2000rpm. Additionally, the total available energy of the waste heat was 40.0-74.8 kW in the single-stage system and 43.0-76.3 kW in the 2-stage system at middle engine speed region (Ne=1000-1400rpm), which are frequently used for practical heavy duty diesel engine. The available energy of waste heat at the LPL-EGR cooler was lower than the other waste heat sources in the high boosted diesel engine. Especially, the available energy of waste heat at the LPL-EGR cooler in the 2-stage system was extremely low Since the volumetric fraction of the HPL-EGR in 2-stage system was higher than that of the single-stage system as described above. The available energy of the waste heat at the heat exchanger and the HPL-EGR cooler accounted for large proportion in both turbocharging system. These analysis results indicate that the waste heat of the heat exchanger and the HPL-EGR cooler is the main heat source for the waste heat recovery in the high boosted diesel engine.

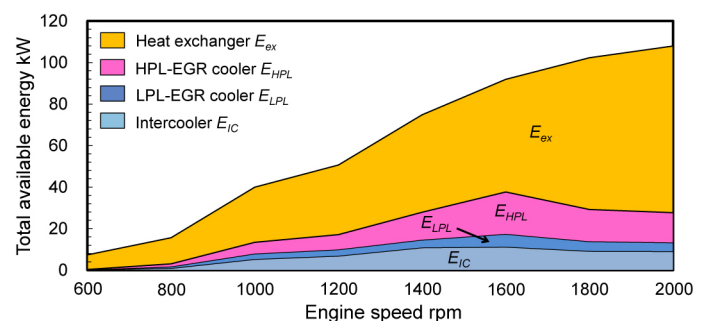


Figure 8. The total available energy of the waste heat from engine at full load operating condition (Single-stage system)

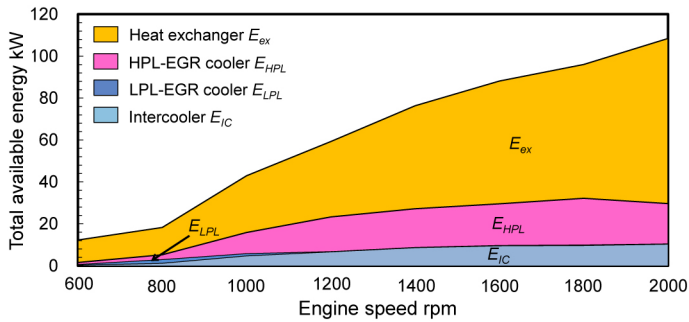


Figure 9. The total available energy of the waste heat from engine at full load operating condition (2-stage system)

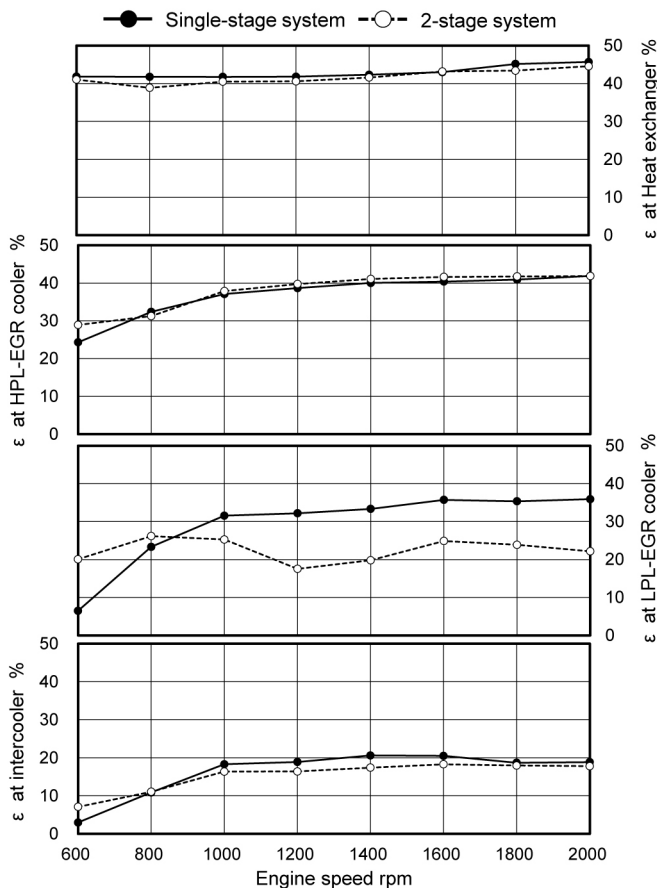


Figure 10. The ratio of available energy ϵ at the heat exchanger HPL-EGR cooler, LPL-EGR cooler and intercooler in the single-stage system and the 2-stage system with engine speed.

Figure 10 indicates the ratio of available energy ϵ at the heat exchanger HPL-EGR cooler, LPL-EGR cooler and intercooler in the single-stage system and the 2-stage system with engine speed. The ratio of available energy at the heat exchanger was 40-45 % in both turbocharging system. Also, the ratio of available energy at the HPL-EGR cooler was about 40 % in both turbocharging systems. These indicate that about 40 % of the waste heat at the heat exchanger and the HPL-EGR cooler is available to be converted into effective work. The ratio of available energy at the LPL-EGR cooler was 30-35 % in the single-stage system and 17-25 % in the 2-stage system at over $N_e=1000$ rpm. The waste heat at the LPL-EGR cooler was not expected to be a better heat source for the waste heat recovery than the waste heat at the HPL-EGR cooler, because the

exhaust gas temperature at the LPL-EGR cooler was lower than the exhaust gas temperature at the heat exchanger and the HPL-EGR cooler. The ratio of available energy at the intercooler was 16-18 % in the single-stage system and the 2-stage system. Because the gas temperatures at the intercooler was lower than other heat source in the high boosted diesel engine. Therefore the waste heat at intercooler was not better waste heat source either.

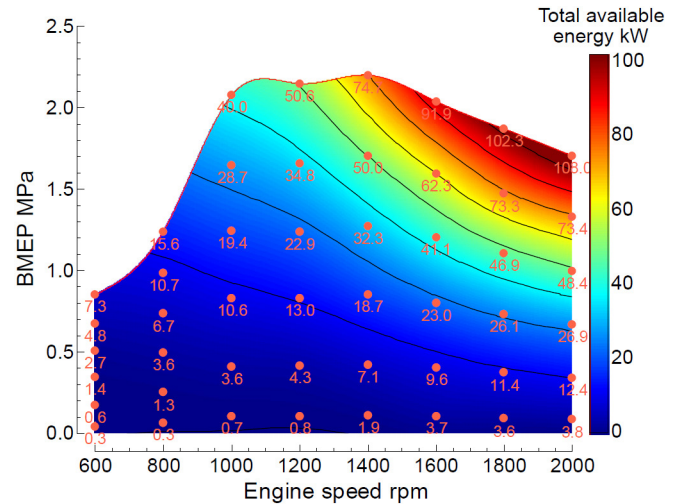


Figure 11. Map of the total available energy (Single-stage system)

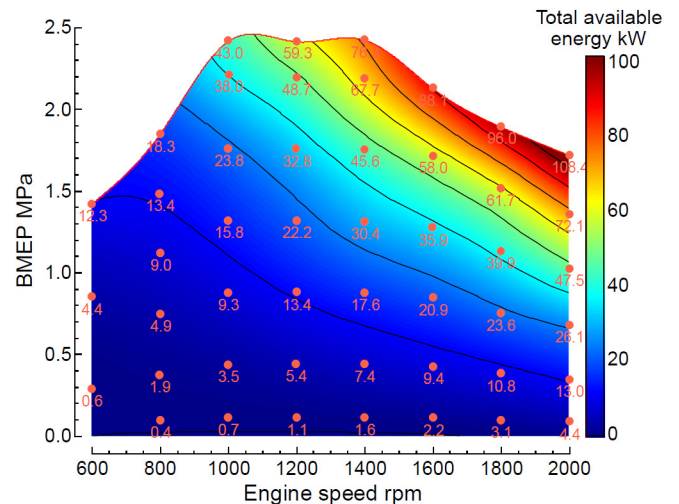


Figure 12. Map of the total available energy (2-stage system)

Figure 11 and Figure 12 show the map of the total available energy of the waste heat for the wide range engine operating conditions in the single-stage system and the 2-stage system. (Please refer to Table A1 and Table A2 in Appendix regarding the detail of experimental condition.) The total available energy of the waste heat was large at high load and high engine speed region. On the other hand, the total available energy of the waste heat at partial load and low engine speed region in both turbocharging system was low level compare with high load and high engine speed region. That's why it is expected that the potential of the waste heat recovery for improvement in fuel consumption decreases remarkably compare to full load operating condition. Therefore the waste heat recovery in high boosted heavy duty diesel engines is suitable for the highway running condition.