

A Sectoral Approach to Modelling Wall Heat Transfer in Exhaust Ports and Manifolds for Turbocharged Gasoline Engines

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ABSTRACT

A new approach is presented to modelling wall heat transfer in the exhaust port and manifold within 1D gas exchange simulation to ensure a precise calculation of thermal exhaust enthalpy. One of the principal characteristics of this approach is the partition of the exhaust process in a blow-down and a push-out phase. In addition to the split in two phases, the exhaust system is divided into several sections to consider changes in heat transfer characteristics downstream the exhaust valves. Principally, the convective heat transfer is described by the characteristic numbers of Nusselt, Reynolds and Prandtl. However, the phase individual correlation coefficients are derived from 3D CFD investigations of the flow in the exhaust system combined with Low-Re turbulence modelling. Furthermore, heat losses on the valve and the seat ring surfaces are considered by an empirical model approach.

Since the comparison between measured and simulated exhaust temperature at turbine inlet serves as an evaluation criterion, a detailed 1D thermocouple model is implemented. Exothermic exhaust after-reactions are represented by a reduced reaction kinetics mechanism. The investigations were carried out for four TC-DI gasoline engines. The low scattering of the correlation coefficients as well as the high agreement between simulated and measured exhaust temperature verify the model quality. Overall, the new sectoral approach shows a significant improvement of wall heat flux calculation in comparison to conventional single-phase approaches from literature.

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INTRODUCTION

The 1D gas exchange simulation is an established CAE tool in the design process of turbocharged gasoline engines. Thereby, the precise calculation of the thermal exhaust enthalpy plays an important part to optimize engines with focus on future challenges like CO₂ - emission limits and real drive emissions combined with the WLTP driving cycle. On the one hand the available level of thermal exhaust enthalpy affects the TC matching process to ensure performance targets; on the other hand the required fuel enrichment for TC protection is typically determined by means of turbine inlet temperature. Against this background a new sectoral 2-phase wall heat transfer model was developed for the exhaust port and manifold to improve result quality of 1D gas exchange simulation.

FUNDAMENTALS

In 3D CFD simulation with Low-Re turbulence modelling, the conservation equations are solved in the boundary layer up to the viscous sublayer. The abstinence of wall function enables a very accurate calculation of heat transfer. Using near-wall cell data the local heat flux is determined by the following equations:

$$\dot{q}_w'' = \frac{\rho c_p u_\tau}{\theta^+} (T_G - T_w) \quad (1)$$

$$u_\tau = \sqrt{\frac{\tau_w}{\rho}} \quad (2)$$

$$\theta^+ = Pr u^+ \quad (3)$$

In 1D simulation there is no detailed information about the boundary layer available. Therefore, a simplified approach is required. The convective wall heat flux is calculated using gas temperature T_G and wall temperature T_w :

$$\dot{q}_w'' = \alpha (T_G - T_w) \quad (4)$$

The determination of the heat transfer coefficient α (HTC) is based on the similitude theory via the Nusselt number which is estimated from approaches like the Colburn analogy [1]



$$Nu_L = c_f \frac{Re_L}{2} Pr^{1/3} \quad (5)$$

or empirical correlations in terms of:

$$Nu_L = C Re_L^a Pr^b \quad (6)$$

Conventional Nusselt correlations for turbulent pipe flows like the one from Gnielinski [2]

$$Nu_D = \frac{(\xi/8) Re Pr}{1 + 12.7\sqrt{\xi/8} (Pr^{2/3} - 1)} \left[1 + \left(\frac{D}{L}\right)^{2/3} \right]$$

$$\xi = (1.8 \log_{10}(Re) - 1.5)^{-2} \quad (7)$$

are not sufficient for describing the heat transfer in exhaust ports and manifolds of combustion engines. The reason for this is the flow profiling which is caused by the high velocity flow around the exhaust valves in combination with the curved port shape. Additionally, a high level of turbulence intensity is generated by the flow around the valves. Both effects lead to a significant enhancement of the heat flux in comparison to conventional turbulent pipe flows.

This correlation was already detected by Zapf in the 60s of the last century and he derived his famous correlation for the HTC in the exhaust port as a function of valve lift and mass flow rate [3].

$$\alpha = 1.785 \left(1 - 0.797 \frac{h_v}{D_{vt}} \right) \dot{m}^{0.5} T^{0.41} D_{port}^{-1.5} \quad (8)$$

Further correlations for heat transfer in the exhaust system were developed by Shayler et al. [4]

$$Nu_D = 0.18 Re_D^{0.7} \quad (9)$$

as well as Depcik and Assanis [5]:

$$Nu_D = 0.07 Re_D^{0.75} \quad (10)$$

Besides, Heller has derived an extended correlation from heat flux measurements which considers the curvature of the exhaust manifold pipes. The approach includes an effective Reynolds number which is calculated from a specially defined effective velocity in order to incorporate the pulsating character of the exhaust flow [6]. In this context, index “n” represents the current time step and “n-1” the previous one.

$$Nu_D = \left(1 + \frac{21 D_{pipe}}{Re_{eff}^{0.14} D_{curvature}} \right) 0.097 Re_{eff}^{3/4} Pr^{1/3}$$

$$u_{eff}^n = \text{Max}\{u_{eff}^{n-1}(1 - 500 \Delta t), u^n\} \quad (11)$$

INVESTIGATED ENGINES AND CORRESPONDING MODELS

The investigations were carried out at four turbocharged gasoline engine with direct injection. The test carriers consist in pairs of engines with conventional and with integrated cooled exhaust manifolds. The technical specifications are summarized in [table 1](#).

The analysis of heat transfer is based on six different operating points; four full load and two part load points, see [table 2](#).

Table 1. Technical Specifications of Engines

	Engine A	Engine B	Engine C	Engine D
Displacement / L	1.0	1.4	2.0	1.2
Bore / mm	73.0	76.0	82.0	77.0
Stroke / mm	78.7	82.6	93.8	85.6
Compression ratio	9.6:1	10.0:1	9.6:1	9.5:1
Number of Valves	4	4	4	4
Exhaust Manifold	integrated	conventional	conventional	integrated

Table 2. Operating Points: Engine Speed (rpm) / BME(MPa) Eng Full Load

Eng	Full Load				Part Load	
A	1500/2.50	2000/2.50	4000/2.50	5500/2.09	2000/0.2	2500/1.0
B	1600/2.20	2000/2.20	4000/2.20	5500/1.70	2000/0.2	2500/1.0
C	1500/1.85	2000/1.85	4000/1.80	5500/1.66	2000/0.2	2500/1.0
D	1500/2.10	2000/2.15	4000/2.20	5500/1.68	2000/0.2	2500/1.0

To measure exhaust temperatures at turbine inlet, the engines are equipped with a thermocouple (type K) with a diameter of 3 mm and a penetration depth of 15 mm. In addition, several surface thermocouples are mounted on the exhaust manifold outer walls to obtain information about the wall temperatures.

3D CFD Model

In addition to the flow guiding surfaces, the CFD exhaust manifold models include the thermocouples at the identical location as the hardware on test bench. The simulations are run with the RANS $k-\omega$ -SST turbulence model by Menter [7] in combination with Low-Re wall treatment. Detailed mesh refinement near the walls as well as in the surroundings of the thermocouples ensures that the condition $y^+ < 1$ is fulfilled for any wall cells. An exemplary section of the polyhedral mesh is represented in [figure 1](#).

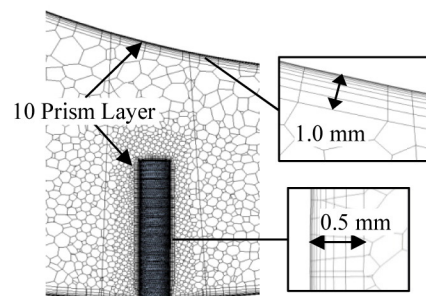


Figure 1. Mesh refinement at the walls and the surroundings of the thermocouple in 3D CFD exhaust manifold model.

A new approach to set realistic inlet boundary conditions was applied by extracting data from in-cylinder flow simulations. For that purpose a section plane was defined downstream the exhaust valves, see [figure 2](#). Profiles of mass flux, total temperature and the turbulent quantities k and ω were written out for a complete cycle as a function of a local coordinate system. This section plane is likewise the inlet of the CFD exhaust manifold model. Due to the identical coordinate system, the profiles can be mapped directly on the cell faces.

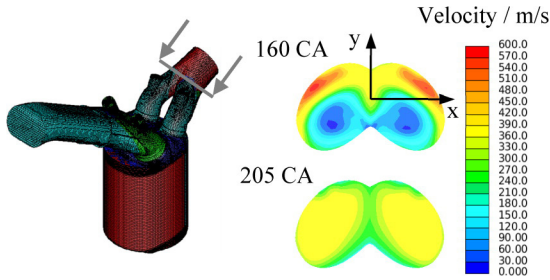


Figure 2. Exemplary location of a section plane downstream the exhaust valves to extract data from a 3D in-cylinder flow simulation

1D Gas Exchange Model

The gas exchange models were calibrated by test bench data with the use of measured quantities like air mass flow rate, injected fuel mass, valve timings, TC speed and several pressure and temperature values. The in-cylinder heat release rates of the six investigated operating points are determined by cylinder pressure analysis from measured indication data.

1D Thermocouple Model

The accordance between measured and calculated exhaust temperature at turbine inlet is used as a criterion to evaluate the quality of the heat transfer model. Therefore, the measurement characteristics of a thermocouple have to be considered in the gas exchange model. The measured temperature of a thermocouple is influenced by thermal inertia, radiation with surrounding walls, heat conduction into the wall and the convective heat transfer from the fluid. The thermocouple temperature T_T can be determined by solving the following differential equation with the variable z for the thermocouple penetration length:

$$\underbrace{\rho \frac{\pi}{4} D^2 c \frac{dT_T}{dt}}_{\text{Inertia}} = \underbrace{\beta_F \bar{\alpha} \pi D (T_T - T_G)}_{\text{Convection}} - \underbrace{\varepsilon \sigma \pi D (T_T^4 - T_W^4)}_{\text{Radiation}} + \underbrace{\lambda \frac{\pi}{4} D^2 \frac{d^2 T_T}{dz^2}}_{\text{Conduction}} \quad (12)$$

The convective term of the equation is extended by the factor β_F :

$$\beta_F(z, t) = \frac{\alpha(z, t)}{\bar{\alpha}(t)} \quad (13)$$

The factor is defined as the relation between the local and the average heat transfer coefficient at a specific position z along the penetration length. In this manner the 3D flow characteristic at sensor position can be also considered in 1D simulation. The average HTC $\bar{\alpha}$ is

typically calculated by a Nusselt number correlation for a circular cylinder in cross flow. For this the authors propose the following correlation, which was derived from previous CFD LES investigations of wall mounted cylinders with low height/diameter ratio in steady-state flow:

$$\overline{Nu} = 0.3813 \overline{Re}^{0.5635} \overline{Pr}^{0.4} \quad (14)$$

Further details regarding the LES investigations as well as the derivation of [equation 14](#) will be published by the authors in future.

The partition of the thermocouple in sections with a length of 1 mm leads to a β_F map for an entire engine cycle as it is shown exemplary in [figure 3](#):

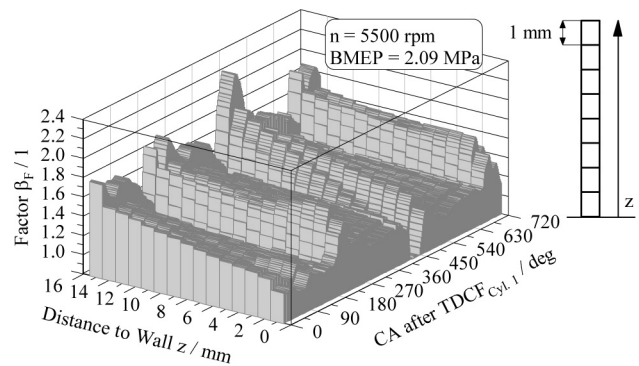


Figure 3. Exemplary map of factor β_F for the rated power operating point of engine A

Previous investigations in steady-state flow on a hot gas test bench have proved that the presented 1D thermocouple model achieves a very high accordance between measured and calculated temperature values ($\Delta T < 4$ K).

Exhaust After-Reactions

The existence of species like unburnt hydrocarbon (HC), CO and H_2 in the exhaust gas enables exothermic after-reactions which increase the thermal exhaust enthalpy. Especially the combination of scavenging due to positive valve overlap and a rich in-cylinder mixture leads to a significant rise of thermocouple temperature at turbine inlet of up to 80 K. The complex reaction kinetics can be modelled precisely by detailed reaction mechanism. However, this results in an unacceptable large simulation duration for 1D gas exchange calculation. Therefore, a reduced iso-octane mechanism by Heufer [8] was implemented in the gas exchange model. In [figure 4](#) there is a comparison between a detailed mechanism by Andrae [9] and the reduced one by Heufer by means of 3D CRFD simulations for two exemplary operating points of engine B.

Despite the slight differences in the traces, there is a good agreement in the integrated released amount of energy. Hence, the reduced mechanism represents a reasonable trade-off between simulation duration and accuracy.