

Transient Fluid Flow and Heat Transfer in the EGR Cooler

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ABSTRACT

EGR is a proven technology used to reduce NO_x formation in both compression and spark ignition engines by reducing the combustion temperature. In order to further increase its efficiency the recirculated gases are subjected to cooling. However, this leads to a higher load on the cooling system of the engine, thus requiring a larger radiator. In the case of turbocharged engines the large variations of the pressures, especially in the exhaust manifold, produce a highly pulsating EGR flow leading to non-steady-state heat transfer in the cooler. The current research presents a method of determining the pulsating flow field and the instantaneous heat transfer in the EGR heat exchanger. The processes are simulated using the CFD code FIRE (AVL) and the results are subjected to validation by comparison with the experimental data obtained on a 2.5 liter, four cylinder, common rail and turbocharged diesel engine.

INTRODUCTION

Exhaust Gas Recirculation (EGR) used in both spark ignited (SI) and compression ignition (CI) engines with the purpose to reduce NO_x emissions has been preferred over other methods of charge temperature reduction for its advantages such as reduced oxygen availability, increased specific heat, relative ease of implementation and effectiveness [1], [2], [3].

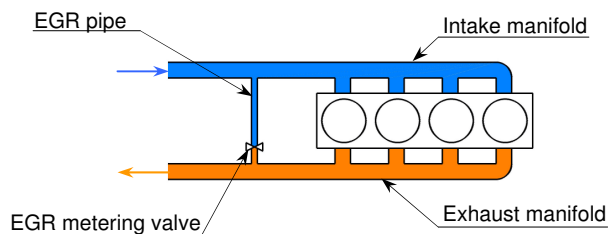


Fig. 1 A simple EGR system

A simplified EGR system is presented in Fig. 1. The intake and exhaust manifolds are connected by a pipe and the EGR flow is regulated by the EGR valve. Of

course, the system gets considerably more complex compared to this representation when other components are added such as turbine, compressor, cooler, flow-meters, etc. This paper focuses on a particular type of EGR, mainly high pressure EGR (HPEGR) (Fig. 2)[4]. It is widely used in modern turbocharged diesel engines because it has a simple design and avoids soot carrying exhaust reaching the blades of the compressor as in the case of low pressure EGR.

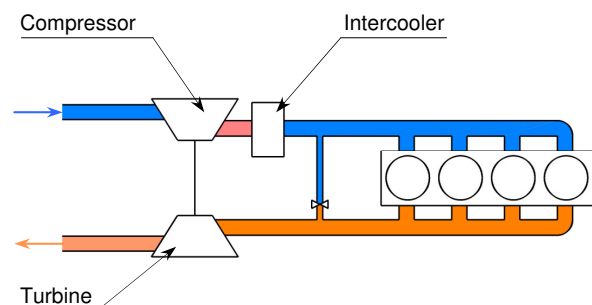


Fig. 2 High Pressure EGR (HPEGR) system

However, there are certain drawbacks associated with HPEGR, and one of them is difficult control. The metering valve has to regulate a highly pulsating flow with variable pressure drop across the EGR cooler as this component gets clogged with soot (more generally known as heat exchanger fouling), a well known and documented problem with EGR systems [5]. Ismail et al [6] performed deposit thickness measurements and concluded that the deposits are non-uniform and with complex shapes.

Unlike the DPF, the EGR cooler cannot be easily cleaned (regenerated) by high exhaust temperature. Actually the only way available today to clean it is during the maintenance of the vehicle. Chemical methods for cleaning the EGR cooler are usually preferred. Other methods such as acoustic or mechanical vibrations will not work well for the type of deposits present in the EGR cooler [7]. More recent investigations [8] have shown attempts for strategies of cleaning a fouled EGR cooler during engine operation. These strategies were proved to be ineffective.

According to research done by Zhang [5], after approximately 10 hours of operation the soot deposits in the EGR cooler were fully developed. This adds to the complexity of the problem, as the variable size/geometry of those deposits will increase the pressure drop across the EGR system for a given EGR flowrate. The EGR valve will have to be opened additionally in order to compensate for the pressure loss. However, as the maximum opening of the valve is reached, the EGR flowrate will become lower than required [9].

An added difficulty in the control of the HPEGR is the variable boost pressure in the intake manifold and the superimposed effect of the pressure waves in both intake and exhaust. For high boost pressures, the EGR valve has to be closed; otherwise the intake air will flow into the exhaust manifold. When such conditions occur, alternative methods have to be used in order to ensure proper pressure difference, such as VGT, exhaust pressure valve, etc [10]

Given the challenges that this technology faces, a better understanding of the flow phenomena in the EGR system is necessary. This paper provides insight into "boundary" flows, meaning operating conditions at which the high boost pressure and pressure waves might make the reversed flow possible in the EGR system. The focus is going to be kept mainly towards the EGR cooler and the model developed is going to take into account all the important physical phenomena, such as heat transfer by convection and conduction.

FORMATION OF SOOT DEPOSITS IN THE EGR COOLER

Unless there is a particulate filter upstream of the EGR system, the exhaust gases flowing through the EGR cooler will contain soot particles of different size. Some of those particles will be deposited on the walls of the heat exchanger due to a number of reasons [9]:

- Turbulent fluid flow; the carbonaceous particles will be deposited by impact with the surface
- Condensation of hydrocarbons; it will create the proper environment for the soot particles to "stick" to the surface
- Diffusion due to gradient in concentration [8]
- Thermophoresis; also known as Ludwig-Soret effect, will determine a flux of carbonaceous particles from the hot exhaust to the cold walls. The force which acts on a given soot particle is given by [11]:

$$F_{th} = cpd^2l \frac{\Delta T}{T} \quad (1)$$

Where c is a constant, p is the exhaust gas pressure, d is the particle diameter, l is the mean free path of the soot particles and T is the absolute temperature and ΔT

is the temperature difference between the hot and cold regions of the fluid, in the case of the EGR cooler, the temperature difference between exhaust gas and the wall.

The above equation presents the parameters involved in this phenomenon. Thus, the higher the particle size and the larger the temperature gradient, more particles will be deposited on the surface of the heat exchanger. Also, turbocharging and pressure pulsations increase the pressure of the exhaust gas which in turn further increase the deposition rate. While there is not much that can be done with respect to pressure and particle size, care has to be taken not to overcool the EGR, which may cause other inconveniences, such as high hydrocarbon and sulfuric acid condensation which may have damaging effects for the integrity of the EGR system and of the engine [12].

To avoid condensation effects, cooled exhaust gas temperature should be kept above 120°C (393K) [12]. With this in mind, Chalgren et al [13] proposed a method to regulate the EGR temperature at a level of 130±8°C, by using a PID controller. Alternative shapes to the classical tubes are sought; one of them [14] uses profiled tubes that increase the speed of the fluid near the surface of the heat exchanger in order to reduce soot deposition.

CFD MODELING APPROACH

The highly non-steady-state flow of the EGR through the system makes its measurement difficult and to assess the actual amount of EGR and its state parameters requires detailed simulation of this process. The simulation presented in this paper is based on the AVL FIRE CFD code version 8.50. The mathematical procedure used by FIRE will be briefly discussed. Detailed information regarding the FIRE solver may be found in ref [15]. The general integral equation is given by [15]:

$$\begin{aligned} & \overbrace{\frac{d}{dt} \int_V \rho \phi dV}^{\text{Rate of change of } \phi} + \overbrace{\sum_{j=1}^n \int_{A_j} \rho \phi (U_k - U_{bk}) dA_k}^{\text{Convection of } \phi} = \\ & \underbrace{\sum_{j=1}^{n_f} \int_{A_j} \Gamma_{\phi}^{kk} \frac{\partial \phi}{\partial x_k} dA_k}_{\text{Diffusion of } \phi} + \underbrace{\int_V s_{\phi}^V dV + \sum_{j=1}^{n_f} \int_{A_j} s_{\phi}^A dA_k}_{\text{Source terms}} \end{aligned} \quad (2)$$

where ϕ represents every mean flow dependent variable that needs to be solved for. s_{ϕ}^V and s_{ϕ}^A are volumetric and surface source terms respectively and Γ_{ϕ}^{kk} is a diffusion coefficient.

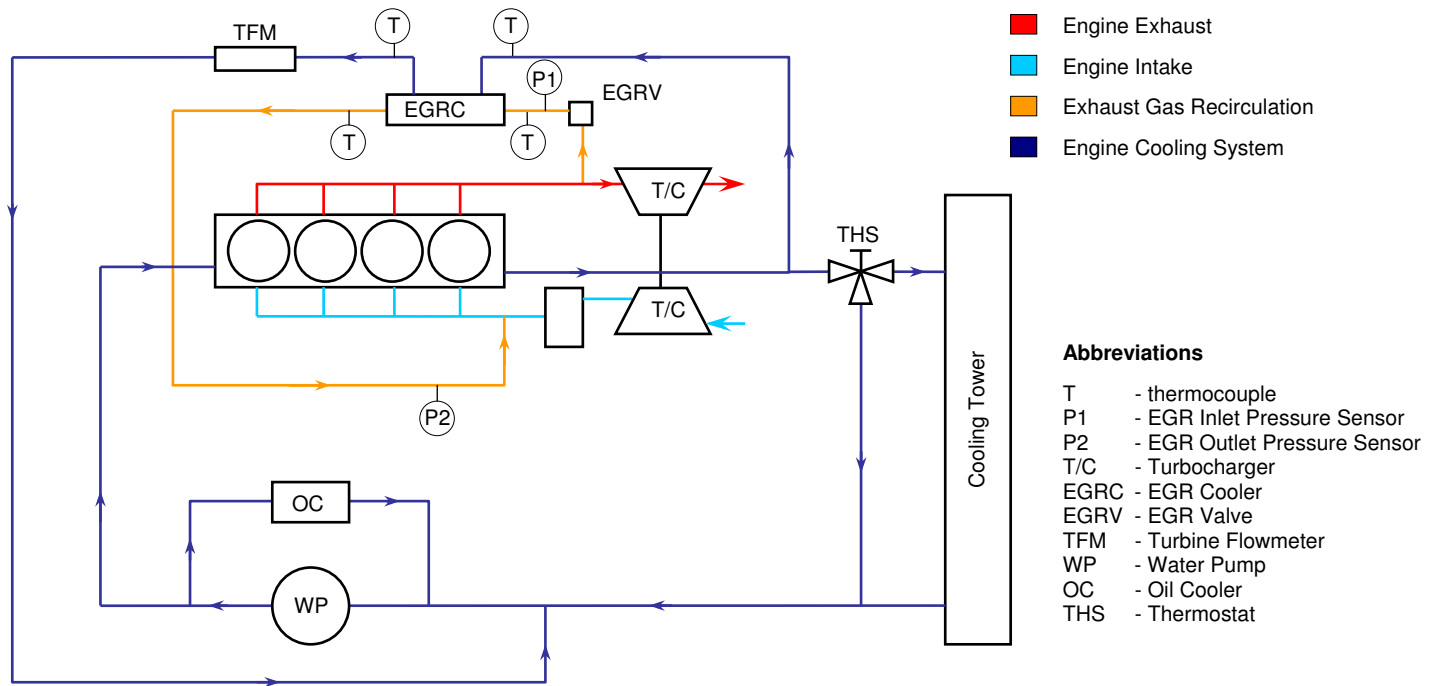


Fig. 3 Engine instrumentation diagram [4]

Each dependent variable ϕ has its own set of source terms and diffusion coefficient. FIRE solves for the following variables $\phi: U_i, k, \varepsilon, \overline{u_i u_j}, [c]_i$ and H (see the Notations section for explanation).

An additional form of the integral equation is obtained if the dependent variable is considered to be unity $\phi = 1$, then:

$$\Gamma_H = 0, s_\phi^V = 0 \text{ and } s_\phi^A = 0$$

and equation (2) becomes the continuity equation:

$$\frac{d}{dt} \int_V \rho dV + \sum_{j=1}^n \int_{A_j} \rho (U_k - U_{bk}) dA_k = 0 \quad (2')$$

The resulting set of integrals will form a system of algebraic equations which needs to be solved. These equations are non-linear and coupled, which makes the solution difficult to obtain. Turbulence modeling has been considered in the simulation by use of the well-known k- ε closure.

The CFD simulation used in this paper assumes temperature dependent fluid properties and fluid compressibility, thus adding extra degrees of complexity. A good stability of the FIRE code was noticed; minor modifications (e.g. under-relaxation factors, differencing schemes) to the solver parameters were sufficient to ensure convergence.

EXPERIMENTAL SETUP

The diagram of the experimental setup is shown in Fig. 3 and the picture showing the instrumented EGR is presented in Fig. 4. The basic data of the engine used in this work are given in Table 1.

Table 1. 2.5L diesel engine parameters

Total swept volume	2.5 L
Number of cylinders	4
Bore	92 mm
Stroke	94 mm
Compression Ratio	17:1
Rated Power	103 kW @ 4000rpm
Rated Torque	340 Nm @ 2000rpm
Maximum boost pressure	2.3 bar
Maximum injection pressure	1350 bar

The engine is loaded by an AVL ELIN AC dynamometer controlled by the AVL PUMA Open 1.2 platform and the tests are conducted by requesting a constant dynamometer speed and continuously adjusting the pedal position sensor of the engine in order to maintain constant engine torque. The EGR system of this production engine, which includes a Valeo tube in shell heat exchanger (Fig. 6) has been instrumented to measure pressures, temperatures and coolant rate of flow. Two Kistler 4075A10 absolute pressure transducers have been installed at each end of the EGR pipe (identified with P1 and P2 in Fig. 3). These sensors have been mounted in the Kistler 7531 cooling adapter

in order to allow the transducers to operate in the high-temperature exhaust environment. Type K thermocouples with 0.02" beads have been used at the inlet and outlet of both exhaust and coolant circuits of the EGR cooler.

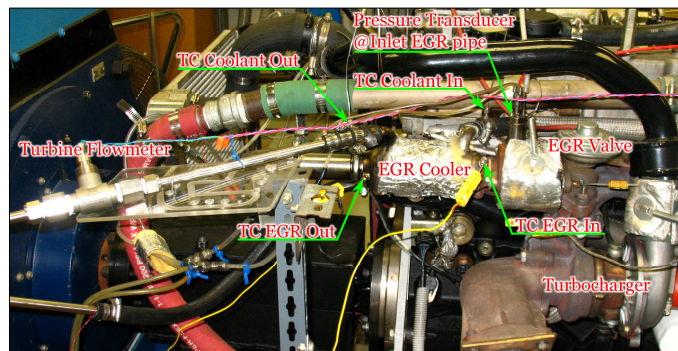


Fig. 4 Engine Experimental Setup

An Omega FTB-903 turbine flowmeter and the DPF-701 totalizer provided data regarding instantaneous coolant flowrate through the EGR cooler. The fuel mass flowrate has been measured using AVL 733 Fuel Flowmeter. Data has been acquired crank-angle based using a 16 channel AVL Indicom system with software version 1.6. Time-based data was acquired using the FEM of the PUMA system. The fuel used in the current experiments was commercial Ultra Low Sulfur Diesel (ULSD).

EXHAUST PRESSURE TRANSDUCERS CALIBRATION/VALIDATION

The pressure data obtained from the two pressure transducers is used as input and fed into a complete CFD model of the EGR system. For this purpose the difference in between the response of the two transducers is very important. The factory calibration has been performed for a wide pressure range (1-10 bars), but the pressures measured in the EGR system will be much lower.

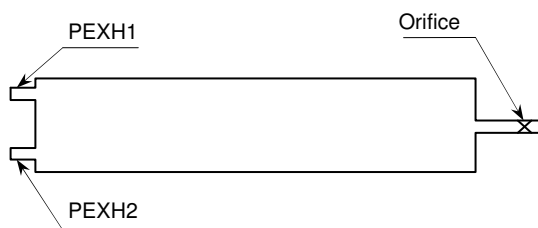


Fig. 5 Exhaust pressure transducers calibration rig

In order to verify the transducers and validate their response in the pressure range of 1-3 bar absolute, a calibration rig has been built (Fig. 5). The reservoir is filled with atmospheric air at 3 bars. As the air in the reservoir is discharged through an orifice, the pressure is recorded simultaneously by both pressure transducers. Ideally, the response should be the same, as they are exposed to the same pressure. However, a

slight deviation from this ideal behavior has been noticed when the factory calibration has been used. There was a small difference in between the two signals which was corrected by providing new calibration constants.

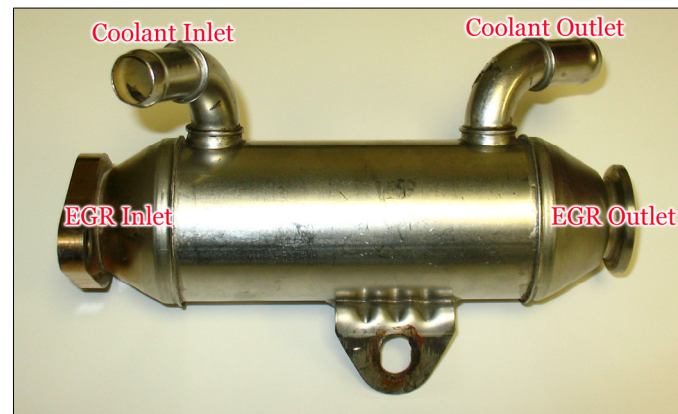


Fig. 6 Tube & Shell EGR cooler

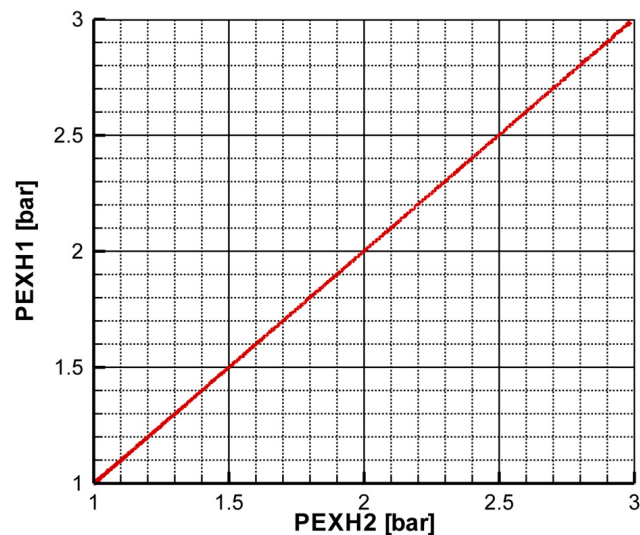


Fig. 7 Pressure transducers response on the rig

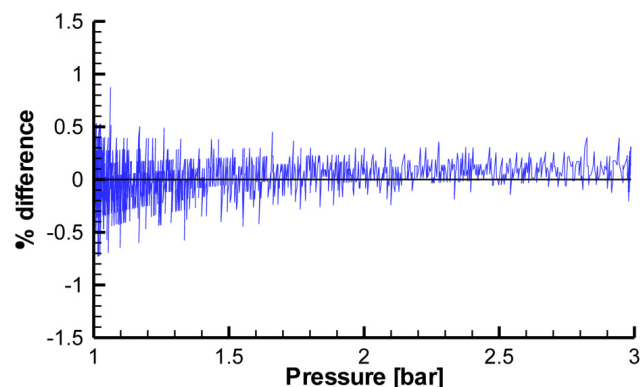


Fig. 8 Difference between pressure transducers