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An experimental study on heat exchange effectiveness in the diesel engine EGR coolers

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Abstract

Both reducing nitrogen oxides (NOx) and particulate matter (PM) emissions from diesel engine and improving fuel consumption are important in meeting government regulations and society needs. Use of the Cooled Exhaust Gas Recirculation (EGR) system is one of the most effective techniques currently available for reducing NOx and PM emissions. However, the EGR system has a trade-off between NOx and PM emissions at high loads. In the present study, engine dynamometer experiments have been performed to investigate the heat exchange effectiveness of EGR coolers with shell & tube-type and stack-type. The results show that the heat transfer effectiveness of the stack-type EGR cooler is 25-50 % higher than that of the shell & tube type due to an increased surface area and a better mixing of the exhaust gas flow.

Keywords: Diesel engine; EGR cooler; Shell & tube-type; stack-type; Heat exchange effectiveness

1. Introduction

In recent years, stringent emissions legislation has been imposed worldwide on nitrogen oxide (NOx) and particulate matter (PM) emissions from diesel engines [1, 2]. Diesel engines are typically characterized by low fuel consumption and very low CO_2 emissions, and are predominantly used to power tractors and heavy trucks. They have also become increasingly attractive for smaller trucks and passenger cars due to their low fuel consumption. However, according to Kakoi et al.[3] and Park et al.[4], the high NOx emission from the diesel engine remains a major problem in the pollution aspect.

Combustion processes and engines will have to undergo further development to comply with future emission limits. In fact, according to Leet et al. [5] and He et al. [6], the limits imposed by the new emission legislation, such as EURO4, EURO5, EPA07, and EPA10, demand a high exhaust gas cooler performance. Thus, by year 2008 the EGR cooler for EURO5 vehicle must meet emission limits of 2.0 g/kWh of NOx and 0.025 g/kWh of PM.

McKinley [7] has demonstrated how a cooled exhaust gas recirculation (EGR) system can reduce emissions. Lim et al. [8] has studied the controlling attribution of EGR temperature and exhaust properties with the change of operation conditions. And both studies show that NOx is significantly reduced by using the EGR cooler, while more PM is produced due to a trade-off relation between NOx and PM. However, neither McKinley [7] nor Lim et al. [8] has dealt with thermal resistance on the effectiveness of EGR cooler.

Since the heat transfer rate on an EGR cooler is normally limited by its thermal resistance, a heat transfer performance test with various enhanced sur-

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faces is essential in developing an EGR cooler that will meet stringent future emission limits. Thus, in the present study, engine dynamometer experiments have been carried to investigate the heat exchange effectiveness for both shell & tube-type and stack-type EGR coolers.

2. Experimental set-up

2.1 Engine and dynamometer

The engine used in this study is a 1991cc 4-cylinder diesel engine that consists of 4 valves per cylinder, common rail fuel injection, turbocharger, EGR control valve, and EGR cooler. Table 1 shows the specifications of the test engine.

The engine is coupled to an eddy current type DC dynamometer (KEDM-130). Six K-type (Chromega-Alumega) thermocouples of 0.26 mm diameter are used to measure exhaust gas and coolant temperatures. All of thermocouples were calibrated against a platinum resistance thermometer (SDL-T23/30) as a temperature standard in a constant temperature water bath (NESLAB/RTE-221D) to an accuracy of ± 0.2 ℃. Temperatures, speeds, and loads are all collected by a digital data acquisition system (DEWE-5000).

2.2 EGR cooler

Fig. 1 shows a schematic diagram for the EGR cooler performance test. In the EGR cooler system, 20-30 % of the hot combustion gases from the diesel engine is recirculatd into the EGR cooler and significantly cooled down by the coolant in the EGR cooler. The inlet coolant and inlet exhaust gas temperatures in the EGR cooler are around 80-90 and 130-150 ^{o}C, respectively.

Table 1. Specifications for the test engine.

Description	Specification
Displacement (cc)	1991
Induction Type	Turbocharger
Bore x Stoke (mm)	83 x 92
Maximum Power (ps/rpm)	126 / 4,000
Maximum Torque (kg_f-m/rpm)	29.5 / 2,000
EGR System	Cooled
Compression Ratio	17.7

As a result of heat exchange between the coolant and the exhaust gas, the outlet exhaust gas temperature is reduced to around 100-120 ^{o}C. It is then mixed with fresh air before being injected into the engine cylinder. This process results in the reduction of combustion chamber temperature as well as NOx emission from the engine [9, 10]. The considerable reduction in NOx level required by the emission legislation is accompanied by an increase in the recirculated mass flow (PM) and a decrease in the exhaust gas temperature [11].

The shell & tube-type and stack-type EGR coolers have been experimentally tested. The shell & tube-type EGR cooler consists of twenty spiraled tubes having a diameter of 8 mm, a pitch of 6 mm and a length of 150mm. In the meantime, the stack-type cooler is the plate-fin heat exchanger and its overall core size is about 70x65 mm and a length of 200mm. Both coolers were thoroughly checked for any leakage before the brazing and turned out to be in an excellent condition after the brazing. Fig. 2 shows the front-view photographs of the two EGR coolers.

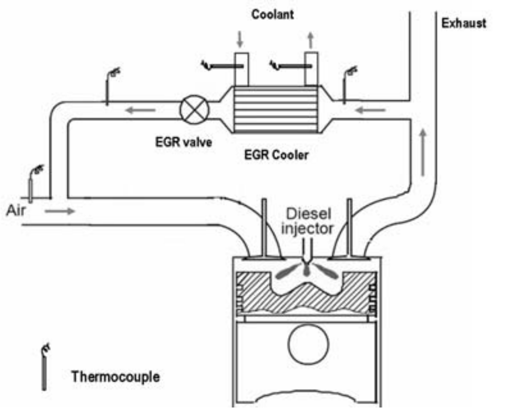
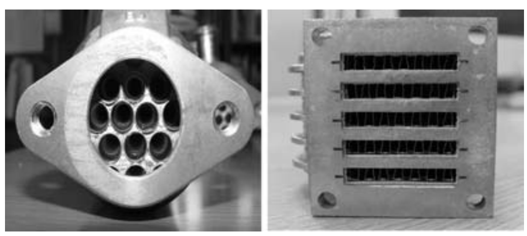


Fig. 1. Schematic diagram for the EGR cooler performance test (Kim and Lee, 2005).



Shell & Tube Type Stack Type

Fig. 2. Front-view photographs of the two EGR coolers.

3. Experimental procedure

An automatic speed regulator (ASR) driving mode was set in the engine dynamometer test. ASR is the test mode under which engine load varies from 0 to 80 %, while the engine speed was set at the same speed ranging from 1100 to 2000 rpm. The EGR cooler performance tests were conducted with the coolant inlet temperature held at 80-90 ^{o}C, the inlet gas temperature at 130-150 ^{o}C, EGR ratio of 20-30%, and constant coolant flow rate. The performance test mode procedure is illustrated in Fig. 3.

First, the engine was idled for 2 hours to obtain steady-state conditions. Then in Step 1, engine loads of 0, 20, 40, 60, and 80% were applied and maintained for 2 hours each at an engine speed of 1200 rpm. In Step 2, 0% engine load was maintained for 50 hours at an engine speed of 1100 rpm.

The engine speed was then increased to 1200 rpm in Step 3 and the load was increased first to 20% and then to 80% for 30 minutes. Then a 75% engine load was maintained for 5 hours at an engine speed of 2000 rpm in Step 4. Finally, the Step 1 loading conditions were repeated while maintaining the engine speed at 2000 rpm. The total time of the experiment was 78 hours, including the 2 hours of idling before Step 1.

4. Data reduction

In the present study, the effectiveness-NTU method is used to determine the EGR cooler effectiveness. It offers some advantages for analysis of problems in which a comparison between various types of heat exchangers has to be made in order to select the best-suited type for a specific heat transfer objective.

The actual heat transfer rate for the EGR cooler can be expressed as the arithmetic mean of the coolant side and gas side heat transfer rates:

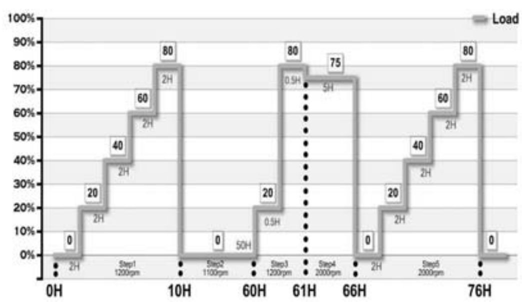


Fig. 3. Performance test mode procedure.

$$Q = \left(Q_c + Q_g\right)/2 \tag{1}$$

Where Q_c and Q_g are the coolant side and gas side heat transfer rates, respectively.

$$Q_c = \dot{m}_c C_{pc}\left(T_{co} - T_{ci}\right),\ Q_g = \dot{m}_g C_{pg}\left(T_{gi} - T_{go}\right) \tag{2}$$

Where T_{ci} ,T_{co} and T_{gi} ,T_{go} are the coolant inlet and outlet temperatures, and the exhaust gas inlet and outlet temperatures, respectively. It is assumed that heat losses in the EGR cooler are small so that $Q_c \approx Q_g$

Now, the EGR cooler effectiveness (ε) can be determined by following the relation suggested by Wang and Webb [12]:

$$\varepsilon = 1 - \exp\frac{NTU^{0.22}}{C^*\left[\exp\left(-C^* NTU^{0.78} - 1\right)\right]},\ NTU = \frac{UA}{C_{\min}} \tag{3}$$

$$\varepsilon = Q/Q_{\max},$$
$$Q_{\max} = (\dot{m}c)_{\min}\left(T_{gi} - T_{ci}\right) = C_{\min}\left(T_{gi} - T_{ci}\right),$$
$$C^* = \frac{C_{\min}}{C_{\max}} = \frac{\dot{m}_g C_{pg}}{\dot{m}_c C_{pc}} \tag{4}$$

Where, the effectiveness, ε **,** becomes the ratio of the actual heat transfer rate for the EGR cooler(Q **)** to the maximum possible heat transfer rate ($Q_{\max}$ **)**, NTU is called the number of transfer units which is indicative of the size of the EGR cooler. U is the overall heat transfer coefficient of the cooler and A is the surface area for the cooler.

$C_{\min}$ and $C_{\max}$ are the minimum and maximum capacity rates, respectively, and their ratio $C^* = C_{\min}/C_{\max}$ is equal to C_c/C_g or C_g/C_c depending on the relative magnitudes of the hot gas and cold coolant heat capacity rates.

5. Discussion of results

Fig. 4 shows variations of the exhaust gas outlet temperature with time at two different types of EGR cooler. In general, gas outlet temperatures for the stack-type cooler are 15-35 ^{o}C lower than those for the shell & tube-type because the finned surface area in contact with the coolant is larger for the stack-type and the exhaust gas flow is better mixed, resulting in the heat transfer enhancement.

It should be observed from Fig. 4 that between the end of Step 2 (60 hrs) and the beginning of step 5 (70 hrs), the exhaust gas outlet temperature at the shell & tube-type is sharply increased at relatively high engine loads. On the other hand, for the stack-type it is steadily decreased during the same time period. The sharp increase in the gas outlet temperature at the shell & tube type cooler may have been caused by PM adhering to the tube inner surface. Fig. 5 shows variations of the temperature difference between the exhaust gas inlet and outlet at two EGR cooler types. Again, from between 60 hrs and 70 hrs, sudden changes (but in the direction opposite to Fig. 4) in the temperature difference at both type coolers are shown from Fig. 5. This behavior is consistent with the outlet gas temperature observed in Fig. 4.

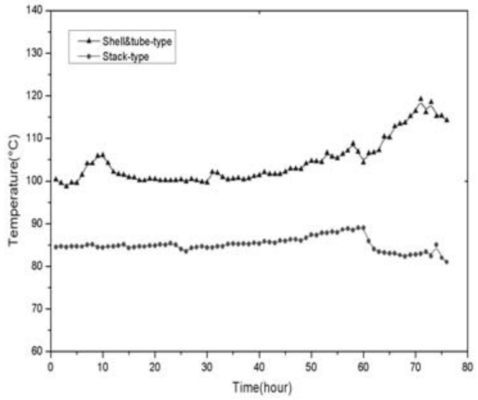


Fig. 4. Variations of the exhaust gas outlet temperature with time for two EGR cooler types.

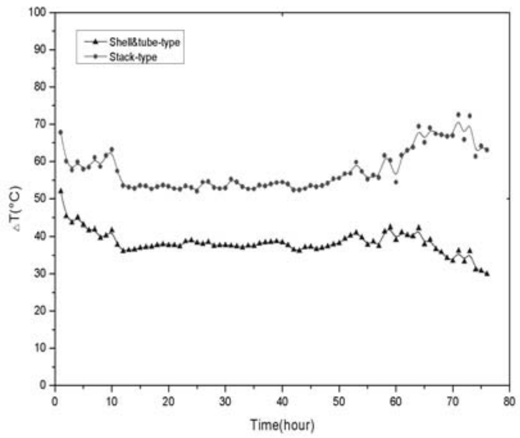


Fig. 5. Variations of the temperature difference with time between the exhaust gas inlet and outlet for two EGR cooler types.

Fig. 6 shows variations of the heat exchanger effectiveness with time at two type coolers. When PM adhered to the cooler surface, drop-offs occurred in the engine speed, load, and vibration due to a self-purification process. It turns out that the stack-type cooler has 25-50 % higher effectiveness than the shell & tube type. This has been confirmed by both the exhaust gas outlet temperature in Fig. 4 and the difference between the inlet and outlet exhaust gas temperature in Fig. 5.

It can be seen from Fig. 6 that the effectiveness of the shell & tube-type appears to somewhat sharply drop during Step 1(10 hrs) as the engine load increases from 0 to 80%, is stable during most of Step 2, and again sharply drops between the end of Step 2 (60 hrs) and the end of step 5 (76 hrs) at high engine loads mainly due to the fouling process caused by PM. However, the effectiveness of the stack-type remains nearly the same up to the end of Step 2. But, at the beginning of step 3 the effectiveness for the stack-type becomes higher than 100 %. This unusually high effectiveness may be attributed to that fact that the exhaust gas outlet temperature is somewhat lower than the dew point, resulting in the condensation of the exhaust gas, which in turn causes a high heat transfer enhancement and pressure drop in the exhaust gas.

It is worth noting that as the low temperature and condensing exhaust gas recirculates in the diesel engine, more PM likely adheres to the tube inner surface at high loads, causing a sharp increase in the gas outlet temperature and a significant reduction in the effectiveness for the shell & tube type cooler as shown

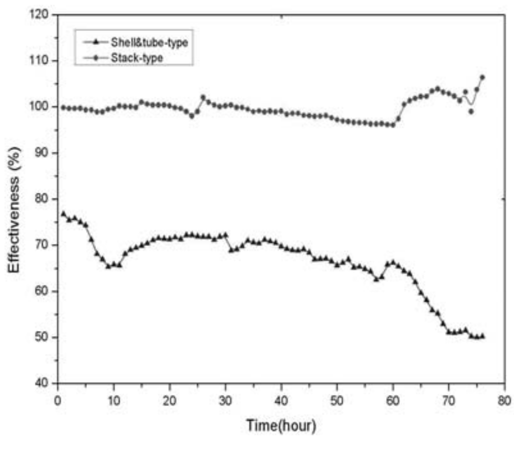


Fig. 6. Variations of the heat exchanger effectiveness with time for two EGR cooler types.

in Figs. 4 and 6, respectively. However, the low exhaust gas temperature tends to produce less NO_x, which is favorable from the environmental point of view. Thus, there is a trade-off between NOx and PM emissions at high loads.

The present experimental results demonstrate that the EGR cooler effectiveness can be improved by modifying its internal shape and configuration. We learned that the stack-type EGR cooler shows a much higher heat transfer performance than the shell & tube-type due to an increased surface area as well as a better mixing of the gas flow. However, it has limits in the heat transfer enhancement due to size, manufacturing, and cost. It must be remembered that more PM is produced at the lower exhaust gas temperature than at the high temperature. Thus, the EGR system has a trade-off between NOx and PM emission. Further studies with this trade-off and limits in consideration can improve the EGR cooler design and meet the stringent emissions legislation imposed on the future diesel engine.

6. Conclusions

Engine tests for two types of the EGR cooler were performed. The following conclusions are drawn.

In general, gas outlet temperatures for the stack-type cooler are 15-35 ^oC lower than those for the shell & tube-type, Thus, the stack-type cooler has 25-50 % higher effectiveness than the shell & tube type, mainly due to an increased surface area and a better mixing of the exhaust gas flow.

The exhaust gas outlet temperature at the shell & tube-type is sharply increased between the end of Step 2 and the beginning of Step 5 at high engine loads. This sharp increase in the gas outlet temperature may have been caused by PM adhering to the tube inner surface.

A condensation of the exhaust gas occurs at the stack-type cooler outlet, which causes a pressure drop in the exhaust gas, resulting in the further heat transfer enhancement.

More PM is produced at the lower exhaust gas temperature than at the high temperature. Thus, the EGR system has a trade-off between NOx and PM emission. Further studies with this trade-off in consideration can improve the EGR cooler design and meet the stringent emissions legislation imposed on the future diesel engine.

Acknowledgments

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Nomenclature

A	:	Area (m^2)
C*	:	Capacity ratio
C_p	:	Specific heat at constant pressure (J/kg K)
T_c	:	Coolant temperature (^oC)
T_g	:	Gas temperature (^oC)
NTU	:	Number of transfer units
$\dot{m}$	:	Mass flow rate (kg/s)
Q	:	Actual heat transfer rate (W)
Q_c	:	Coolant side heat transfer rate (W)
Q_g	:	Gas side heat transfer rate (W)
ε	:	Effectiveness

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