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DEPOSITION OF NANO-SIZED SOOT PARTICLES IN VARIOUS EGR COOLERS UNDER THERMOPHORETIC AND ISOTHERMAL CONDITIONS

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ABSTRACT

The present study aims at investigating the performance of various types of EGR coolers i.e. smooth tube, corrugated tube and plate-fin, when subjected to particulate fouling by soot particles. Experiments were carried out for different temperature gradients of 170 and 320°C (thermophoretic) and 0°C (isothermal). Soot particles with an average diameter of 130 nm were produced by a soot generator. Experimental results showed that generally soot deposition under isothermal conditions is negligible compared to thermophoresis for any given cooler geometry, but is not universal. It may become appreciable when complex coolers with extended surfaces, i.e. plate-fin type, are used due to impaction and settlement of soot particles onto the extended surfaces, which act as barrier to the flow. Contrariwise under thermophoretic conditions, the plate-fin cooler performed best followed by the corrugated tube and smooth tube cooler. Coolers with larger heat transfer surface area are also found to be less sensitive to the loss in effectiveness, but show a higher pressure drop.

1. INTRODUCTION

The widely used approach to reduce NO_x emissions in diesel engines is to re-circulate part of the exhaust gas to the engine intake. This is usually done via a heat exchanger known as an exhaust gas recirculation (EGR) cooler. A further reduction of nitrogen oxides requires higher ratios of EGR and lower gas temperatures. Kowada et al. (2006) tested a diesel engine with a base EGR cooler and found that with better charge cooling, the engine produces significantly less NO_x with no loss in specific fuel consumption. In turn, they found that the use of EGR may result in increased production of particulate matter i.e. mainly soot.

EGR coolers are subject to severe fouling which may reduce their effectiveness and increase pressure drop. The deposit layer is usually a mixture of mainly particulate matter i.e. soot particles and sticky heavy hydrocarbons. The geometry of the EGR coolers also can considerably influence its performance as they have to meet several technical requirements such as minimal deposition, least pressure drop and maximum effectiveness. In reality, nevertheless, it is a challenge to simultaneously maintain all these requirements.

Kim et al. (2008a) investigated the thermal effectiveness of various EGR coolers experimentally. Plate & fin heat exchangers turned out to be more efficient than shell & tube heat exchangers. Bravo et al. (2007) also compared shell & tube heat exchangers to plate-fin type heat exchangers. For the two given heat exchangers with similar thermal effectiveness under clean conditions the loss in effectiveness was higher for the shell & tube heat exchangers. They concluded that heat exchangers with higher heat transfer surface area are less sensitive to the impact of fouling on effectiveness. However, there was no information about the influence on pressure drop. In addition they compared similar shell & tube heat exchangers with various tube lengths and observed no variation in the fouling resistance. Nevertheless the loss in effectiveness was significantly different with 40% reduction for a tube length of 110 mm compared to 26% for tube length of 200 mm.

Kim et al. (2008b) also showed that the thermal effectiveness of shell & tube EGR coolers can be optimized by modifying its internal shape. Three coolers were examined: plain tubes (6 mm OD), spiral type-1 (6 mm OD) and spiral type-2 (8 mm OD). The latter consisted of only 20 tubes, while the first two consisted of 31 tubes. Furthermore corrugation of spiral type-2 differed in pitch and depth to that of spiral type-1. All tubes were made of stainless steel with a thickness of 0.5 mm. Due to fouling, severe clogging occurred in the plain type and the spiral type-1 coolers. The spiral type-2, nevertheless, remained unclogged and showed the best heat transfer performance among the investigated coolers. The effectiveness of the spiral type coolers decreased more rapidly as time elapsed. On the contrary, Castaño et al. (2007) found that the loss in effectiveness is lower for corrugated tubes compared to plain tubes of the same size. The corrugation causes a secondary flow, which breaks the boundary layer and consequently the flow velocity close to the tube wall is increased. Depth, pitch and corrugation angle are important parameters to optimize the cooler performance and to avoid sensitivity to fouling.

Ismail et al. (2004) characterized the fouling layer for EGR cooling devices by using a digital neutron radiography imaging technique. For a single and a three-tube inlet it was found that the thickness of the soot layer is highest at the entrance region in the first 25 mm or 4 tube diameters, where the flow develops. For turbulent flow the entrance region with a thicker deposition layer is smaller than for laminar flow. Fouling in the entrance region depends on how the flow enters the tubes and on the entrance geometry.

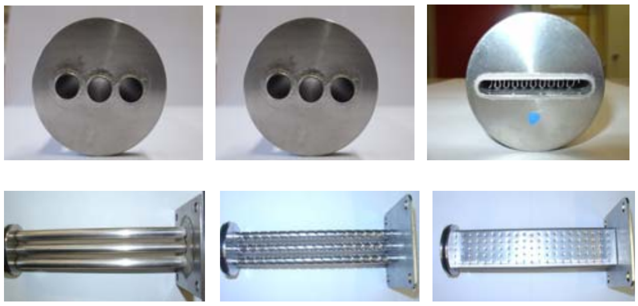
The brief review of previous studies presented above reveals that more investigations are required to characterize various aspects of cooler geometries that would mostly influence fouling. Accordingly, the present study endeavours to investigate rigorously the performance of various cooler geometries under isothermal and thermophoretic conditions when subjected to particulate fouling of soot particles. Experimental results under isothermal conditions are particularly rare, but imperative as diesel engines may get close to such conditions at low load or idle. Soot particles with an average diameter of 130 nm were the only foulant in this investigation. Experiments were carried out under highly controlled production of soot particles of similar size range as typically measured in diesel engine exhaust (Kittelson et al., 2006; Lapuerta et al., 2007; Wentzel et al., 2003). Visual observations also helped to determine the structure of deposit layer at the inlet and outlet of the coolers.

2. EXPERIMENTAL SET-UP AND PROCEDURE

The test rig and experimental procedures are fully described by Abd-Elhady et al. (2011) and are only presented here briefly. The rig consists of a soot generator which produces the exhaust gas by burning a rich mixture of ethylene and accompanied air. The soot generator delivers soot particles in the range of 10 to 300 nm at an average diameter of 130 nm with a constant mass flow of about 2 g/h. By injecting additional air, which is controlled by a flow controller, it is possible to vary the total mass flow. This allows investigation of fouling in EGR coolers at accelerated rate in relatively short period of time in the lab. In real diesel engines, though, the concentration of soot increases for higher velocities of exhaust.

The inlet temperature of the exhaust gas is controlled by a tube furnace, which can achieve inlet gas temperatures up to 600°C. Pressure and temperature are continuously measured by pressure transducers and K-type thermocouples respectively. The coolant of the EGR heat exchanger, here water, is supplied and controlled by a thermostat and its temperature can be varied from 25 to 95°C. The composition of the exhaust gas is analyzed by FT-IR spectroscopy and the mass concentration of particulate matter is measured with a smoke-meter using the filter paper method.

The examined EGR cooler is a stainless steel shell & tube heat exchanger. The hot exhaust gas passes through the tubes, while counter-flow cooling water passes through the shell. Three different tube geometries were used including: three smooth tubes, three corrugated tubes and an integrated plate-fin tube with an oval inlet (see Fig. 1a, 1b and 1c). The corrugated tubes have an outer diameter of 10 mm and thickness of 0.4 mm, while 12 mm and 1 mm are the outer diameter and the thickness of the smooth tubes, respectively. The plate-fin cooler has an oval inlet with a width of 44 mm and a height of 6 mm. The total cross-sectional area of the tubular coolers is 212 mm^2 and 215 mm^2 for the plate-fin cooler.



(a) (b) (c)

Fig. 1 Pictures of a) Smooth tube, b) Corrugated tube and c) plate-fin coolers.

The use of various structures has some implications on their heat transfer performance that are worthwhile to mention. For extended surfaces such as plate-fin coolers, as per following equation, larger heat transfer surface area results in lower fouling resistance compared to plain cooler. Instead for enhanced surfaces such as corrugated coolers, the heat transfer coefficient increases due to secondary flow generated by the grooves though the surface area is similar to that of plain tube.

$$\dot{Q} = U \cdot A \cdot LMDT = m_g C_p (T_{G,in} - T_{G,out}) \quad (1)$$

The thermal effectiveness "ε" of the EGR cooler can be defined as the ratio between the actual heat transfer to the theoretical maximum heat transfer as follows:

$$\varepsilon = \frac{T_{G,in} - T_{G,out}}{T_{G,in} - T_{C,in}} \quad (2)$$

where $T_{G,in}$ and $T_{G,out}$ are the inlet and outlet temperatures of the exhaust gas respectively, and $T_{C,in}$ is the coolant inlet temperature. The velocity "v" of exhaust gas which passes through the EGR cooler is determined by measuring mass flow rate m_g of the flowing gas and the cross-sectional area A_c of the cooler tubes as follows,

$$v = \frac{m_g}{\rho A_{cr}} \quad (3)$$

where ρ denotes the density of air at the average temperature of inlet and outlet. The gas velocities reported in this paper are the gas velocities at the inlet of the EGR cooler. Fouling occurs as soon as the gas-particle mixture passes through the cooler. The performance of the EGR cooler during operation is monitored through the respective inlet and exit gas/coolant temperatures, which are measured using calibrated thermocouples with an accuracy of ±0.1°C. The gas pressure is measured before and after the EGR cooler with an accuracy of ±0.4%. The overall thermal resistance, R_{th}, across the EGR cooler is calculated as:

$$R_{th}(t) = \frac{A_o \times LMTD}{\dot{Q}} \tag{4}$$

where $\dot{Q}$ is the rate of heat transfer across the EGR cooler, and is calculated from the measured temperatures of inlet and outlet of hot exhaust gas in the cooler at the corresponding specific heat capacities as shown in Eq. (1). A_o is the overall tube bundle heat transfer surface area of the gas-cooler section considered, and LMTD is the logarithmic mean temperature difference. LMTD is calculated from the inlet and outlet temperatures on the coolant/gas side of the EGR-cooler. The thermal resistance of the developed fouling layer, R_f, is then calculated at any time from:

$$R_f = R_{th}(t) - R_{th}(c) \tag{5}$$

where $R_{th}c$ is the overall thermal resistance under clean conditions.

Table 1 presents the matrix of operating conditions, which were used for various cooler geometries in this study. The total mass flow of exhaust gas is adjusted according to the desired initial gas velocity at the cooler inlet. The gas velocity is calculated based on the air density calculated at an average temperature of the exhaust gas inlet and the coolant temperature. During the test runs gas analysis is done in time intervals of one hour. The experiments are executed for about 4-6 hours. If the pressure drop across the cooler is close to 18 Pa, the experiment has to be terminated as the aluminium disc, which is installed above the soot collector as safety measure, may burst. Nevertheless, severe fouling has been observed even within the first 5 hours.

Table 1 Matrix of operating conditions.

Cooler type	($T_{G,in}$-$T_{C,in}$) (°C)	Gas Velocity (m/s)				
Smooth	320			30	80	
	170			30		
	0	10		30		
Corrugated	320			30	70	120
	170			30		
	0	10	20	30		
Plate-fin	320			30		
	170			30		
	0	10	20	30		

3. EXPERIMENTAL RESULTS AND DISCUSSION

3.1 Effect of gas velocity

Abd-Elhady et al. (2011) showed for micron and nano-sized particles, that operating above a critical gas velocity can substantially prevent particulate fouling in EGR coolers. The particles are removed from the surface due to exerted shear force on the surface. The critical flow velocity, which is necessary to prevent fouling is inversely proportional to the particle size. The critical gas velocity was calculated using the average particle diameter of 130 nm and was found to be about 67 m/s. Accordingly, several experiments were carried out for flow velocities below, at and above the critical gas velocity, 30, 70 and 120 m/s, respectively.

The cooler with corrugated tubes was operated at an inlet gas temperature of 400°C and an inlet coolant temperature of 80°C. The inlet and outlet temperatures of the coolant varied only marginally with a maximum increase of 2°C. The influence of gas velocity on fouling resistance is shown in Fig. 2. As it can be seen from this figure, an increase in the flow velocity causes a significant reduction in fouling resistance. For instance, after four hours of operation the fouling resistance for 30 m/s initial gas velocity is 0.008 (m^2K)/W, which is already four and eight times higher than for 70 and 120 m/s, respectively. In addition, a somewhat asymptotic fouling behaviour was observed for 70 and 120 m/s gas velocities, while in case of 30 m/s gas velocity, a more or less linear fouling curve was observed. It should be pointed out that due to the small particle diameters used in this study the critical gas velocity is very high and practically hard to achieve. Nonetheless in this work, it was possible to reach the high flow velocities of 70 and 120 m/s by blocking one or two tubes of the cooler inlet. Similar trends were also found for the smooth tube and plate-fin coolers but are not shown here for reasons of brevity.

Another parameter that may influence deposition is the residence time of precursors in the cooler as indicated by Abd-Elhady et al. (2011). Shorter residence times of the soot particles, which can occur at higher gas velocities, may not allow particles to deposit on the surface. Nevertheless further experimental studies are required before this hypothesis can be established.

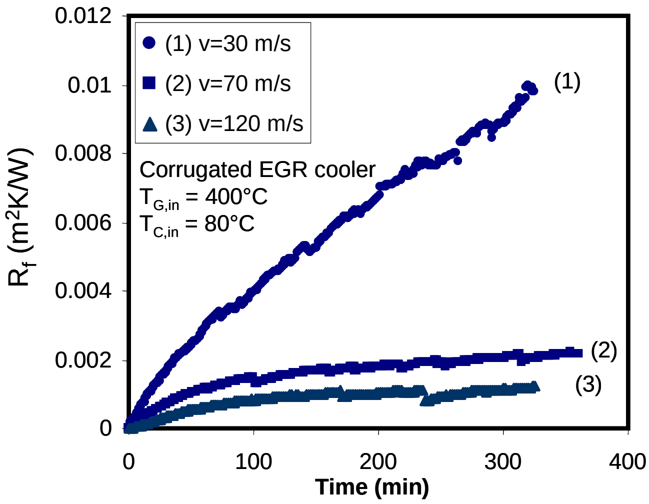


Fig. 2 Effect of velocity on fouling resistance for the corrugated cooler.

Table 2 presents results for fouling resistance, loss in effectiveness and pressure drop after five hours of operation. It shows the increase in fouling resistance and the loss in effectiveness decreases with increasing flow velocity. In contrast, the pressure drop increased considerably for higher flow velocities. It should also be noted that though both metrics of effectiveness and fouling resistance decrease with velocity but not in the same order of magnitude. This is because the effectiveness, as defined in Eq. (2) is mainly a macroscopic metric and gas outlet temperature is the only parameter that changes with time. Nevertheless fouling resistance can be considered as a more accurate measure that takes into account also the changing physical properties of the foulant here the exhaust gas.

The results presented in this table underline the importance of gas velocity in reducing particulate fouling in EGR coolers. However, due to the small size of the soot particles in diesel exhaust, the critical gas velocity is very high, and is not practically achievable. Furthermore increase in gas velocity through the cooler leads to increased pressure drop and pumping work for the engine. Even with gas velocity of 120 m/s the loss in effectiveness is still very high with about 40% after five hours.

Table 2 Performance of corrugated cooler after 5 hrs of operation for conditions stated in Fig. 2.

Velocity (m/s)	R_f (m^2K/W)	Loss in effectiveness $[(\varepsilon_c-\varepsilon)/\varepsilon_c]\times 100$	$\Delta p-\Delta p_c$ (kPa)
30	0.0088	57.83	0.9
70	0.002	40.4	1.2
120	0.0011	38.52	4.6

Increased gas velocity through the cooler might thus help in mitigating fouling due to the increased removal of particles from the deposit layer due to fluid shear force, but will have practical limitations due to increased pressure drop across the cooler and associated engine pumping work.

3.2 Effect of temperature

3.2.1 Isothermal tests

Isothermal conditions, where the temperature gradient between the exhaust gas at cooler inlet and coolant is close to zero or marginal are relevant for engine operation at low load and low exhaust gas inlet temperatures. Under these conditions the thermophoretic velocity is significantly lower. Accordingly, numerous experiments were conducted to discern how different coolers behave when investigated under isothermal conditions. However, due to the absence of a temperature gradient, it was not possible to determine a fouling resistance similar to the runs conducted under thermophoretic conditions. Instead a normalized pressure drop across the cooler was measured to characterize fouling in the absence of any temperature gradient:

$$\Delta p_{norm} = \frac{\Delta p_f}{\Delta p_c} \qquad (6)$$

where Δp_f and Δp_c denote pressure drop across the cooler under fouling and clean conditions.

Fig. 3 compares experimental results for the smooth tube cooler for two velocities of 10 and 30 m/s. Both the coolant and exhaust gas were maintained at 90°C for all the experiments. The results show that overall there is not an appreciable change in the normalized pressure drop for the smooth tube cooler at different velocities. For both gas velocities of 10 and 30 m/s the normalized pressure drop increases slightly and reaches an asymptotic value of around 1.2 after about 2 hours of operation.

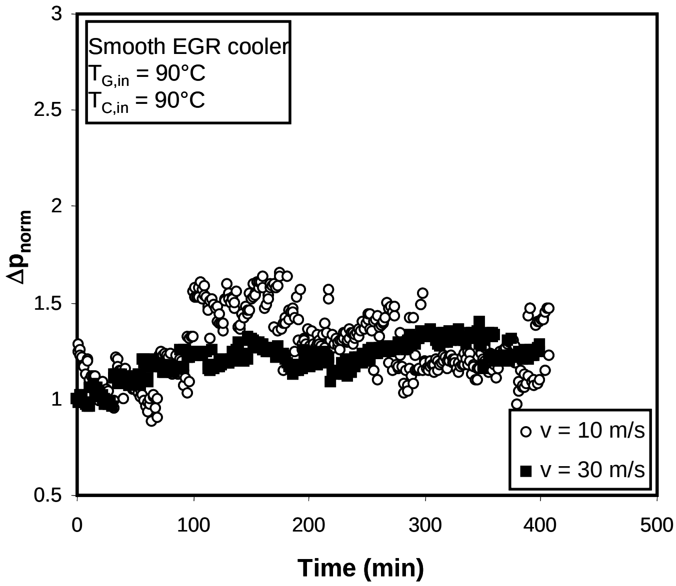


Fig. 3 Normalized pressure drop of the smooth cooler under isothermal conditions for different velocities.

For this particular cooler, the surface does not have any asperity or any other structure that would obstruct the movement of soot particles. As a result, the impaction of particles on the surface is minimal. This was confirmed with a visual inspection of the cooler, where only some soot particle deposition was observed inside the tubes for a velocity of 10 m/s. Noticeable soot deposition was observed on the inlet-header (see Fig. 4) for a gas velocity of 10 m/s. However, the deposition layer was more porous and coarse grained than the layer observed at an exhaust gas inlet temperature of 400°C that will be shown later on. For 30 m/s, the tubes were almost clean inside and inlet-header had minimal soot deposition as well. These results might be important in studying the formation of deposits on EGR valves or EGR cooler by-pass actuators. The EGR valve is typically located downstream of the EGR cooler and isothermal conditions could exist between the cooled exhaust gas and the valve surface, under low load engine operation. If the gas velocity is low enough soot particles might deposit on the valve surface similar to the deposition layer observed on the inlet-header at 10 m/s gas velocity. Condensation of hydrocarbons or water vapour in the exhaust might lead to densification of this deposit layer over time and cause the valve to get stuck.

(a) (b)

Fig. 4 Fouled smooth tube cooler under isothermal conditions for a) 10 m/s and b) 30 m/s.

The experiments under isothermal conditions were also repeated for the plate-fin cooler with oval inlet, and quite different behaviour was observed. Fig. 5 shows how the normalized pressure drop increases with decreasing flow velocity for the plate-fin cooler. The results show that unlike the smooth tube cooler, the normalized pressure drop increases continuously with time reaching a final value that is almost 2.5 times the initial pressure drop. In addition, the variation of normalized pressure drop differs for the two gas velocities. The presence of fins could be responsible for this behaviour as they may hamper the flow of soot particles and in turn promote impaction.

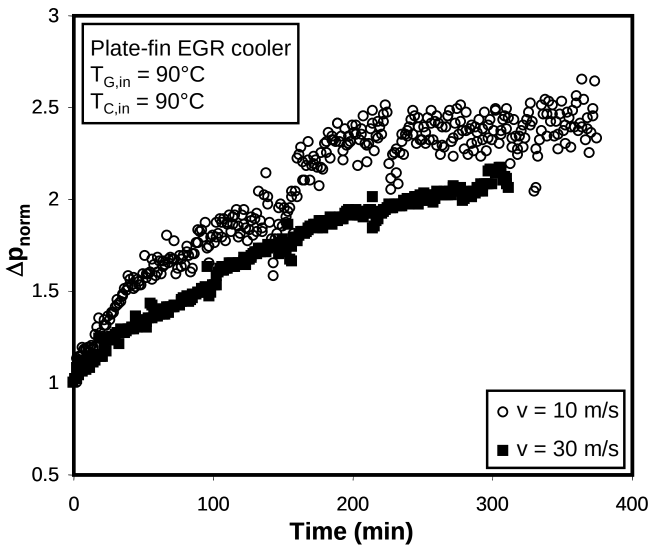


Fig. 5 Normalized pressure drop of the plate-fin cooler under isothermal conditions for different velocities.

It should also be pointed out that there are two competing factors in the formation of a deposit layer under isothermal conditions. Particle deposition due to driving forces such as diffusion, impaction and gravitational drift and particle removal mechanisms such fluid shear force. An increase in gas velocity leads to increased diffusion of particles towards areas with lower concentration i.e. the surface. Simultaneously the increased velocity is proportional to higher shear forces which in turn enhance removal. The results presented in Fig. 5 demonstrate that particle removal from the deposit layer is more significant as the gas velocity increases leading to lower net deposition of particles on the tube surface. Further, isothermal runs were conducted for the corrugated cooler, which resulted in similar behaviour.

Overall particle deposition is minimized under isothermal conditions compared to thermophoretic conditions. Deposition could still occur under isothermal conditions on surfaces such as the EGR valve or EGR cooler by-pass if the gas velocities are low enough such as under low load engine operation. However from these experiments it was observed that the deposit layer on the inlet-header was very porous and coarse compared to thermophoretic conditions and could thus be easily removed by an abrupt increase in gas velocity.

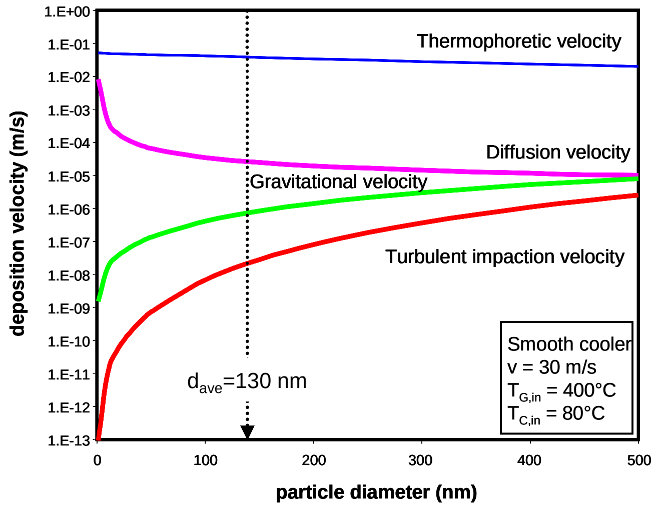


Fig. 6 Variation of different deposition velocities as a function of particle diameter.

3.2.2 Impact of Thermophoresis

Thermophoresis is the main deposition mechanism of particulate matter i.e. soot in EGR coolers. Chang et al. (1995) showed that the thermophoretic deposition increases with increasing temperature gradient. Fig. 6 shows the magnitude of various deposition velocities that influence particulate fouling of soot particles for a temperature gradient of 320°C at 30 m/s gas velocity for the smooth tube cooler (Warey et al., 2012). As it can be seen, the thermophoretic velocity is larger than the other deposition velocities by several orders of magnitude particularly for the average size of soot particles investigated in this study (130 nm). In the absence of thermophoresis, however, diffusion, gravitational drift and impaction may become considerable.

It should be emphasized that present modern diesel engines produce smaller average soot particles in the range of 50-80 nm. Nevertheless a glance at Fig. 6 shows that even for smaller particle size of 50-80 nm the thermophoretic force is still at least three orders of magnitude larger than the other forces. This would indicate that at isothermal conditions, in the absence of thermophoresis, the diffusion would become even more dominant compared to impaction and gravitational forces.

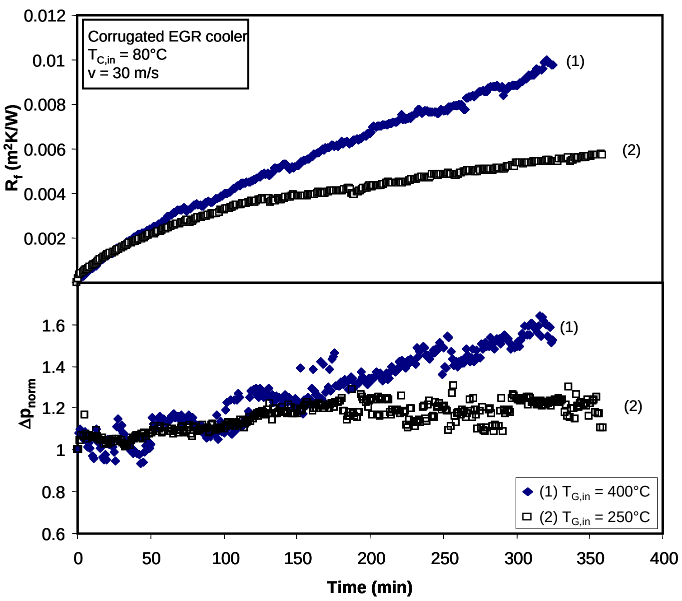


Fig. 7 Effect of temperature on thermophoresis in corrugated cooler for a given velocity of 30 m/s.

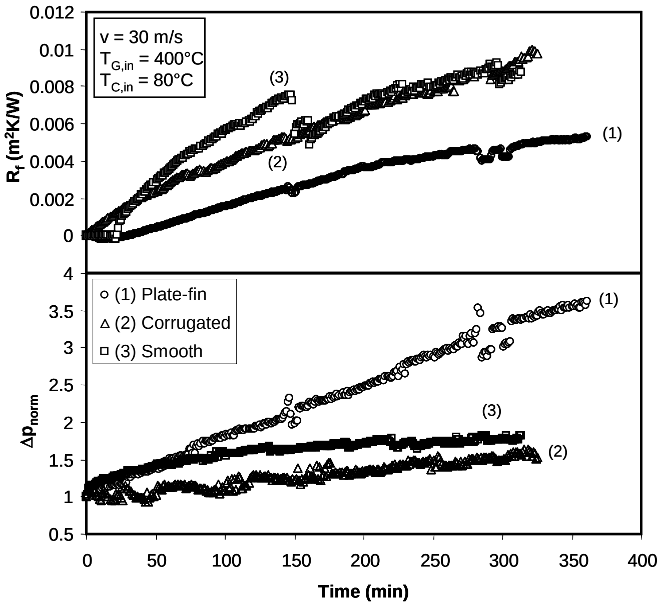


Fig. 8 Comparison of various cooler types under similar operating conditions.

Numerous experiments were performed for two temperature gradients of 170 and 320°C to characterize the extent of particulate fouling and to identify influential parameters. Fig. 7 illustrates how the fouling resistance and normalized pressure drop vary versus time for the corrugated cooler for a given velocity of 30 m/s. The fouling resistance and normalized pressure drop increase with time, but more notably for the higher temperature gradient as expected. The initial rate of increase of the fouling resistance is similar for both values of temperature gradient, but after about an hour of operation the fouling curve for ΔT of 170°C increases at a slower rate than the fouling resistance for ΔT of 320°C. However, a different trend is observed in variation of the normalized pressure drop. The normalized pressure drop for ΔT of 320°C increases more or less linearly with time, while the normalized pressure drop for ΔT of 170°C seems to reach an asymptotic value around 1.2 after about 3 hours of exposure. The fact that for 170°C, both fouling resistance and normalized pressure drop tend to level off might be due to increases in the interface temperature between the deposit layer and gas that reduces the thermophoretic force.

3.3 Comparison of various coolers

EGR coolers should usually be compact due to limited available space in cars but with maximum required effectiveness for cooling the exhaust gas. Furthermore they need to have a long life expectancy and should sustain harsh fouling conditions. Pressure drop across the cooler is another defining parameter which should always be kept below a certain limit in order to assure continuous and steady flow of cooled exhaust gas. This also has a direct impact on turbocharger efficiency and in turn on engine efficiency. The variation of fouling resistance and normalized pressure drop versus time for three cooler types are presented in Fig. 8 for a flow velocity of 30 m/s.

The thermal performance of the plate-fin cooler is apparently the best with respect to other two coolers in terms of fouling resistance. As for the normalized pressure drop, however, the corrugated cooler had the minimum increase in pressure drop, while the plate-fin cooler performed worst among all three coolers. After 5 hrs of operation, the normalized pressure drop for the plate-fin cooler increased up to 3 times the initial value, whereas the corrugated cooler showed an increase of only 1.5.

A closer scrutiny of Fig. 8 also reveals that if only the smooth and corrugated tube coolers are compared, the initial fouling resistance of the smooth tube cooler is higher than that of the corrugated tube cooler. Nevertheless after approximately 150 min they overlap and then continue to increase linearly. This was initially thought to be only an artefact of the fouling experiments where possibly part of deposit was spalled off the surface. Nevertheless no such abrupt change was noted for the normalized pressure drop data as shown in Fig. 8.

In order to verify whether this is a consistent trend, similar comparison was made in Fig. 9 for higher velocities of 70 and 80 m/s. It is evident that the fouling resistance of the smooth tube cooler even with a higher velocity of 80 m/s is greater than the corrugated cooler. Nonetheless both sets of experimental results overlapped again after approximately 200 min. As both coolers have similar inner diameter, it can be speculated that some of the corrugations inside the cooler tube are filled with soot particles leading it to behave somewhat like the smooth tube cooler. In addition, the fouling curves of both coolers exhibit a peculiar trend in which fouling resistances starts to fluctuate after approximately 100 min but more notably for the plain tube. This may be due to partial removal of the soot deposit layer from the surface due to exerted shear forces particularly for the high velocities shown in this figure. As for the plain tube, the fouling resistance somehow decreases sharply then the partial removal of the deposit layer can be termed as flake off. No such fluctuation was observed for 30 m/s (see Fig. 2). Additional experiments are obviously required before making a firm conclusion.

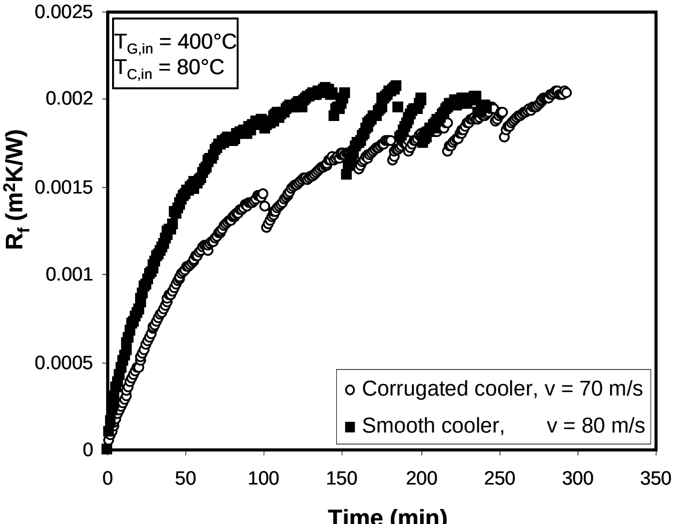
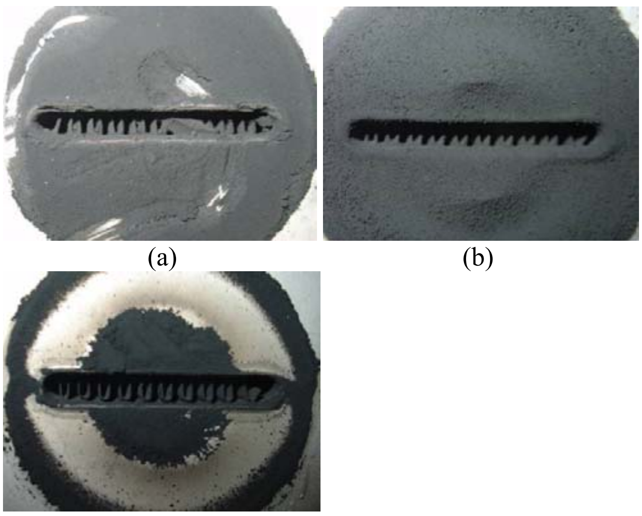


Fig. 9 Comparison of corrugated and smooth coolers for higher velocities of 70 and 80 m/s.

Fig. 10 shows the fouled entrance of the plate-fin cooler with oval inlet at the end of each fouling run for different temperature gradients. In case of ΔT=320°C, the entrance is partially blocked by soot deposits, which might explain the sharp increase in pressure drop for this particular cooler. In Fig. 10b there is no such blockage but there is significant deposition on the inlet-header. There is minimal soot deposition under isothermal conditions as seen in Fig. 10c. Under isothermal conditions, the soot deposit layer was more porous compared to ΔT=320°C and weakly attached to the surface so that it could be easily removed from the cooler.

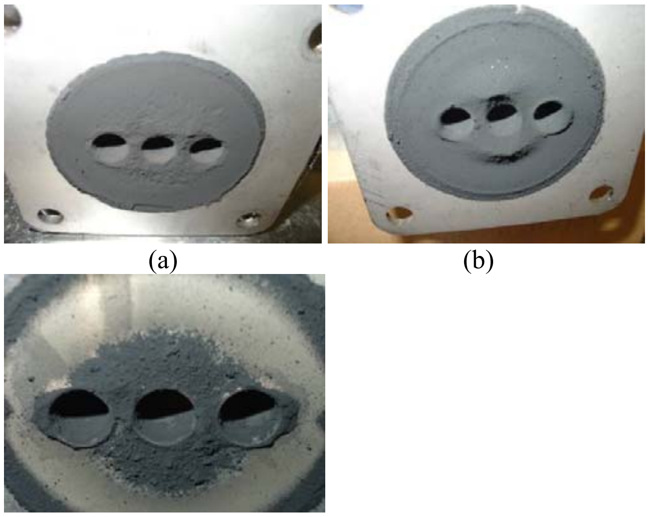


(a) (b)

(c)

Fig. 10 Inlet of fouled plate-fin cooler for ΔT of a) 320°C, b) 170°C and c) 0°C for 30 m/s.

The same behaviour was observed again for the corrugated tube cooler which is presented in Fig. 11 and also for the smooth tube cooler that is not shown here. Furthermore, for all cooler geometries it was observed, that the fouling layer was much more porous and coarse grained under isothermal conditions. Higher the temperature gradient inside the cooler, more compact the fouling layer and adhesion of the soot particles to the surface.



(a) (b)

(c)

Fig. 11 Inlet of fouled corrugated cooler for ΔT of a) 320°C, b) 170°C and c) 0°C for 30 m/s.

Table 3 Fouling resistance and loss in effectiveness after 4h of operation for various cooler types and temperature gradients.

Cooler	R_f (m²K/W)	R_f (m²K/W)	Loss in effectiveness $[(\varepsilon_c-\varepsilon)/\varepsilon_c]\times100$	Loss in effectiveness $[(\varepsilon_c-\varepsilon)/\varepsilon_c]\times100$
	320°C	170°C	320°C	170°C
Smooth	0.0084	0.0063	52.54	50.08
Corrugated	0.0077	0.0047	54.12	46.92
Plate-fin	0.0053	0.0051	18.02	14.37

Table 3 presents a summary of the results obtained after four hours of operation. It is noticeable that the loss in effectiveness is enormous with about 50% in only four hours for both corrugated and smooth tube coolers. In contrast to the circular tubes, the loss in effectiveness for the plate-fin cooler with oval inlet is only 18%. Furthermore the fouling resistance of the corrugated tubes is lower than for smooth tubes, but the plate-fin cooler performed best among the three investigated coolers at all conditions with respect to thermal efficiency.

From these results it can be concluded that cooler geometry plays an important role under thermophoretic conditions. Optimization of cooler geometry might be critical in minimizing fouling while maintaining good pressure drop characteristics.

4. CONCLUSIONS

The quantitative fouling results under isothermal and thermophoretic conditions presented in this paper facilitated a better understanding of soot deposition in EGR coolers with different geometries. Regardless of other technical considerations, increased gas velocity, low temperature gradient between inlet gas and coolant, and complex geometrical texture that would increase local heat transfer area are shown to reduce particulate fouling. Deposition under isothermal conditions is marginal as long as the surface is smooth and gas velocity is not too low. For the plate-fin cooler with extended surfaces, however, this may become appreciable due to impaction of soot particles on to the surface. The presence of spiral grooves in corrugated cooler initially helped to increase local shear forces. Nevertheless their advantages deteriorated as soon as the grooves were fouled by soot particles.

NOMENCLATURE

A_o	Overall heater transfer area of the EGR cooler tubes, m^2
A_{cr}	Cross-sectional area of the EGR cooler tubes, m^2
C_p	Heat capacity, J/kg.K
d_{ave}	Average diameter of soot particles, m
m_g	Mass flow rate of flowing gasses, kg/s
$\dot{Q}$	Rate of heat transfer, W
$R_{th}(t)$	Overall thermal resistance, m^2K/W
$R_{th}(c)$	Overall thermal resistance under clean conditions, m^2K/W
R_f	Fouling resistance, m^2K/W
t	Time, hr
T	Temperature, K
v	Gas velocity at the inlet to the EGR cooler, m/s
U	Heat transfer coefficient, W/m^2K

Greek symbols

ΔT	Temperature gradient between the inlet gas and coolant temperatures, K
ε	Effectiveness, -
ρ	Density of air, kg/m^3

Subscripts

C	Cooling water
c	Clean
f	Fouling
G	Gas
h	Hot gases
in	Inlet
out	Outlet

Abbreviations

EGR	Exhaust gas recirculation
LMTD	Logarithmic mean temperature difference, [K]
PM	Particulate matter

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