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Gao, J, Chen, H, Tian, G et al. (2 more authors) (2019) An analysis of energy flow in a turbocharged diesel engine of a heavy truck and potentials of improving fuel economy and reducing exhaust emissions. Energy Conversion and Management, 184. pp. 456-465. ISSN 0196-8904

https://doi.org/10.1016/j.enconman.2019.01.053

An analysis of energy flow in a turbocharged diesel engine and potentials of improving fuel economy and reducing exhaust emissions

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Abstract:

The impetus of the internal combustion engine developments is the reductions of the fuel consumptions and exhaust emissions. Thermal management is an efficient method to decrease the exhaust emissions and enhance fuel economy. In order to further optimize the thermal management of internal combustion engines, a detailed analysis of the energy flow in each component of internal combustion engines is indispensable. In this paper, the test bench of a heavy duty diesel engine was established to obtain the target parameters. The energy distributions in each component of the diesel engine, including compressor, intercooler, shaft power, turbine, coolant and exhaust, were calculated using tested parameters. The lubricating oil consumption was also taken into consideration. In addition, the potential influences of different turbochargers on the total thermal efficiency were analyzed. The results showed that the thermal efficiency of the diesel engine was more than 38% when the engine operated at 50%~100% engine load and 1000 rpm~1700 rpm conditions. The energy loss by coolant was more than 50% of the total fuel energy consumption in the low power output conditions. However, it was lower than 30% in high power output conditions, and the thermal loss was more than 150 kW around rated power conditions. The maximum proportion of the energy being consumed by turbine was ~10% of the fuel energy; additionally, the exhaust energy distributions changed significantly after the turbine expansion. 1%~3% of the fuel energy was recycled by the turbocharger, then, flowed into the cylinders. The energy loss through the intercooler accounted for ~6% of the fuel energy. Significant reductions of exhaust emissions and fuel consumptions can be achieved by optimizing the coolant and lubricating oil thermal conditions. Turbochargers presented a huge effect on exhaust temperature distributions at high power output conditions, and the total thermal efficiency changed significantly if all kinds of energy recovery approaches were applied.

Keywords: diesel engines; energy distributions; thermal management; fuel economy; exhaust emissions

1. Introduction

As the main power sources of the vehicles, internal combustion engines have attracted much attention. Internal combustion engine powered vehicles will dominate the vehicle productions in the following years, although electric and hybrid vehicles have more and more shares in the vehicle market. Nevertheless, internal combustion engines encounter the challenges of decreasing exhaust emissions and meeting the 95 g/km CO_2 emission targets in 2020 [1, 2]. Much energy is wasted in the engine operation process by friction, coolant, exhaust and intercooler etc [3-6]. The energy percentages in gasoline engines are ~40%, ~30% and ~25% for exhaust, coolant and effective power output, respectively, while the power output is ~35% for diesel engines. The energy loss is mainly in the form of heat; also, the energy grades of the waste heat are different, which causes the systems complicate to recycle the waste heat.

Compared with the coolant heat, exhaust heat is at a high energy grade level, which has high potentials of increasing the combined thermal efficiency of the engine and energy recovery systems. Advanced technologies were used to recover exhaust energy, such as the thermoelectric generators [7, 8], organic Rankine cycle (ORC) [9-11], the six-stroke cycle internal combustion engines [12, 13] and novel turbochargers [14-16]. As the most successful technology of exhaust energy recovery, turbochargers were used to increase the mass flow rate of fresh air for achieving a high energy power density [17]. In order to meet the requirements of variable engine operation conditions on the turbine expansion ratio, waste gate, variable nozzle turbines and variable geometric turbines [18] were used, which significantly increased energy recovery efficiency, especially at low speed conditions. The energy recycling percentage was closely related to turbine and compressor working conditions, which made it important to excellently match the turbocharger with the engine. Although turbocharger can effectively make use of exhaust energy, most of the exhaust energy flowed into the atmosphere in the status of heat.

The researches about the exhaust and coolant thermal energy recovery are mainly focused on ORC technology, which has the advantages of low requirements for energy grade level. Hou et al [19] combined free piston expander-linear generator and ORC to recycle exhaust thermal energy. This device was characterized with a small volume and a high power density, which made a huge step of applying ORC to passenger vehicles. What's more, the maximum energy conversion efficiency could reach 73.33% [20]. A dual-loop ORC system [21] was applied to achieve a higher energy recycle efficiency. The net power output of a dual-loop ORC system reached 115.1 kW, which led to a 11.6% increase of the engine power output. Compared with the exhaust heat, coolant heat was at a low energy grade level, but the same order of magnitude in waste heat quantity. In order to effectively recycle the coolant heat, CO_2-based transcritical Rankine cycle (CTRC) was researched [22]. Boretti [23] used the ORC system to recycle the coolant heat of a 1.8 L naturally aspirated gasoline engine, which was the power of a hybrid vehicle. Coolant ORC system could enhance the fuel economy by averaged 1.7%, with a maximum value of 2.8%, which was lower than half of the exhaust ORC system. In order to enhance the ORC system efficiency, the coolant was used to pre-heat the working fluid of ORC system, as researched by Shu et al [24] who introduced an improved CTRC system containing a pre-heater (namely, the intercooler in diesel engines) and a regenerator (PR-CTRC). The net power output was 9.0 kW for a 43.8 kW engine, whose power output increased by ~50% compared with the CTRC system. However, it had great challenges for the ORC systems to be applied to the vehicles, which was partly caused by the huge size of the systems and variable working conditions of internal combustion engines in the real driving conditions. The dynamic engine operation conditions led to the working point variations of the ORC components. Zhao et al [25] investigated the control strategy of the engine and ORC combined system, where the optimal rotation speed of the fluid pump changed with engine conditions. Variable-speed working fluid pumps could significantly increase the energy recover efficiency of the ORC system. Jiménez-Arreola et al [26] analyzed the dynamic behaviors of the heat exchanger used in the ORC system, where the louver fin multi-port flat tube evaporators had a shorter response time compared with fin and tube evaporators, but penalty of high pressure drops of exhaust and working fluid.

In most cases, coolant flow rate was higher than demanded values, which led to a lower engine thermal efficiency and more exhaust emissions [27]. It was caused by the fact that the coolant flow rate was proportional to the engine speed for conventional coolant pumps. So that the coolant and lubricating oil were over-cooled for ~95% operation time [28]. In addition, the precisions of coolant and lubricating oil temperature control were poor. Lower coolant and lubricating oil temperatures meant poorer in-cylinder combustion, more serious cylinder quenching effect and higher friction loss. Reference [29] demonstrated that 2%~5% fuel consumption drop, 10% HC and 20% CO emission reductions were obtained when electric coolant pump was used to enhance the coolant temperature from 90 °C to 110 °C. In reference [30], exhaust heat was stored in a heat storage material, then, the heat was used to enhance the coolant and lubricating oil thermal condition when necessary. This method effectively recycled parts of exhaust heat, also, decreased exhaust emissions and friction loss. CO and HC emission reductions were 64% and 15%, respectively.

In order to recycle the energy of coolant, exhaust and intercooler with high efficiency, further, to improve the fuel economy and decrease exhaust emissions, the energy flow in each component of the internal combustion engine should be analyzed individually. Based on the energy flow, the potentials of fuel economy improvement and emission reductions could be further analyzed. Luo and Sun [31] researched the effect of the engine operation parameters on the energy flow in a turbocharged hydrogen engine. Power output, coolant heat, lubricating oil heat, intercooler heat, exhaust energy and missing energy were investigated. Rakopoulos and Giakoumis [32] used the second-law to analyze the energy balance of a internal combustion engines, also, indicated that turbocharger was an excellent second-law process to increase the engine power density. Turbocharger system coupled the intake air and exhaust systems, which significantly influenced the energy flow in diesel engines.

To authors' knowledge, a majority of the references [33-35] about the energy flow in internal combustion engines united the turbocharger and engine. Literatures are limited to date about the individual analysis of the energy flow in the engine and turbocharger. Also, the recycled energy by turbocharger decreased greatly due to the heat loss through the intercooler. However, the researches about the recycled energy flowing into the cylinders are few to refer. In addition, the heat distributions of the intercooler were seldom investigated. The lubricating oil consumption during engine operations was also neglected in many researches.

In this paper, the engine test bench was set up. Energy flow in the combustion process was calculated based on the experimental data, with lubricating oil consumption being taken into consideration to analyze the energy flow with a high precision. Different from previous studies, the energy distributions in the engine and turbocharger were analyzed separately, which laid the foundations of enhancing energy recovery efficiency. The recycled energy by turbocharger after the intake air going through the intercooler was obtained. Then, the potentials of reducing exhaust emissions and fuel consumptions were estimated. Further, the effect of different turbochargers on the total thermal efficiency was investigated. The test bench refers to operative conditions ranging between 20% and 100% of the load and between 800rpm and 2100 rpm.

2. Experimental section

The demonstrator used in this study was a turbocharged heavy duty diesel engine; the specifications of the diesel engine are shown in Table 1. Figure 1 shows the performances of the diesel engine at full load conditions. The engine speed corresponding to the minimum brake specific fuel consumption (BSFC) was ~1250 r/min, and BSFC increased dramatically when the engine speed was deviated from that value.

Table 1 Specifications of the diesel engine

Items	Content
Engine type	In-line six cylinders, four-stroke
Max power/ kW	258
Max torque/ N·m	1450
Displacement/ L	8.6
Cylinder stroke/ mm	112
Cylinder bore/ mm	145
Compression ratio	17.5
Intake type	Turbocharged intercooler
Valve number per cylinder	4

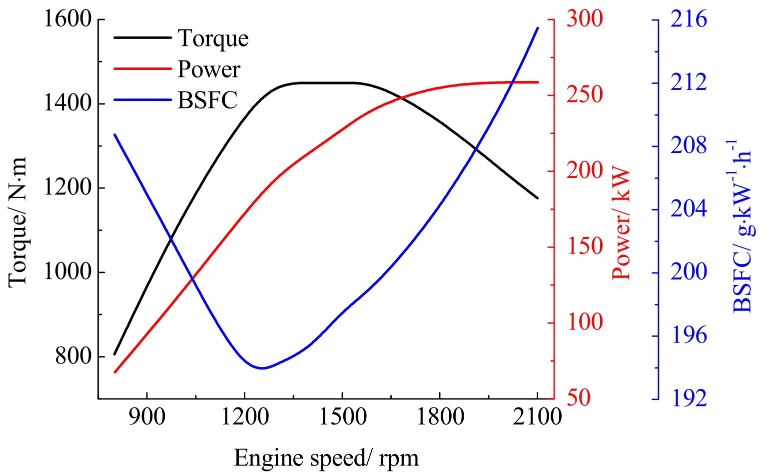


Figure 1 Performances of the diesel engine at full load conditions

Figure 2 shows the experimental system layout. The diesel engine was coupled with a dynamometer. Temperatures and pressures were tested at the target points, such as compressor inlet and outlet, turbine inlet and out let, intercooler outlet, coolant channel inlet and outlet, and oil sump. Coolant, air flow rates and fuel consumption rate were also measured in the experiment. The temperature, pressure and fluid flow rate were collected using a computer. The atmospheric temperature and pressure were 20 °C and 1.0 bar, respectively. The engine operated for no less than 20 minutes to ensure the engine fully warmed up after it started. It should be guaranteed that the coolant and oil temperatures were stabilized when the data (temperature, pressure and fluid flow rate) was collected.

Instability of coolant and oil temperature could cause significant fluctuations of heat transfer, friction loss and engine power output, which would influence the energy distributions in the diesel engine. The lubricating oil consumption was considered as 0.2% of fuel consumption, which met the standard of the engine manufacturers.

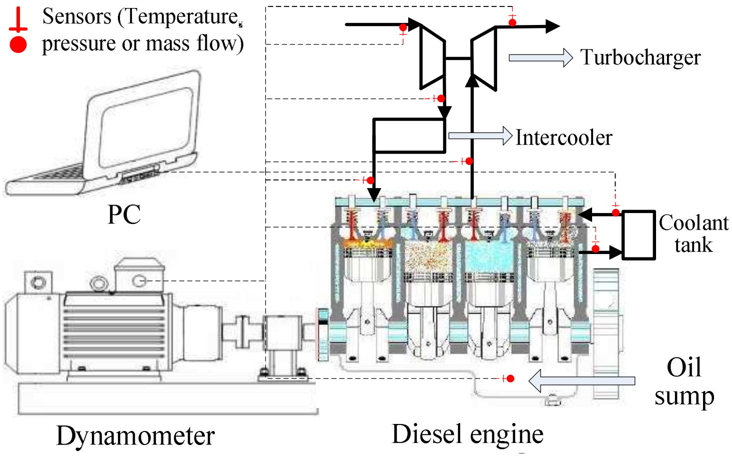


Figure 2 Layout of the experimental system

3. Energy distribution calculations

The temperature, pressure and mass flow rate collected in Section 2 were used to calculate the energy distributions of each component in the whole combustion process. A control volume was assumed around the engine to analysis the energy balance. Intercooler and turbocharger were contained in the control volume, rather than the after-treatments. Fresh air, fuel and lubricating oil were injected into the control volume; meanwhile, heat, torque and exhaust flowed out of the control volume. The energy flow in a turbocharged diesel engine is shown in Figure 3, and the exhaust gas recirculation (EGR) was neglected in the figure. Some process was considered as ideal conditions, for example, the fuel injection and combustion was momentarily completed at the top dead center (TDC), as shown in Figure 3. In the process of intake and compression strokes, the heat transferring into the atmosphere was neglected due to its small value. Different from the naturally aspired engine, parts of exhaust energy were recycled for the turbocharged engine, which complicated the energy flow due to the turbocharger coupling the intake air and exhaust systems.

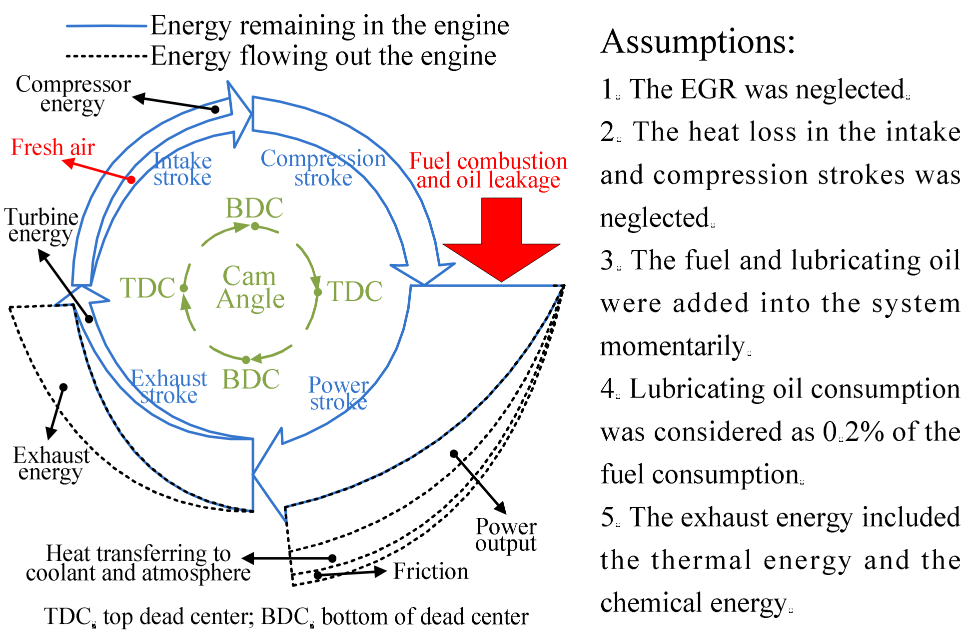


Figure 3 Energy flow in the turbocharged diesel engine

Based on the first law of thermodynamics, the energy balance in the diesel engine is shown in Equation 1,

$$\dot{Q}_f + \dot{Q}_a + \dot{Q}_{cc} = \dot{Q}_p + \dot{Q}_e + \dot{Q}_c + \dot{Q}_i + \dot{Q}_u + \dot{Q}_T \qquad (1)$$

Where, $\dot{Q}_f$, chemical energy of the fuel (including diesel fuel and leaked lubricating oil); $\dot{Q}_a$, energy of the fresh air; $\dot{Q}_{cc}$, energy consumption by the compressor; $\dot{Q}_p$, power output through shaft; $\dot{Q}_e$, exhaust energy flow out of the system; $\dot{Q}_c$, heat transferring to the coolant; $\dot{Q}_i$, heat transferring to the intercooler; $\dot{Q}_T$, energy generated by the turbine; $\dot{Q}_u$, unaccounted heat loss. Unaccounted heat loss refers to the heat transferring to the atmosphere by the convection and radiation transfer. In the process of the engine operations, the friction loss accounts for a large proportion of the energy loss. This part of energy is lost in the status of heat. In order to decrease the piston wear and friction loss, lubricating oil film is existed in the gaps of cylinder liners and pistons, which unavoidably leads to the combustion of lubricating oil in cylinders. The lubricating oil leakage is considered to be 0.2% of the fuel consumption [36, 37]. The chemical energy supplied by the fuel and lubricating oil is shown in Equation 2,

$$\dot{Q}_f = Q_{f\_LHV} \cdot \dot{m}_f + Q_{o\_LVH} \cdot \dot{m}_o \qquad (2)$$

Where, Q_{f_LHV} and Q_{o_LVH} are the low heating values of the fuel (46.04 MJ/kg) and lubricating oil (36.00 MJ/kg) respectively; $\dot{m}_f$ and $\dot{m}_o$ are fuel and lubricating oil consumption rates (kg/s), respectively.

The energy of the fresh air is calculated using Equation 3,

$$\dot{Q}_a = \Delta h_a \cdot \dot{m}_a \qquad (3)$$

Where, Δh_a is the enthalpy difference of the fresh air from the standard status (kJ/kg); $\dot{m}_a$ is the fresh air flow rate (kg/s). The enthalpy is referred from the data of National Institute of Standards and Technology (NIST), which takes the water and CO_2 in the air into consideration.

The exhaust energy is as following,

$$\dot{Q}_e = \Delta h_e \cdot \dot{m}_e \qquad (4)$$

Where, Δh_e is the enthalpy difference of the exhaust from the standard status (kJ/kg), the value is obtained from NIST database; $\dot{m}_e$ is the exhaust flow rate (kg/s).

Energy transferring to the coolant is calculated using Equation 5. Because the heat transferring to the lubricating oil is low, it is classified into the unaccounted heat loss $\dot{Q}_u$. Also, most of the heat of lubricating oil is transferred to the coolant.

$$\dot{Q}_c = \dot{m}_c \cdot C_c \cdot \Delta T_c \qquad (5)$$

Where, $\dot{m}_c$ is the coolant flow rate (kg/s); C_c is the heat capacity of the coolant (4.2 kJ·kg^{-1}·K^{-1}); ΔT_c is the temperature difference of the coolant that flows into and out of the engine block.

For a turbocharged diesel engine, the air temperature of the compressor outlet is high, which drops the air density significantly. Intercooler can effectively decrease the fresh air temperature. The heat transferring to the intercooler is calculated using Equation 6,

$$\dot{Q}_i = \dot{m}_a \cdot \Delta h_i \qquad (6)$$

Where, $\dot{m}_a$ is the fresh air flow rate (kg/s); Δh_i is enthalpy difference of fresh air from the standard status (kJ/kg). The value is based on NIST database.

Turbocharger couples exhaust and intake air systems, which makes energy flow more complicated. In addition, the efficiency of the compressor and turbine dominates the exhaust energy utilization efficiency. Equation 7 shows the efficiency calculation of the compressor,

$$\eta_c = (h_{2s} - h_1)/(h_2 - h_1) = (T_{2S} - T_1)/(T_2 - T_1) \qquad (7)$$

Where, h_1, h_2 and h_{2s} are air enthalpy of compressor inlet (kJ/kg), air enthalpy of compressor outlet (kJ/kg), and air enthalpy of compressor outlet (kJ/kg) if isentropic compression, respectively. T_1, T_2 and T_{2s} are compressor inlet temperature (K), compressor outlet temperature (K), and compressor outlet temperature (K) if isentropic compression, respectively. Equation 8 shows the T_{2s} calculation,

$$T_{2S} = (P_2 / P_1)^{\frac{\kappa_a}{\kappa_a - 1}} \cdot T_1 \qquad (8)$$

Where, P_1, P_2, T_1 and κ_a are compressor inlet pressure (Pa), compressor outlet pressure (Pa), compressor inlet temperature (K), and specific heat ratio of air (1.4), respectively.

Equation 9 shows the calculation of compressor power consumption,

$$P_c = (h_2 - h_1) \cdot \dot{m}_a \qquad (9)$$

The efficiency calculation of the turbine is shown in Equation 10,

$$\eta_T = (h_3 - h_4)/(h_3 - h_{ST}) = \Delta h_T / \Delta h_{ST} \qquad (10)$$

Where, h_3, h_4 and h_{ST} are exhaust enthalpy of turbine inlet (kJ/kg), exhaust enthalpy of turbine outlet (kJ/kg), and exhaust enthalpy of turbine outlet (kJ/kg) if isentropic expansion respectively. The calculation of h_{ST} is shown in Equation 11,

$$\Delta h_{ST} = \frac{\kappa_3}{\kappa_3 - 1} RT_3 \left[1 - \left(\frac{1}{\pi_T}\right)^{\frac{\kappa_3 - 1}{\kappa_3}} \right] \qquad (11)$$

Where, κ_3, T_3, R and π_T are exhaust specific heat ratio of at turbine inlet, exhaust temperature of turbine inlet (K), gas constant (8.314 J·mol^{-1}·K^{-1}), and expansion ratio of turbine respectively.

Equation 12 shows the power consumed by turbine,

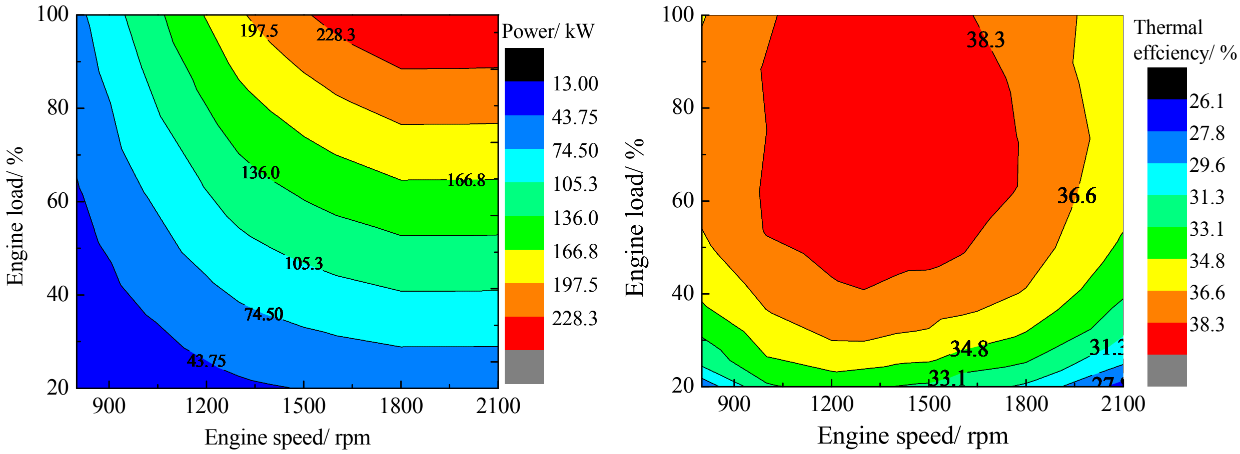
$$P_T = (h_3 - h_{ST}) \cdot (\dot{m}_a + \dot{m}_f + \dot{m}_o) / (\eta_{mT} \cdot \eta_{mC} \cdot \eta_{sh}) \qquad (12)$$

Where, η_{mT}, η_{mc} and η_{sh} are mechanical efficiency of turbine, compressor and shaft, respectively.

4. Results and discussion

4.1 Engine power output distributions

The target engine is used for a heavy duty truck, the engine power output and thermal efficiency at different engine operation conditions are shown in Figure 4. The thermal efficiency of the diesel engine was higher than 26% at all the researched operation conditions, and it was higher than 38% only when engine load was higher than 40%. However, internal combustion engines regularly operate at low speed and load conditions when vehicles run in urban driving cycles. Much energy is lost in the status of heat, mainly by the coolant and exhaust. There are huge potentials for the diesel engine to improve the energy utilization efficiency by thermal management, such as, recycling the heat of the coolant, intercooler and exhaust, and optimal coolant and lubricating oil temperature. Analysis of the energy flow in the diesel engine operation process provides the foundations of thermal management. Gharehghani et al. [34] investigated the energy flow of a turbocharged spark ignition engine, which considered the intercooler heat loss and leaked lubricating oil combustion, also, calculated the unaccounted heat loss. The unaccounted heat loss was mainly caused by the convection and radiation transfer. However, the heat loss by the coolant and intercooler was analyzed jointly although different energy grade levels. The energy distribution analysis of turbine and compressor were neglected, which had a huge effect on the energy flow in internal combustion engines.

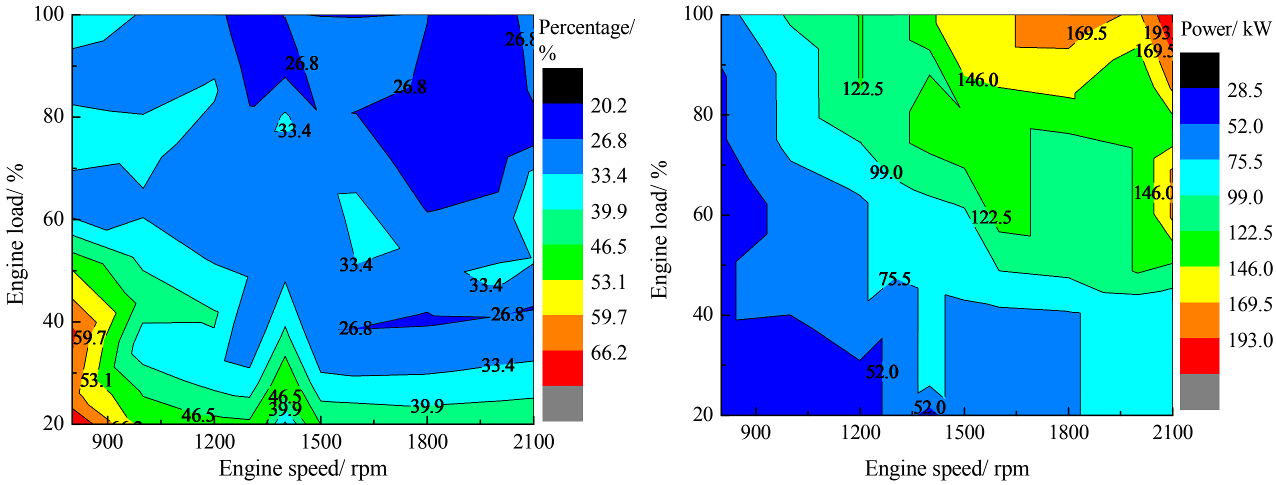


(a) Engine power output (b) Thermal efficiency

Figure 4 Engine power output and thermal efficiency distributions

4.2 Energy distributions of coolant system

In order to prevent the engine block overheating, cooling system is necessary to keep the engine block at an appropriate temperature level. Large percentage of heat is transferred from the coolant to the atmosphere; also, it greatly depended on the engine operation conditions. Due to the lubricating oil being cooled by the coolant, the coolant accounted for ~96% of the total cooling heat. As indicated by Jung et al. [38] that the heat loss caused by the coolant accounted for 70.1% of the total cooling loss, and it was 16.1% for the lubricating oil, in addition, 4.8% heat loss was caused by convection and radiation transfer. Figure 5 shows the coolant energy loss distributions at different engine operation conditions. The percentage of the energy calculated in this paper referred to the percentage of target energy in the total fuel chemical energy, except where noted. The share of the coolant heat loss was more than 50% for the engine operating at low power output conditions, and it was ~30% for high load and speed conditions. The maximum energy loss was ~193 kW when engine operated around the rated power condition. High coolant heat loss led to a low brake thermal efficiency that such large amounts of energy should be effectively recycled. The energy recycling devices should consider the high power output conditions, because large quantities of heat were lost although engine seldom operated around the rated power conditions. The design point of the traditional cooling system was the maximum heat loss condition, meanwhile, the coolant flow rate was proportional to the engine speed for the conventional coolant pump, which led to the redundant heat loss at part load conditions [39]. More heat loss partly caused lower thermal efficiency, also, excess coolant flow rate led to more energy consumption of coolant pump [28, 29, 40]. As can be seen, some unexpected contour lines existed (e.g. around 60% load and 1650 rpm), which was resulted from data interpolation error.



(a) Energy percentage (b) Energy quantity

Figure 5 Energy loss distributions by coolant

4.3 Energy distributions of turbocharger system

Coolant and exhaust energy dominate the energy loss of the diesel engines, the exhaust energy distributions before turbine are shown in Figure 6. The energy loss percentage increased with the engine speed; in addition, the engine load presented a smaller effect on the energy distributions than the engine speed. The energy percentage ranged from ~25% to ~38%, which was in the same level with the engine brake thermal efficiency. High exhaust temperature and flow rate caused a huge exhaust energy percentage at high speed conditions, which partly caused the low thermal efficiency.

Turbocharger as an effective method to recycle the exhaust energy was widely applied to the internal combustion engines [41-43]. Figure 7 shows the distributions of the energy consumed by the turbine. Being enslaved to the maximum cylinder pressure and thermal load, the waste-gate was in a large opening at high power output conditions, which caused less energy to be recycled by turbine. A noteworthy phenomenon was observed by comparing Figures 6 and 7 that the exhaust energy percentage was the lowest (~27%) around 800 rpm and 100% load conditions, where the percentage of the energy consumed by the turbine was at a general level (~5%). Higher brake thermal efficiency around this point (800 rpm and 100% load) partly led to the low percentage of exhaust energy before turbine, meanwhile, high exhaust temperature and pressure partly explained the high recycle efficiency (recycled energy over exhaust energy) of turbine (Figure 8).

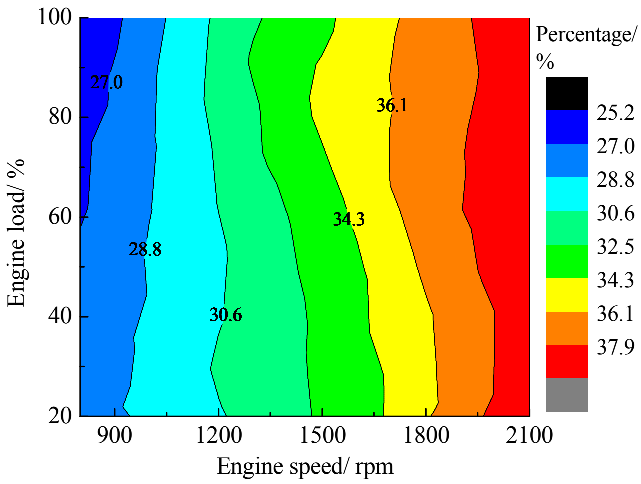


Figure 6 Exhaust energy distributions before turbine

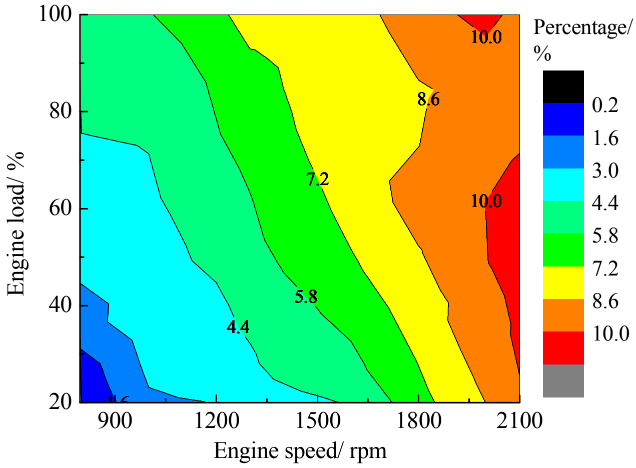


Figure 7 Distributions of energy consumed by turbine

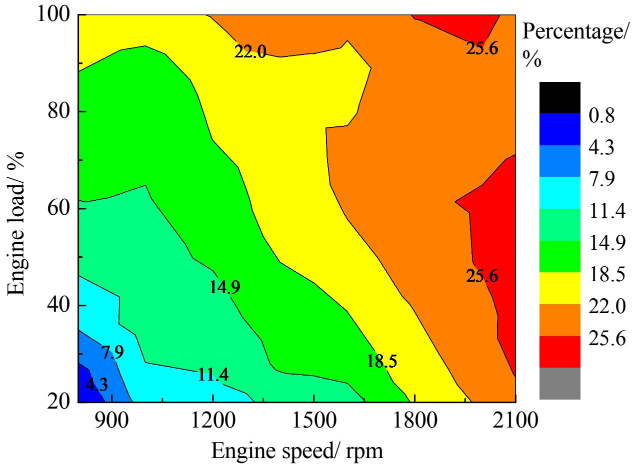


Figure 8 Distributions of energy recycled percentage

The exhaust energy percentage after turbine was more than 25% at majority of the engine operation conditions, as shown in Figure 9. The exhaust energy was ~30% for the high speed conditions. The exhaust energy was further recycled using a two-stage turbocharger which contributed to the engine downsize [44]. In reference [45], a two-stage turbocharger was used to recover the engine power at plateau or to obtain higher power density, with the results of more complicated energy flow in the system. Exhaust temperature decreased greatly after passing the turbine, which lowered the energy grade, however, the heat quantity was still ~25% in the fuel chemical energy. ORC systems [46-48] and CO_2 transcritical waste heat recovery systems [49, 50] were used to recycle the exhaust energy after turbine. After passing through the evaporator of the single stage Rankine cycle, the exhaust temperature was still high despite of a lower grade level of the heat energy. In order to further recycle the exhaust energy, the double stage and triple stage Rankine cycle were used [51], a maximum power of 517.27 kW was recycled from a 2928 kW diesel engine, whose exhaust temperature was ~470 °C. Although complicated equipments for multiple stage Rankine cycle, ~50% of the exhaust energy could be recycled. Estimated form the reference [51], a maximum recycled power of 50 kW could be achieved using the system for the diesel engine in the paper. It should be noted that high energy recycling efficiency was obtained in many researches, where after-treatment systems were neglected that the after-treatment should be positioned before ORC system. High temperature was need to light-off catalysts, and much heat was lost through catalysts and pipes, which would decrease the energy recycling efficiency. Meantime, the oxidations of hydrocarbon and carbon monoxide released much heat, however, the hydrolysis of ammonium hydroxide was a endothermic reaction. The research about the effect of ORC system and after-treatment layout on the energy recycle efficiency should be further performed.

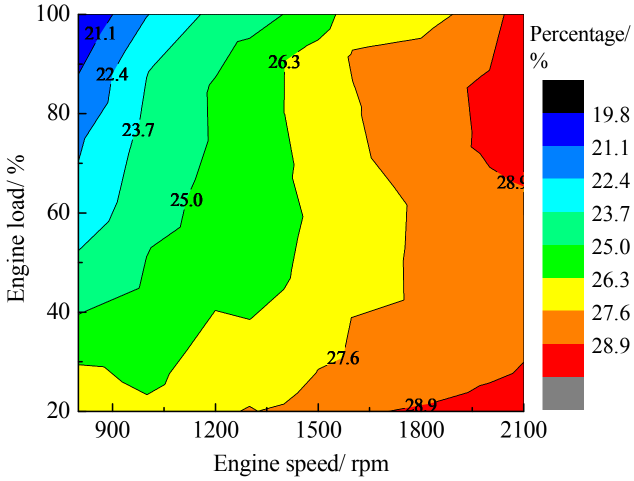


Figure 9 Exhaust energy distributions after turbine

In order to meet the requirements of the stringent emission regulations, diesel oxidation catalyst (DOC) and selective catalytic reduction (SCR) are used to decrease the exhaust emissions. High temperature is needed to light-off the catalysts to achieve high catalytic efficiency. The exhaust energy could be used to heat the catalysts when catalysts were in the inefficiency conditions. Figure 10 presents the exhaust temperature distributions after turbine, and 12 operation points in "13-mode test cycle". As can be seen, the temperatures of 3 low load operation points were ~220 °C where DOC and SCR were fully light-off. As indicated in references [52-54], PM ignition temperature was more than 450 °C for non-catalytic DPF. Temperature was much low to achieve DPF regeneration at part load conditions, the phenomenon was more serious at cold start and warm up conditions. Thermal management was adopted to increase the exhaust temperature [55], such as delaying the start of combustion [56], burners [57], heat storage materials [30] and electrical heated catalysts [58]. Kauranen et al. [59] adopted a latent heat accumulator to storage exhaust thermal energy which could be used to fast light-off catalyst at cold start conditions. This device could replace the extra heater and eliminate the fuel penalty. Exhaust energy recycling should be combined with after-treatment systems to achieve the optimal energy efficiency under the conditions of meeting emission regulations.

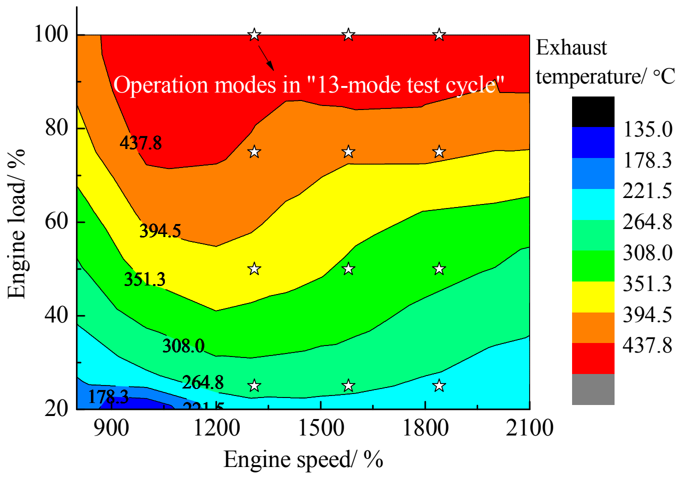


Figure 10 Distributions of exhaust temperature after turbine

Turbocharger couples intake air and exhaust systems, which make the energy flow cross-correlations in turbine and compressor. Energy consumption percentage distributions of compressor are shown in Figure 11, and the efficiency distributions are presented in Figure S1. Compressor efficiency was higher than 65% for majority of the operation points. Energy consumption by compressor ranged from ~2% to ~8%, and the distributions were dramatically different from turbine. The energy consumed by compressor increased generally with engine power when the engine load and speed were low. Waste-gate of turbine was in a large opening when engine power output was huge, being restricted by the cylinder peak pressure and thermal load, which decreased the energy consumption percentage.

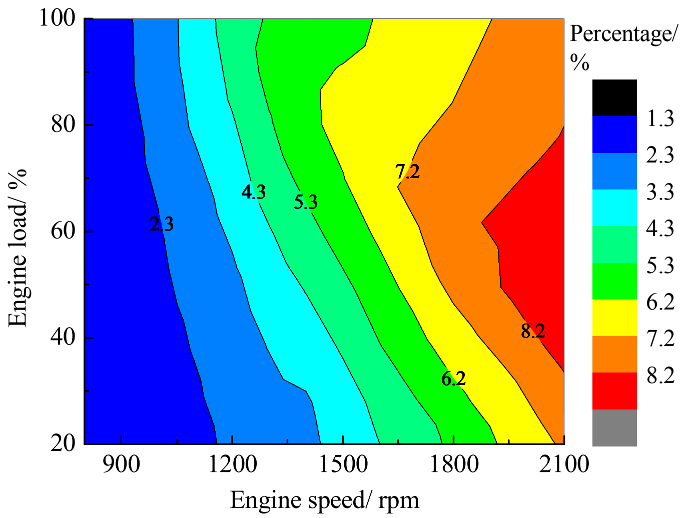
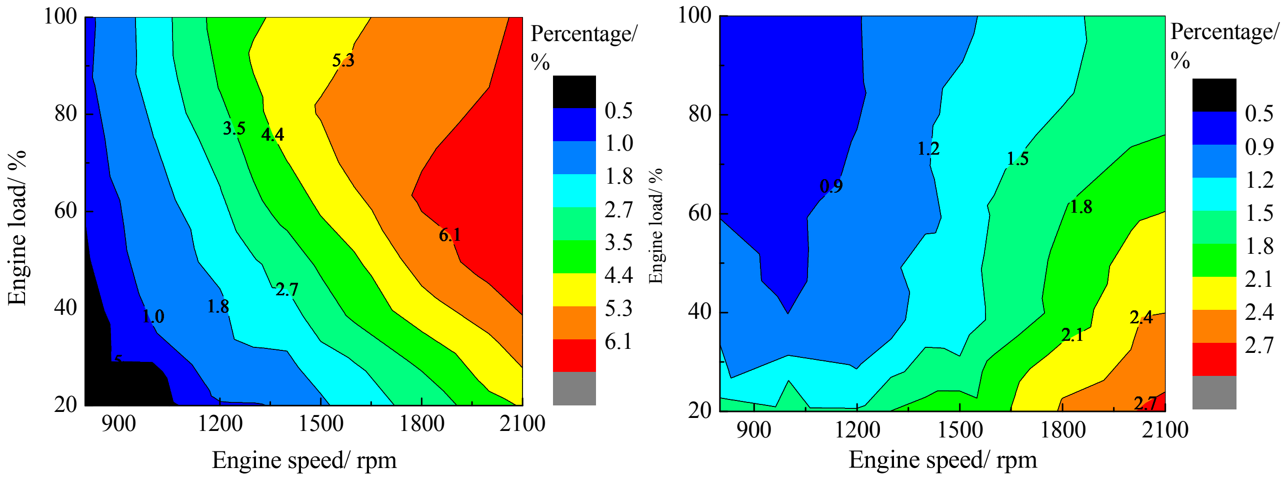


Figure 11 Energy consumption percentage distributions of the compressor

4.4 Energy distributions of intercooler system

In order to increase the intake air density, decrease engine thermal load and NO_x emissions, intercooler is necessary to decrease the intake air temperature. Much heat was transferred to the atmosphere through intercooler, as shown in Figure 12(a). The air flow rate, inlet and outlet temperatures of intercooler are shown in Figure S2~S4, respectively. The maximum percentage of the heat loss was more than 6.1% of the total fuel chemical energy. Pressure ratio of compressor was small at low engine speed and load conditions, where the heat loss by the intercooler was smaller than other conditions. Figure 12(b) shows the distributions of recycled energy flowing into the cylinders. Only 1%~3% of the total fuel chemical energy flowed into the cylinders eventually to increase the intake air flow rate. High engine speed and low load conditions showed the hugest percentage. Due to its low temperature of compressor outlet, this made it hard to recycle this part of energy. Shu et al. [24] used a pre-heater (improved intercooler in the diesel engine) to recycle the heat of intercooler, using CO_2-based transcritical Rankine cycle. The working fluid passed the pre-heater firstly to absorb the heat of the intercooler, then, flowed to the evaporator to recycle the exhaust thermal energy. This method showed an excellent performance of recycling intercooler and exhaust energy.



(a) Distributions of energy transferred by intercooler (b) Recycled energy flowing into cylinders

Figure 12 Energy distributions of intercooler system

4.5 Potentials of decreasing emissions and BSFC by optimizing coolant and oil temperature

For the conventional internal combustion engines, coolant pumps were actuated directly using shaft that the coolant flow rate was proportional to engine speed [28]. In addition, the designs of the coolant pumps were based on the maximum heat transfer conditions, with the results of engines' overcooling at the other conditions. Also, waxy thermostats were insensitive to the engine coolant temperature, which made the coolant temperature control with a low precision. Allen et al. [60] showed that the coolant pump in conventional engines produced much more coolant flow rate than required value that the phenomenon reached up to 95% the total operation time. The temperature of the coolant flowing out of the engine block is shown in Figure S5. Overcooling of the engine bock caused poor cylinder combustion and serious cylinder quenching effect, with the results of high BSFC and exhaust emissions. References [61-63] adopted electric pump and electric thermostat to control the coolant temperature to ensure the excellent thermal status of coolant. As indicated by Chanfreau et al. [29], it reduced 2%~5% fuel consumption, 10% tailpipe HC and 20% CO emissions when the coolant temperature was increased from 90 °C to 110 °C. The potentials of BSFC and emission reductions are shown in Figure 12 The optimal temperature of the coolant flowing out of the engine block was considered as 90 °C in this paper. The coolant temperature optimization could be achieved by electric pump or heat storage approaches [55]. Low engine speed and load conditions showed the maximum performance improvement that the BSFC decreased by ~3%, and emission reduction was ~10% by increasing the coolant temperature to 90 °C. The operation conditions around the rated power showed the smallest BSFC and emission reductions. Higher coolant temperature meant less heat transferring to the coolant from the engine block, which contributed to enhanced thermal efficiency. Higher coolant temperature also contributed to the air/fuel mixture formation, which caused better in-cylinder combustion and less quenching effect. In addition, the coolant pump could be downsized to further decrease the coolant pump energy consumption. Less heat transferred to the coolant could also decrease the power consumption of fan. Also, some unexpected contour lines existed were observed, which was resulted from data interpolation error.

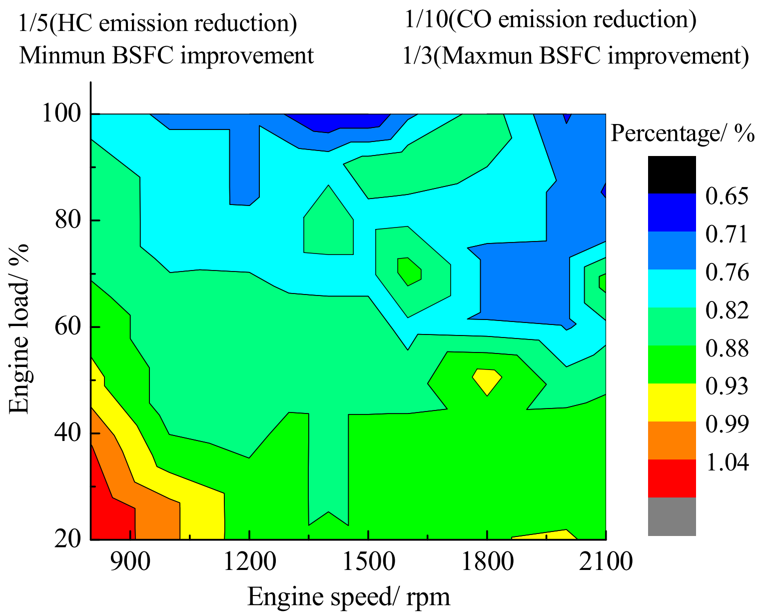


Figure 13 Potentials of BSFC and emission reductions by optimizing coolant temperature

Temperature of the lubricating oil was lower than the optimal value in one third of the trips [64], and the maximum friction losses in the warm up process were 2.5 times higher than the optimal temperature conditions [65, 66]. Similar to the electric coolant pumps, electric oil pumps could be used to decrease the energy loss and emissions, also, the energy consumption of oil pumps. The optimal temperature was considered as 110 °C in this paper, the estimated BSFC reduction after enhancing lubricating oil temperature is shown in Figure 13. Temperature distributions of the lubricating oil in different engine operation conditions are shown in Figure S6. The potential of maximum BSFC reduction by using the electric oil pump was ~ 1 g/(kW·h), where the engine ran at low speed and load conditions. The least BSFC improvement was ~ 0.4 g/(kW·h), where the engine operated at high load and speed conditions, in which the in-cylinder temperature was much high. The potentials for the hybrid electric vehicles (HEVs) improvement is huger, because regular start-stop conditions are common, which makes the engine more frequently operate at low coolant and oil temperature conditions. Intelligent control of the coolant pump and oil pump is necessary to control the coolant and oil temperature with a high precision.

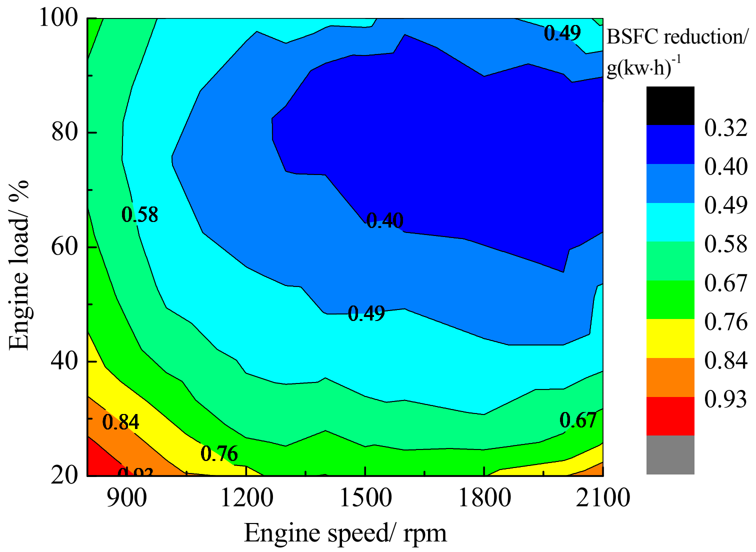
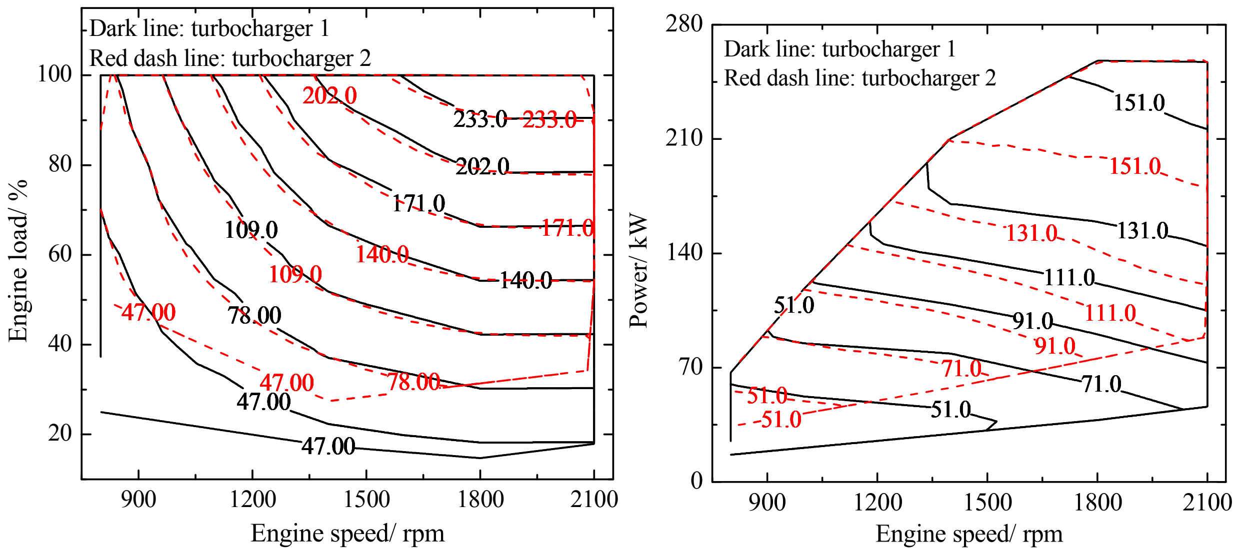


Figure 14 Potentials of BSFC reduction by increasing oil temperature

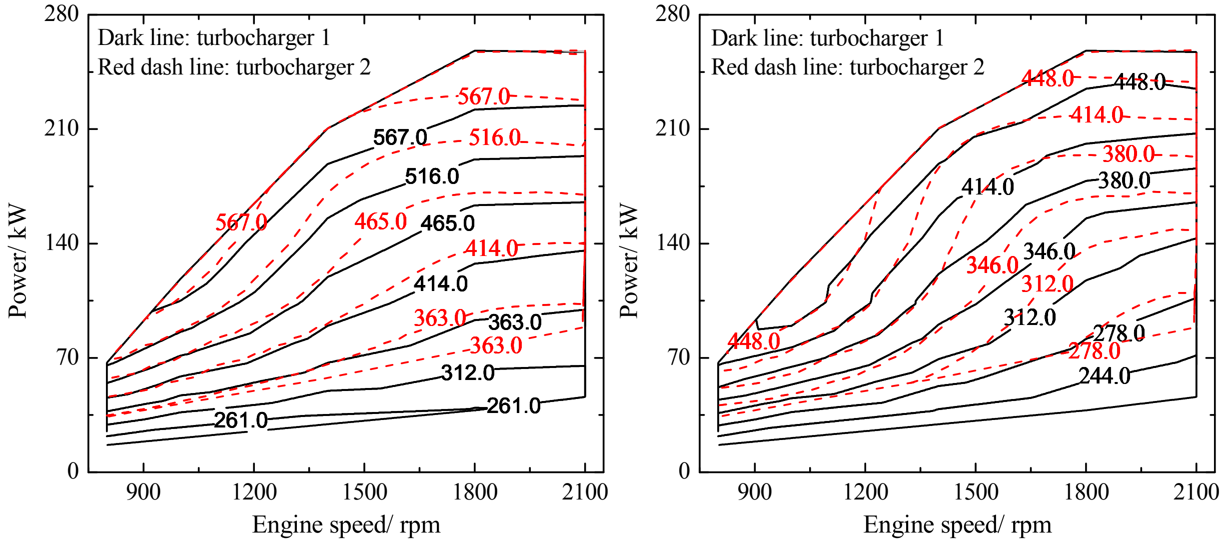
4.6 The effect of different turbocharger systems

In the process of engine operations, turbochargers have a huge effect on the energy flow, resulting from the coupling of the intake and exhaust systems. Higher power output of the turbine contributes to higher intake air density; however, it causes higher engine backpressure. Abedin et al. [35] reviewed the effect of turbocharger on the energy balance. The BSFC of the internal combustion engine decreased by ~5%, and with another ~5% improvement if the intercooler device was adopted. The heat loss by diesel exhaust decreased from ~35% to ~30% because of the turbocharger application, however, the coolant heat loss increased due to high thermal load caused by a higher power density.

Figure 15 shows the effect of different turbochargers on the engine performance. In the above research, the turbocharger 1 was used because only the power output and brake thermal efficiency were considered when choosing the turbocharger. However, the situation may change if all kinds of energy recovery approaches were conducted. As can be seen from Figure 15(a), the power output of the diesel engine increased slightly at high engine speed and low load conditions after turbocharger 2 was applied; however, it decreased at low and medium engine speed and load conditions, where the engine normally operated in real driving conditions. The outlet temperature of the compressor increased after adopting turbocharger 2 at majority of the operation conditions. Also, the temperature difference was huger at higher power output conditions. Higher compressor outlet temperature was caused by higher compression ratio, which also caused more heat loss by intercooler. The inlet and outlet temperatures of the turbine were lower for turbocharger 2 compared with turbocharger 1. It presented a small influence on turbine inlet and outlet temperatures at low power output conditions. The temperature differences of turbine outlet would cause different energy recovery efficiency, such as the two-stage turbochargers, ORC systems. In addition, the outlet temperature of turbine was closely related to after-treatment performance. In the enhanced after-treatment systems, effective heating measures were needed to fast light-off catalyst [66, 67]. The detailed investigations of the effect of different turbochargers on the energy flow in diesel engines will be further conducted.



(a) Engine power output (b) Temperature of compressor outlet



(c) Temperature of turbine inlet (d) Temperature of turbine outlet

Figure 15 Effect of turbochargers on the energy distributions

5. Conclusion

In order to decrease the diesel exhaust emissions and fuel consumptions using thermal management methods, energy flows in a turbocharged diesel engine were analyzed, including the energy distributions of the engine power output, coolant, lubricating oil, turbine, compressor and intercooler systems. The main conclusions are as the following:

(1) The brake thermal efficiency of the diesel engine was more than 33% at majority of the engine operation conditions, with the maximum value being more than 38%.

(2) The coolant energy ranged from 30 kW~190 kW, and the minimum percentage was more than 26% of the total fuel chemical energy, also, the maximum value was ~66%. The potentials of coolant energy recycling were promising, although the energy grade level was low, if excellent thermal management methods were used. Exhaust energy consumed by the turbine was in the range of 1.6%~10.4%, which changed the exhaust energy distributions; meanwhile, the changes were significant when the engine loads were smaller than 50%.

(3) The tendency of the contour lines of energy distributions were similar for compressor and intercooler, resulting from the fact that higher energy consumption by compressor would led to higher heat loss by intercooler. The recycled exhaust energy eventually flowing into the engine cylinders was in the range of 1%~3% due to much heat loss in the intercooler.

(4) The estimated of BSFC improvement was 0.65%~1.95% by optimizing coolant temperature, also, it was 0.31 g/(kW·h)~0.93 g/(kW·h) for lubricating oil. In addition, the HC and CO emissions

could be reduced by 3.25%~5.2% and 6.5%~10.4%, respectively when the coolant temperature was kept at 90 °C using electric coolant pump and electric thermostat.

It should be noted that the choosing of the turbocharger for a diesel engine is basically based on the power output and brake thermal efficiency. The conditions will change if the energy recycling and thermal management approaches (e.g. ORC) are used. The effect of different turbochargers on the energy flow in the diesel engine after the application of energy recycling approaches should be further analyzed. It may lead to the changes of the matching criteria for turbocharger with the engine.

Acknowledgement

The research described in this paper was supported in part by the EU-funded project optiTruck (grant agreement No 713788) which has the ultimate aim to develop and test a prototype 'global optimizer', capable of achieving a fuel reduction of at least 20% for 40 tonne trucks while still meeting relevant Euro VI emission standards.

Support information

This section provides the information about the efficiency distributions of compressor, air flow rate distributions, temperature distributions of intercooler inlet, temperature distributions of intercooler outlet, temperature distributions of coolant flowing out of engine block and Temperature distributions of lubricating oil.

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