

An EGR Cooler Fouling Model: Experimental Correlation and Model Uses

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ABSTRACT

Thermal effectiveness of Exhaust Gas Recirculation (EGR) coolers used in diesel engines can progressively decrease and stabilize over time due to inner fouling layer of the cooler tubes. Thermophoretic force has been identified as the major cause of diesel exhaust soot fouling, and models are proposed in the literature but improvements in simulation are needed especially for the long-term trend of soot deposition. To describe the fouling stabilization behavior, a removal mechanism is required to account for stabilization of the soot layer. Observations from previous experiments on surrogate circular tubes suggest there are three primary factors to determine removal mechanisms: surface temperature, thickness, and shear velocity. Based on this hypothesis, we developed a 1D CFD fouling model for predicting the thermal effectiveness reduction of real EGR coolers. The model includes the two competing mechanisms mentioned that results in fouling balance. The deposition rate is calculated on a theoretical basis, while the removal rate calculation is based on empirical correlations derived from surrogate tubes. In combination with real world EGR cooler fouling data, the 1D fouling model is used to predict the long-term fouling behavior of production representative EGR coolers on a medium duty diesel engine for steady state conditions. Applicability of this model for various size and geometries of cooler tubes is discussed. Reduction of EGR cooler effectiveness is explored by running this model with regulatory driving cycle conditions and steady state conditions over the engine speed-load map.

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INTRODUCTION

EGR coolers are added to diesel EGR systems to reduce charge temperatures, increase EGR rates and reduce in cylinder temperatures. In addition to exhaust gas in combustion product, EGR coolers are exposed to condensable particulate matter (PM) and hydrocarbons (HC). These constituents deposit on the EGR cooler surfaces due to the temperature gradient between gas and cooled surfaces inside the heat exchanger. Fouling results in degraded heat transfer performance and increased pressure drop across the cooler and in extreme cases can even plug the cooler [1, 2]. Thermal effectiveness of EGR coolers decreases rapidly at first, and usually asymptotically stabilizes over time. As such, to meet heat rejection requirements or maintain flow rate as the cooler ages, the common solution is using a larger EGR cooler to offset passage area reduction or thermal resistance increment. To estimate the extra capacity, a fouling model that could predict fouling stabilization is necessary.

Soot fouling in EGR coolers is a multi-physics and multi-scale problem [3]. Physics involved include flow and thermal boundary layer, soot particle transportation and aggregation, hydrocarbon adsorption to soot particles, and hydrocarbon condensation on the heat exchanger wall. Its length scale spans from the 10^{-8} meter diameter of a soot particle to the 10^{-3} meter deposit layer thickness.

The resident time of a soot particle in a cooler tube can be as short as 10^{-2} second but time to reach cooler effectiveness stabilization is 10 or more hours. A model covering all physics involved is expansive due to space and time complexity. Instead a practical approach is simulating exhaust gas field under lower spatial dimensions, with a solid phase model for evolving deposit layer. Reduced methods are less accurate than full-dynamics modeling, but useful for quick sweeping over various conditions. Since current engines are calibrated with very low engine-out HC levels, we focus on the dominant insulation source, exhaust soot accumulation, and leave hydrocarbon condensation and interaction effect to the deposit layer for future study. To reduce run time and required computational power, the fouling model is based on macroscopic perspective, 1D CFD framework, and describe sub-micron detail such as deposit layer properties derived from experimental data.

Modeling of particle deposition was studied widely in various areas. Most of the literature proposed analytical thermophoretic deposition models, either for laminar flow [4], for developing flow with empirical fit in certain range of Prandtl number and thermophoretic coefficients [5], and for laminar and turbulent flow under any Prandtl number, Pr , and thermophoretic coefficient, K_{tr} [6]. In addition to analytical description of deposition efficiency by dimensionless parameter Pr and K_{tr} , thermophoretic Sherwood number Sh_{tr} was

factored in an analytical model [7]. Results of these works are mainly one-dimensional formulations of deposition efficiency and particle concentration, but particle accumulation on the cooler wall is not part of the solution. With assumptions of constant flow properties, insulation layer and boundary conditions, these analytical forms can be derived from energy conservation of the carrier flow and particle transportation equation. In summary, these models are limited to predict fouling rate for clean tubes.

A mathematical description of particle mass balance on the deposit layer is needed to predict the evolution of thermal effectiveness of EGR coolers. Abarham et al. proposed a 1D CFD model included variation of deposit layer thickness for non-isothermal tube flow [8]. Based on this framework, a model has been developed in Ford Motor Company to simulate diesel exhaust fouling stabilization for circular tubes, and favorable agreements with short-term experimental measurements were obtained (less than 5 hours). For long term tests, where experimental data shows asymptotic trend of exhausts outlet temperature, the model results in over-growth of deposition layer and eventually plugs the tube, which suggests an anti-accumulation or removal mechanism is missing. Here experimental data mentioned were conducted in Oakridge National Laboratory (ORNL) and the University of Michigan which are part of a research project of Ford Motor Company [9, 10].

The net rate of soot accumulated on the cooler wall can be expressed as the sum of deposition and removal rate, as shown in equation (1)

$$\frac{dm_{dep}}{dt} = \dot{m}_{foul} - \dot{m}_{rem} \quad (1)$$

For diesel exhaust fouling in EGR coolers, the deposition rate is dominated by thermophoresis. The deposition flux can be well-predicted by particle thermophoretic velocity formulation proposed by Talbot et al. [11] if the boundary conditions of tubes are known [7]. Several particle removal rates were proposed in the literature. Watkinson modeled the asymptotic fouling resistance of gas oil fouling as a function of fluid velocity [12]. Taborek et al. proposed the removal rate to be a function of shear stress, deposit thickness and bonding strength of the deposit for water in pipes [11]. Detailed mathematical form of the removal rate function depends on systems, but introducing these three factors into removal rate is rational for most of fouling problems.

Removal Rate Function

In diesel EGR coolers, the source of bonding strength between aggregated particles on deposit is the van der Waal's force and chemical bonding between volatile and soot. The physics behind the bonding strength is sufficiently complex that a molecular-level bonding strength model is difficult to integrate in the macroscopic 1D fouling model. One of the solutions is to bring the bonding strength into a removal rate function by introducing deposit temperature as a surrogate of bonding strength. Han et al. experimentally demonstrated three factors have strong relation to removal rate: in-situ pressure drop as a form of shear stress; deposit thickness; and surface temperature. A series of controlled experiments were conducted to statistically derive the soot removal rate of diesel exhaust in circular-

tube coolers as a function of pressure drop, heated deposit surface temperature, and deposit layer thickness [13]. The removal rate function extracted from data is shown in equation (2), where ΔP , T_{dep} , and δ are exhaust pressure drop due to shear stress, deposit surface temperature, and deposit thickness respectively.

$$\begin{aligned} \text{If } \Delta P < c_0 T_{dep} + c_1, \quad \dot{m}_{rem} &= 0 \\ \text{otherwise,} \quad \dot{m}_{rem} &= e^{(c_2 + c_3 T_{dep} + c_4 \delta + c_5 \Delta P)} \end{aligned} \quad (2)$$

Here $c_{j,j=0-5}$ are constant coefficients obtained from experimental data. Note that a threshold pressure drop as function of surface temperature was found. Below this pressure drop there is no soot particle collected at the outlet of tubes during blow out tests. Another interesting finding in this work is that deposited tubes with longer exposure (more than 12-hour deposition) show the peak thickness of deposit layer is at the center of tube, while shorter-exposure (6 hours) tubes have deposit peak at their inlet. Results from analytical models discussed [7] and 1D fouling model of Abraham et al. [8] without the removal function always shows decreasing deposition rate along the tube flow, which suggests deposition is controlled by radial temperature gradient in a cooler tube. Sul et al. integrated the removal rate function as given in equation (2) in the 1D fouling model and found the soot accumulation at the tube inlet was much lower due to higher local removal rate [14]. Their result shows the peak of deposit layer gradually shifts downstream with time and eventually stabilized, which agrees with the trend observed in experiments.

Extension to Production EGR Coolers

Because production EGR coolers often include turbulence inducing geometries to enhance heat transfer, it is necessary to understand the multi-dimensional effects on fouling stabilization. A cooled exhaust passage with parallel wavy wall was tested to illustrate two-dimensional effect on the soot fouling, as shown in Figure 1. The height of the wavy channel is 1.4mm and the aspect ratio (width/height) of its cross section is 16. Exhaust conditions are 4.5kg/hour flow rate at 483K inlet temperature, 1.5 FSN (smoke number), and 70°C coolant temperature. The resulting inlet exhaust velocity is 70m/s, which is beyond regular EGR operating conditions. Under these conditions, both thermophoretic deposition and removal rate due to shear stress are strong such that deposit stabilization is reached shortly.

There are three findings in this experiment. First, staggered deposit distribution is observed after 6-hour of exhaust exposure, as shown in Figure 2. Soot accumulation exists on the downstream side of the peaks while bare metal is observed near the peaks due to stronger removal rate. Second, deposit width between two bare-metal regions decreases when moving toward downstream. Third, deposit layer is thicker at the inlet and thinner at the outlet, as given in Figure 3. The third observation is due to the decreasing thermophoretic deposition rate, which determines soot distribution globally for short-term exposure. On the other hand, removal rate based on equation (2) should hardly change from the inlet to the outlet, but vary substantially in a local way. It enhances the shear stress at the upslope near the peaks, and may create recirculation zone at the valley peaks such that a shear stress-induced removal is very limited. The flow

passages with deposit may evolve progressively toward a flatter, less abrupt change features. This finding suggests that the 2D effect may warrant the development of a 2D fouling model, although the 1D generalization may still be sufficient for long-time fouling simulation, as long as the heat transfer coefficients are calibrated. We will see later in this paper that the 1D model is predictive for “wavy fin” real world cooler geometries.



Figure 1. Side view of a wavy-fin insert in a cooled exhaust channel. Pitch between peaks is 9.4mm and height of the passage is 1.35mm, which is a typical dimension of flow passages in production EGR coolers with wavy fins for heat transfer enhancement.

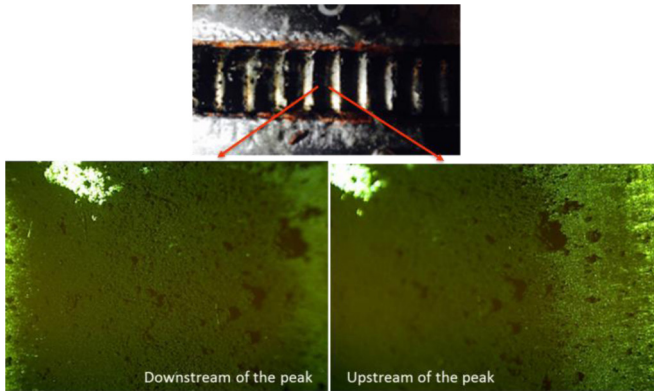


Figure 2. Top views of deposit layer at the downslope-wind side, valley, and upslope-wind side of the wavy channel. Bare metal is seen near the peak of wavy channel.

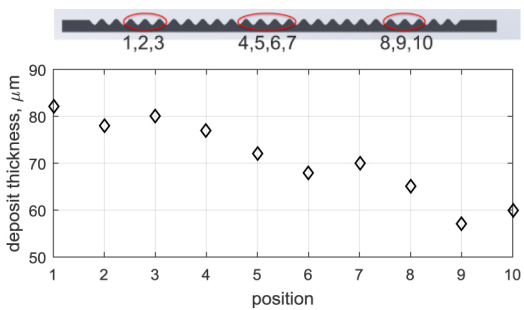


Figure 3. Deposit layer thickness at the downslope side of peaks along the channel after 6-hour exposure. Thickness along the tube was measured using the 3D imaging technique developed by Salvi et al [15].

1D MODEL

The computational domain considered is one of the exhaust gas passages constituting the coolers. Exhaust flow rate is scaled down accordingly. Rectangular flow passages or more complex configuration such as wavy fin are approximated as one-dimensional circular domain with an equivalent hydraulic diameter. The cross section area of the domain reduces with growing deposit layer, which is represented by a time-dependent moving boundary. Constant cooling wall temperature at coolant side is assumed over the domain due to negligible spatial variation of coolant temperature under current EGR system design. The cooler wall material is stainless steel for most of production EGR coolers.

The computational domain includes exhaust gas passage with a diameter D_{int} and a time-dependent deposit layer in between gas and cooler wall as shown in Figure 4. Steady-state transport equations are solved at the discretized domain for mean exhaust pressure P , temperature T , velocity u , density ρ , exhaust-carried soot mass concentration Y , and deposit layer thickness δ , as functions of axial distance from the cooler inlet. Exhaust properties such as specific heat, viscosity, and thermal conductivity are assumed to be constant over the domain at the average of inlet and outlet temperature for a time step. The fouling stabilization is assumed to be a quasi-steady state problem for the time scale of deposit layer growth is much slower than the transportation of exhaust gas in the cooler. As a result, steady state governing equations are used for exhaust gas and air-borne soot particle, and the transient mass conservation of deposit mass is considered as given in equation (1).

Other assumptions of the model are as follows: Exhaust flow is incompressible. To reduce computation cost, turbulent kinetic energy dissipation is ignored. Convection heat transfer in the radial direction is described by empirical correlations for pipe flow for various Reynolds number conditions. Entrance effect is considered because higher heat and shear stress are expected in the inlet regions. Pressure drop is calculated by friction loss according to friction factor correlations in the pipe flow and momentum change due to cross-section area variation. For soot particles, the source terms in the transportation considered are thermophoretic deposition and Brownian diffusion. The diameter of soot particles and its thermal properties such as conductivity are assumed to be constant.

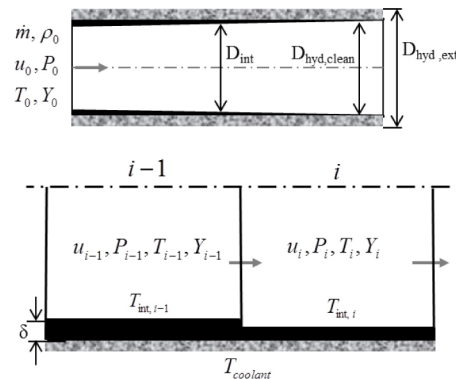


Figure 4. Computational domain for 1D model

Transport Equations

Equations (3)-(5) are the mass conservation, momentum, energy equations of the exhaust flow solved in this model. Note that the Darcy friction factor, f , is calculated by the Zigrang-Sylvester's correlation from local Reynolds number, roughness, and hydraulic diameter D_{int} [16]. The cross section area A should change with the exhaust gas hydraulic diameter due to the growth of deposit layer.

$$\frac{\partial(\rho Au)}{\partial x} = 0 \quad (3)$$