Heat Transfer Phenomena During Engine Cold Start Operation

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1. Abstract

Regulated emissions are in-vehicle controlled by aftertreatment devices such as catalysts and particulate filters. Both have a common necessity in order to perform adequately, temperature. As the average commutes in Europe are short thus characterized by low engine and exhaust temperatures these devices only start to operate at optimum levels at a late stage of the drive. In this research paper, the phenomenon of heat transfer in an exhaust system is modelled under cold start engine transient conditions. We propose a model which is applicable to both gasoline and diesel powertrains, where there is at least one aftertreatment device present in a single wall exhaust system. It enables the prediction of the temperatures, of the exhaust manifold as well as temperatures of the catalysts\filters and piping, over time. Special attention was given to the particular cases of wet surface heat transfer (due to dew point temperature) and subsequent dry surface heat transfer. In the first case, the Dittus-Boelter correlation was used, implying a homogenous temperature of the exhaust fluid, with a wet surface as the heat transfer media. On the second case, the Gnielinksi correlation for turbulent flows attached to a calculated factor due to the pulsating nature of exhaust gases as they exit the combustion chamber. To analyse the pipe-exterior interaction, the Churchill-Chu correlation for convection was used. The catalytic converter\filter heat transfer was assessed by performing both heat and mass balances. The model is expected to be validated within a short period of time.

2. Introduction

During the cold start phase the exhaust gases include, overall, more emissions than normal operating temperature gasoline and diesel-engine running vehicles. Temperature is the key elements during this phase were the aftertreatment devices are not performing at optimum levels.

Cold start exhaust emissions are also strongly influenced by the low cylinder wall and coolant temperature that prolong ignition delay and lead to incomplete combustion. During turbocharged diesel engine cold starting, the engine speed is lower than the required governing speed, so the fuel pump rack is for supply. The fuel-air ratio is higher owing to the mismatch between fuelling and air-supply. When the combustion becomes stable and the engine speed reaches or exceeds the governing self-sustained speed the rack will reduce fuelling with a subsequent decreased of soot, unburned hydrocarbons and nitrogen oxides emissions.

There are many models that mimic aftertreatment devices' heat exchange but most do not take into consideration the specificities of the cold start problem. Also, the presence of various emission control devices shapes more complexity in the problem as manufacturers tend to develop, nowadays, proprietary exhaust architectures.

Regarding the causes of the poor performance of the catalysts and filtering devices, two main factors determine the quality of the combustion – low ambient temperature and low cranking speeds as explained in figure 1.

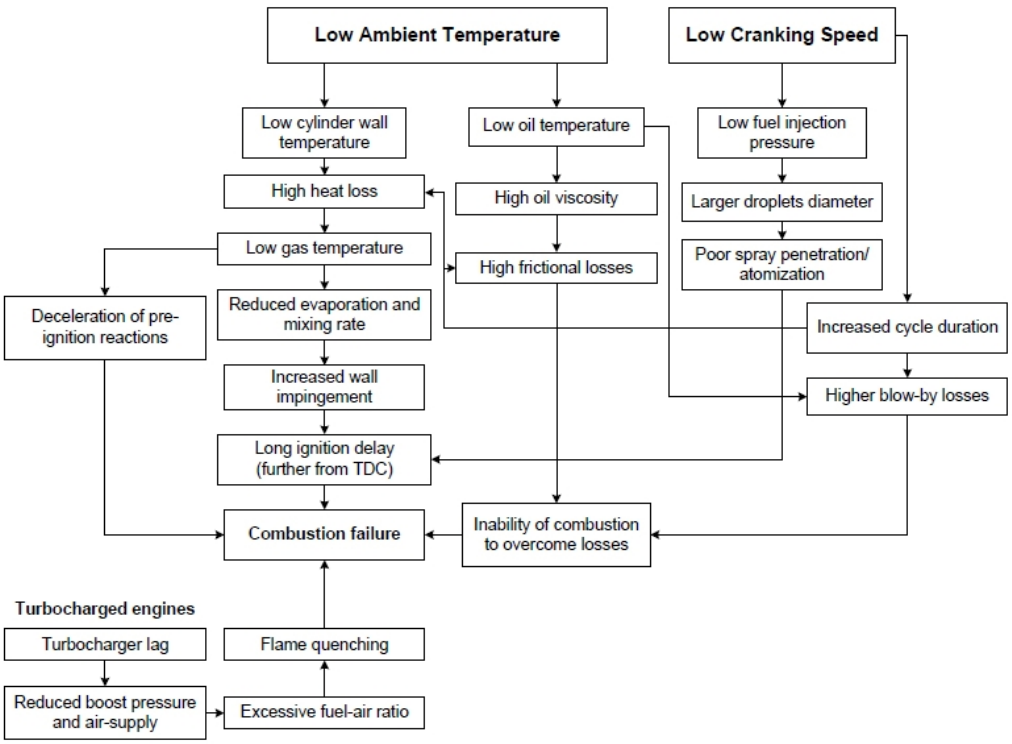

Figure 1 – Factors that determine combustion quality

3. Exhaust system modelling

In order to perform the modelling of the exhaust system, specific exhaust architecture was chosen, a simple 4-to-one manifold attached to a catalyst (NOx trap). A similar system is used by many manufacturers, such as Volkswagen in the VW Golf IV, a very common vehicle on our streets.

3.1 Heat Transfer in exhaust pipe

The figure below shows schematically the pipe flow and its associated heat transfer elements that are represented by a control volume.

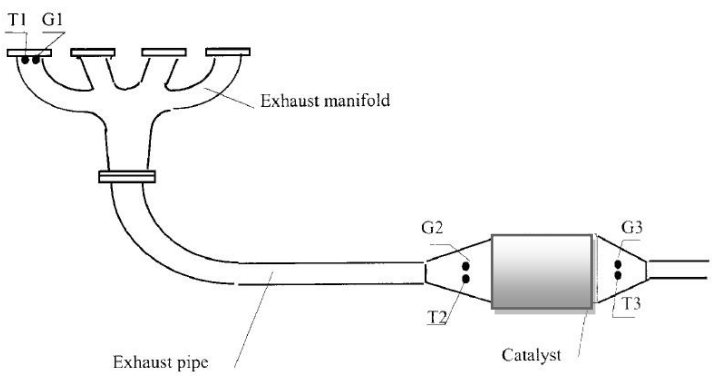

Figure 2 Schematic layout of exhaust system (4-to-one)

The flow condition at any location of interest is described by three independent variables, velocity, density and pressure. Assuming a quasi-steady, incompressible flow, the one dimensional energy equations for the exhaust gas and pipe wall take the following form:

$$\frac{\partial T_g}{\partial t} + u\frac{\partial T_g}{\partial t} = -\frac{q_{gp}}{\rho_g C_{pg} V_1} \qquad [3.1]$$

$$\frac{\partial Tp}{\partial t} = \alpha\frac{\partial^2 T_p}{\partial x^2} + \frac{q_{gp} - q_{cva} - q_{rad}}{\rho_p c_p V_2} \qquad [3.2]$$

Equation 3.2 is a one-dimensional unsteady heat conduction with uniform thermal conductivity applied to the pipe wall.

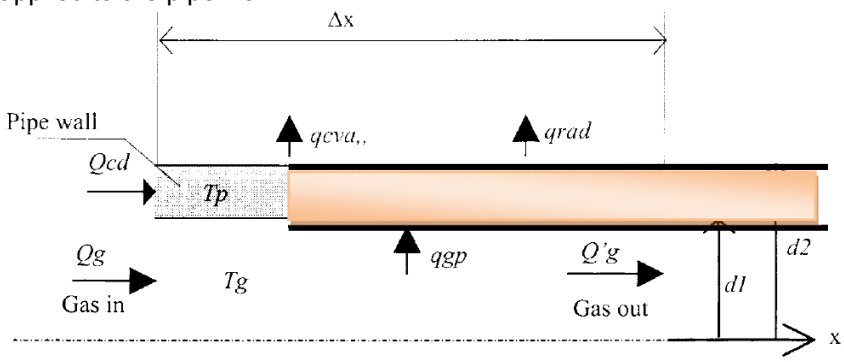

Figure 3 Control volume of an exhausted pipe

3.2 Heat convection from exhaust gas to pipe wall

The heat flux from the exhaust gas to the inner wall of the exhaust pipe may be expressed as:

$$q_{gp} = h_{gp}\pi d_1 \Delta x(T_g - T_p) \qquad [3.3]$$

Where hgp is the convective heat transfer coefficient between the exhaust gas and the pipe wall, and is determined by the following Nusselt number:

$$N_u = \frac{h_{gp} d_1}{k_g} \qquad [3.4]$$

At engine cold-start, the cold pipe walls quench the hot exhaust gases from the engine, thus forming a thin layer of water film on the pipe wall affecting the heat transfer process. Two cases are considered in this study, the wet wall and the dry wall heat transfer.

3.3 Wet surface heat transfer

The wet surface serves as the media for heat transfer from the exhaust gas to the pipe wall. Heat transfer phenomena during the engine cold-start period is very complicated as it involves liquid and vapor phases, and is beyond the scope of the present study. Instead, and empirical correlation for wet surface heat transfer is adopted to account for the effect of water film on the heat transfer. A commonly used Dittus-Boelter correlation for wet surface heat transfer is adopted.

$$N_u = \frac{h_{gp} d_1}{k_g} = 0{,}023 Pr^{0,4} Re^{0,8} \qquad [3.5]$$

Where

$$Re = \frac{G d_1}{\mu} \qquad [3.6]$$

$$Pr = \frac{c_p \mu}{k_g} \qquad [3.7]$$

$$G = \frac{4\dot{m}}{\pi d_1^2} \qquad [3.8]$$

The thermal properties of the gas are evaluated at (Tg + Tsat)/2.

3.3 Dry surface heat transfer

Assuming that the exhausted pipe is fully warmed, the Nusselt correlation is based on the average flow rate of the exhaust gas. There are a number of heat transfer correlations available, the correlation proposed by Gnielinski [13] was the one chosen:

For 104 < Re < 5 × 106

$$N_u = \frac{\left(\frac{f}{8}\right)(Re)Pr}{1{,}07+12{,}7\left(\frac{f}{8}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}}-1\right)} \qquad [3.9]$$

For Re < 104

$$N_u = \frac{\left(\frac{f}{8}\right)(Re-100)Pr}{1{,}07+12{,}7\left(\frac{f}{8}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}}-1\right)} \qquad [3.10]$$

Where the friction factor (f) is determined by the following equation:

$$\frac{1}{\sqrt{f}} + 2\log\left(\frac{\varepsilon}{3{,}7d} + \frac{2.51}{Re\sqrt{f}}\right) = 0 \qquad [3.11]$$

To take into account the effects of the bended pipe on the heat transfer coefficient, the correlation proposed by Hausen for this factor, is used,

$$F_B = \frac{N_{u\,bent-pipe}}{N_u} = 1 + \frac{21 d_1}{Re^{0,14} d_{bent}} \qquad [3.12]$$

As the exhaust gas release from the combustion chamber is not continuous, due to the nature of the engine and valve actuators, a gas pulsating factor (Fpul), with values between 1.6 – 3, was introduced in the Nusselt correlation in order to represent this behavior. Hence, equations 3.9 and 3.10 become,

For 104 < Re < 5 x 106

$$N_u = \frac{F_B \times F_{pul} \times \frac{f}{8} \times Re \times Pr}{1{,}07+12{,}7\left(\frac{f}{8}\right)^{1/2} \times \left(Pr^{\frac{2}{3}}-1\right)} \qquad [3.13]$$

For Re < 104 $$N_u = \frac{F_B \times F_{pul} \times \frac{f}{8} \times (Re - 1000) \times Pr}{1{,}07 + 12{,}7(\frac{f}{8})^{1/2} \times (Pr^{\frac{2}{3}} - 1)} \quad [3.14]$$

3.4 Heat convection from pipe surface to ambient air

The heat transfer from the outer surface of the pipe to the ambient air is by free convection. Therefore:

$$q_{cva} = h_{cva} \pi r_2^2 \Delta x (T_p - T_a) \quad [3.15]$$

Where

$$h_{cva} = \frac{N_u k_a}{d_2} \quad [3.16]$$

The Nusselt number is determined from the correlation proposed by Churchill and Chu as below:

$$N_u = 0{,}6 + \left(\frac{0{,}387 Ra^{\frac{1}{6}}}{[1 + (0{,}599/\mathrm{Pr})^{\frac{9}{16}}]^{\frac{8}{27}}}\right) \quad [3.17]$$

Where

$$R_a = \frac{l^3 g \beta (T_p - T_a}{\nu_a^2} Pr \quad [3.18]$$

and the wetted length is given by: $l = \pi d_2$ [3.19]

The β is evaluated at T_a, while the other properties are assessed at the film temperature, T_f,

$$T_f = T_p - 0{,}38\,(T_p - T_a) \quad [3.20]$$

3.5 Heat radiation from pipe surface to surroundings

The radiative heat transfer rate from the pipe surface to surroundings is given by,

$$q_{rad} = \varepsilon \sigma \pi r_2^2 (T_p^4 - T_a^4) \Delta x \quad [3.21]$$

4. Heat and mass transfers and catalytic converter

Heat and mass transfers and chemical reactions occur simultaneously in the catalytic converter. The whole process is quite different for a fully warmed up catalytic converter and a warming up catalytic converter at engine cold start. In the early stage of the latter, condensation of water vapour present in the hot exhaust gas onto the surface of the monolith cells occurs. When the temperature of the cells exceeds the dew point temperature of the water vapour, evaporation dominates and the process ceases when the monoliths cells are in thermal equilibrium with the exhaust gas. The reactions take place at a very thin boundary layer in the neighbourhood of a gas/solid interface At the vicinity of this boundary layer, the concentrations of main gas species are smaller than those in the mainstream through the cell channels (due to the chemical reaction) causing a mass transfer, which is characterized by a mass transfer coefficient hDj .

The overall reactions are as follows:

$$R_1 = CO + \frac{1}{2}O_2 \rightarrow CO_2 \qquad [4.22]$$

$$R_2 = C_3H_6 + 4{,}5O_2 \rightarrow 3CO_2 + 3H_2O \qquad [4.23]$$

$$R_3 = CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O \qquad [4.24]$$

$$R_4 = H_2 + \frac{1}{2}O_2 \rightarrow H_2O \qquad [4.25]$$

The kinetic rates suggested by Voltz et al. and Oh and Cavendish are the most appropriate ones for this type of model. They were used making the assumptions that in the catalytic converter, the conditions are:

- Uniform properties at front face of the monolith,
- Heat radiation and conduction in the gas phase are negligible compared to the heat convection,
- Catalytic converter is well insulated,
- Gas temperature and concentrations are identical for all cell channels.

As a result, the enthalpy of the combustion of the species (-ΔHj) is as follows:

$$\Delta H_{CO} = -2{,}832 \times 10^5 kJ/kmol$$

$$\Delta H_{C_3H_6} = -1{,}928 \times 10^6 kJ/kmol$$

$$\Delta H_{CH_4} = -6{,}420 \times 10^5 kJ/kmol$$

$$\Delta H_{H_2} = -2{,}420 \times 10^5 kJ/kmol$$

5. The heat and mass transfer correlations

The heat transfer correlations suggested by Baba et al. to use for the catalyst monolith are:

$$h = \frac{Nuk_g}{d} \qquad [4.26]$$

where

$$Nu = 0{,}571(Re\frac{d}{x})^{2/3} \qquad [4.27]$$

The mass transfer coefficient h_{Dj} in eqs. 4.42 and 4.44 is as follows:

$$h_{Dj} = \frac{ShD_j}{d} \qquad [4.28]$$

$$Sh = 0{,}705(Re\frac{d}{x})^{0{,}43}Sc^{0{,}56} \qquad [4.29]$$

The diffusivity of species j (Dj) used is a Slattery-Bird formula:

$$\frac{pD_j}{(p_{cj})^{\frac{2}{3}}(T_{cj})^{\frac{10}{12}}(\frac{2}{M})^{1/2}} = a(\frac{T}{T_{cj}})^b \qquad [4.30]$$

Where a = 2,745 × 10^{-4} , b = 2,334.

6. Conclusions

A theoretical model for an exhaust system with a catalytic converter was developed, which allows the prediction of the exhaust gas temperature along the exhaust pipe and across the catalyst monolith, both spatially and temporally. These types of models are of the upmost importance because they allow studying the effect of modifications to the exhaust system without the need to carry out expensive and time consuming experiments. The assumptions made do simplify the level of complexity of the subject. Nevertheless, they do not contribute, in our opinion, to a significant deterioration of the predicted values. The model allows us to theoretically experiment the application of insulation materials, location of aftertreatment devices, and exhaust architecture by only modifying and adding elements to the model. The validation was not possible due to time constrains.

7. References:

C.D. Rakopoulos, E.G. Giakoumis, *Diesel engine transient operation – Principles of operation* andsimulation analysis, Springer, London, 2009.

D.B. Spanding, N.H. Afgan, *A numerical study of a interchanging vapour, droplet and film flows in a gasoline engine manifold*, in: B.E. Milton, M. Behnia (eds), *Heat and Mass Transfer in Gasoline and Diesel Engines*, Hemisphere, New York, 1989.

H. Hausen, *Heat transfer in counter flow, parallel flow and cross flow*, McGraw-Hill, New York, 1983

J.G. Collier, *Convective boiling and condensation*, McGraw-Hill, New York, 1981

N. Baba, K. Ohsawa, S. Sugiura, *Numerical approach for improving the conversion characteristics of exhaust catalysts under warming-up conditions*, SAE paper 962076, 1996.

S.E. Voltz, C.R. Morgan, D. Liederman, S.M. Jacob, *Kinetic study of carbon monoxide and propylene oxidation on platinum catalysts, Ind. Eng. Chem. Prod. Res. Dev*., 12, (4), 1973.

S.H. Oh, J.C. Cavendish, *Ind. Eng. Chem. Prod. Res., Dev.* 21, (1982), 29-37.

S.H. Chan, D.L. Hoang, *Heat transfer and chemical reactions in exhaust system of cold-start engine, Int. J. Heat Mass Transfer*, 42, (1999), 4165-4183

S.W. Churchill, H.H.S. Chu, *Correlating equations for laminar and turbulent free convection from a horizontal cylinder, Int. J. Heat Mass Transfer*, 18, (1975), 1049-1053.

V. Gnielinski*, Inst. Chem. Eng*., 16, (1976), 359.

8. Nomenclature

C – Gas concentration by volume (%)

c_P – heat capacity of pipe material ($J.kg^{-1}.K^{-1}$)

c_{Pg} – specific heat of gas at constant pressure ($J.kg^{-1}.K^{-1}$)

D_j – mass diffusivity for gas species j (m^2/s)

d – Hydraulic diameter of monolith cell channel (m)

d_1 – pipe inner diameter (m)

d_2 – pipe outer diameter (m)

d_{bent} – pipe bent diameter (m)

f – Frictional factor

g – Gravitational acceleration (9.81 m/s^2)

h – Heat transfer coefficient ($W.m^{-2}.K^{-1}$)

h_{Dj} – mass transfer coefficient of gas species j ($m.s^{-1}$)

k – Thermal conductivity ($W.m^{-1}.K^{-1}$)

k_j – kinetic rate constant ($mol.K.m^{-2}.s^{-1}$) (j = 1,2,3,4)

K_j – adsorption equilibrium constant (j = 1,2,3,4)

M – Molecular weight of gas (kg/kmol)

$\dot{m}$ – Mass flow rate of gas (kg/s)

Nu – Nusselt number

NTU – number of transfer units

Pr – Prandtl number

Q – Heat transfer (kJ)

q – Heat flux (W/m^2)

R – Reaction rate ($mol.m^{-2}.s^{-1}$)

R – universal gas constant (8.314 $kJ.kmol^{-1}.K^{-1}$)

Ra – Rayleigh number

Re – Reynolds number

S – Geometric surface area per unit catalyst volume (m^{-1})

Sc – Schmidt number

S_{cat} – catalytic surface area per unit catalyst volume (m^{-1})

Sh – Sherwood number

T – Temperature (K)

t – Time (s)

u – Axial gas velocity (m/s)

V_1 – volume of gas (m^3)

V_2 – volume of pipe segment (m^3)

x – Axial coordinate of exhaust system (m)

ΔF – Surface area of monolith segment exposed to heat transfer (m^2)

ΔH – Exothermal heat released (kJ/kmol)

α – Thermal diffusivity (m^2/s)

β – Volumetric thermal expansion coefficient (K^{-1})

μ – Dynamic viscosity ($kg.m^{-1}.s^{-1}$)

ν – Kinematic viscosity (m^2/s)

ρ – Density (kg/m^3)

σ – Stefan-Boltzmann constant (5.67×10^{-8} $W.m^{-2}.K^{-4}$)

δ – Void fraction of the monolith

ε – Surface roughness (m) or emissivity

a – air

c – Critical

cva – convection to ambient

g – Gas

i – Spacing grid

j – Gas species j

gp – convection between exhausted gas and pipe wall

p – pipe

Rad – radiation

s – Catalytic surface

n – time step

in – inlet