Numerical Analysis of the Potential of a Water-cooled Exhaust Manifold on Diesel- and Gasoline Engines

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ABSTRACT: External or integrated water-cooled exhaust manifolds are mentioned to be favourable for the overall engine behaviour on diesel and gasoline engines. To validate this potential, an analytical study has been conducted. A diesel and a gasoline engine model were built-up in GT-SUITE to assess the influence of water-cooled exhaust manifolds at full load. These engine models have been combined with VeLoDyn – a tool from IAV for vehicle simulation considering heat fluxes between engine and vehicle cooling system – to show the potential of the different designs in a driving cycle, including the catalyst heating up phase.

KEY WORDS: heat engine, engine component or element, cooling, fuel improvement/ fuel additive [A1]

1. INTRODUCTION

Currently, combustion engine improvements focus on the reduction of fuel consumption while maintaining or further improving pollutant emissions in order to respect legislation standards. In the legislative drive cycles, the main part of HC and CO emissions is generated before the catalytic converter (TWC for gasoline, DOC for Diesel) reaches the temperatures for reasonable conversion rates. The main purpose of this study is to reduce catalyst light-off time by positioning the entire exhaust system closer to the cylinder head[1-3].

Following this idea, an issue shows up: For spark ignition engines, the exhaust temperature at full load and nominal running would be too high. Current material with reasonable cost limits the exhaust gas temperature to a maximum of 950°C. A measure as mixture enrichment to keep the exhaust gas temperature below 950°C is limited, and obviously also hurt fuel consumption and emissions. Thereby, the idea came up to cool the exhaust manifold through the engine water circuit. This idea was also applied to a Diesel engine focusing on an altered turbocharger match at rated power expecting improved behaviour in fuel consumption and emissions.

The study will deal with S.I. and Diesel engines to highlight advantages and drawbacks of different solutions for both engine types, well knowing that reducing the exhaust temperature diminishes the quantity of enthalpy getting in the turbine.

The other challenge of this survey is to assess the dynamic response of the vehicle with such a cooling system. Hence, three submodels had to be set-up in order to gain the desired results: the internal engine model, the vehicle model and the cooling system model.

2. SETUP OF SIMULATION TOOLS[4]

2.1. Identification of software tools

For a flexible simulation of the combustion engine in the overall vehicle environment in terms of degree of detail, calculation performance and connectivity a couple of questions concerning model depth and choice of simulation tools have to be answered.

Early stages of engine development demand a high degree of abstraction. The degree of abstraction should individually be chosen for each subsystem depending on the project definition, the available modeling data resources and the calculation time constraint. In that way the optimum simulation quality and efficiency can be reached. For example the combustion engine can be simulated in less than real-time using a 0D-gas exchange calculation combined with a mean-value model based on an artificial neural network for the in-cylinder process. If, as in the here described problem, the energy and heat management is the main scope, a high degree of detail can be chosen here – e.g. for subsystem specific concept or parametric studies – at minimized overall calculation time effort.

Fig. 1 shows the segmentation into subsystems and the appropriate software tool in the chosen tool chain for an overall vehicle simulation. The software tools are linked together via the vehicle simulation software VeLoDyn based on MATLAB/
Simulink which serves as integration platform since most of today’s off-the-shelf tools offer this interface.

Depending on the application, GT-SUITE is used for the engine process, Dymola/Modelica for energy management, cooling system, powertrain and electric system and AxiSuite for exhaust aftertreatment.

With this modeling architecture it is possible to create new submodels, reuse submodels via libraries and to modify them individually to the project specific needs or to integrate models provided by the customer.

2.2. Engine modeling

Engine modeling focuses on two points:

- gas exchange including the turbo/supercharging and EGR system,
- combustion process with energy flow analysis and engine out emission evaluation

Both should be calculated with high accuracy for transient operation – ideally in real-time and three dimensional. However three-dimensional overall engine modeling is still far away from reasonable calculation time for overall vehicle simulation making a reduction of the degree of detail unavoidable. GT-SUITE is used here for engine modeling supplemented by zero-dimensional single- and two-zone combustion and exhaust emission calculation models developed and experimentally validated in-house. Experimental data from engine dynometers delivers the necessary parameters for calibration of the base engine model. The results of the calculation and the component test bench data (i.e. turbocharger, heat exchanger, fuel injector etc.) serve as parameters for the GT-SUITE engine and the AxiSuite exhaust aftertreatment model with heat release curves, exhaust emission data, valve discharge coefficients etc. After the model adjustment and verification, the virtual full-size engine with exhaust aftertreatment model is ready for performing parameter variations or concept studies. Through the repeated interchange and comparison of simulated and measured data the engine simulation quality evolves step by step to a high level concerning precision, reusability and calculation time. On this basis of system understanding the extrapolation of the engine behavior can be extended.

For an overall vehicle simulation vehicle, driver, powertrain and auxiliary systems have to be modeled with adequate precision, too.

2.3. Vehicle and auxiliary system modeling

The graphical user interface shown in the middle of Fig. 1 represents the modular architecture of the integration platform VeLoDyn with its basic submodels and the signal and control bus connections. The main parts are drive train (including engine, clutch and transmission), vehicle, driver and road profile supplemented by vehicle and control models (SoftECU). Within this system each submodel can be replaced by another software tool via its MATLAB interface. Other features are the options to compile a real-time capable vehicle model with MATLAB’s real-time workshop and/or to transfer the vehicle model to the engine test bench for emulating different drive cycles by controlling engine torque and speed with different parameter sets for gear ratios or gear shift strategies in order to optimize fuel consumption and emissions.

The aim of the SoftECU is not to copy the functionality of a real ECU or to display prototype functions, but rather to control all virtual actuators as fuel injectors, spark advance, VTG, EGR valves, intake throttle, VVA etc. during transient simulation in an optimal way, using all the information available from the simulation model. In order to obtain variables of interest as for example oxygen amount and peak pressure in the cylinder, EGR mass flow, or DPF loading the real ECU code has complex observers implemented since this information can not be measured directly with sensors. Since the SoftECU has access to all this information directly, the complexity of the SoftECU is rather small in comparison to the real world ECU or even prototype functions.
3. S.I. ENGINE SIMULATION

3.1. Motivation

The model used for the following investigation was calibrated for a four-cylinder 1.6l turbo charged gasoline engine. As pointed out within the introduction, this part of study is divided in two subsections, steady-state and transient simulation. The steady-state part is conducted at full load to assess the potential of a water cooling system for fuel consumption, whereas the transient simulation is performed at part load driving through the NEDC using all coupled submodels which are described in the simulation tools section. The latter considers the impact of an exhaust cooling system on the catalyst light-off of an S.I. engine.

3.2. Energy distribution in-cylinder

Heat transfer leads engine efficiency and behavior, thereby observing the energy distribution of the engine before any simulation work will provide a serious knowledge of the possible benefits of the study. A simplified first law energy balance is used, giving an insight of the energy distribution into the engine. For a control volume, applied to steady-flow, an application of the first law of thermodynamics is represented by the equation (5):

\[ \dot{m}_f h_f + \dot{m}_a h_a = P_b + \dot{Q}_{misc} + (\dot{m}_f + \dot{m}_a) h_e \]  

(1)

where \( P_b \) is the effective brake power, \( \dot{Q}_{misc} \) is the heat-transfer rate to the oil and the cooling medium including convection and radiation from the engine’s external surface. Of course, a complete energy balance of an engine is much more complex, but the Equation (1) is sufficient here in order to obtain a first estimation of the energy distribution. However, to understand the heat fluxes through the engine, the friction losses should be described within the energy balance. The total friction work \( W_f \) in an engine is represented by the mechanical friction and the fluid friction resulting of the following effects: rubbing friction work of the mobile coupling system \( W_{gf} \), accessory work \( W_a \) and pumping work \( W_p \) (5):

\[ W_f = W_{gf} + W_a + W_p \]  

(2)

Therefore, two more terms are added to Equation (1), the mechanical friction power \( P_f \) and the pumping power \( P_p \), both before included in the heat-transfer \( \dot{Q}_{misc} \), thus the energy distribution becomes:

\[ \dot{m}_f h_f + \dot{m}_a h_a - \dot{m}_e h_e = P_b + P_f + P_p + \dot{Q}_{ht} \]  

(3)

where \( \dot{Q}_{ht} \) is the heat-transfer rate to the oil and the cooling medium including convection and radiation from the engine’s external surface.

Then, the Equation (3) is applied to the engine model in Fig. 2 without any additional cooling system – at full load throughout each engine speed showing the potential of a solution enabling the use of the entire fuel energy.

![Engine energy balance with a standard exhaust manifold](Image)

Fig. 2 Engine energy balance with a standard exhaust manifold

The fuel power is the power produced by the fuel whether the entire efficiency was 100%. Instead of an additional fuel mass injected for cooling purposes, the exhaust flow would be cooled through the exhaust cooling system in order to minimise the unburnt excess fuel at full load.

3.3. Steady-state investigation at full load

The modeling is based on measurements conducted with a standard and compact exhaust manifold. To keep a comparison criteria, for every simulation run, the power output is kept constant at each engine speed.

First, the exhaust manifold is built-up in GT-SUITE defining material properties (steel) and an external medium (ambient air or water circuit), afterwards the exhaust manifold is connected to the entire engine model and to the cooling circuit model in case of water-cooling. It is assumed that an exhaust manifold entirely cooled throughout its geometry would provide more surface in contact with water than an integrated exhaust manifold into the cylinder head – as well cooled. Therefore, the surface in contact with water is varied and reduced to simulate the effect of an integrated exhaust manifold. As there is no real layout or no CFD calculation existing, the surface in contact with the cooling medium is varied to 50% and 70% (surface remaining in contact with water) to assess the effect of an integrated manifold concept, which would be shorter than an external exhaust manifold.

As indicated in Fig. 3, the global cooling power (depending on the heat exchanger geometry and efficiency) is limited to remain realistic in order to still fit within a passenger car front...
end. This limit is set – as far as possible – to 30% maximum above the reference value of the model running with a standard exhaust manifold at nominal power. If this limit is reached or exceeded, as for the entire water-cooled exhaust manifold due to a high heat transfer rate, an additional cooling is necessary by lambda enrichment. Unfortunately, in case of a complete water-cooled exhaust manifold, lambda has to be seriously reduced to a minimum value of 0.75 approaching the ignition limit (curve with cross symbols in Fig. 3).

Likewise, in Fig. 4 the exhaust temperature upstream the turbine is controlled – through the lambda ratio in Fig. 5 – to a maximum value of 950°C (standard value for material resistance), whereas the exhaust temperature of the air-cooled manifold is controlled according to the measurements (close to 950°C). Actually, if a water cooling system is employed, the maximum exhaust gas temperature permitted is not achieved because of the water cooling.

A distinguished impact of the water-cooling on the lambda enrichment is visible in Fig. 5 and Fig. 6, giving the real potential of an integrated exhaust manifold. A complete water-cooled exhaust manifold would need too much cooling power to enhance the fuel consumption whereas an integrated manifold can bring a slight advantage.

Basically, a higher vehicle cooling system power would provide more fuel saving avoiding a severe enrichment. However, a water-cooling system is worthwhile as long as the absorbed heat from the exhaust manifold can be handled by the vehicle heat exchanger or by adapting the contact area of the exhaust manifold. Thereafter in Fig. 7, if the water-cooling is restricted integrating the exhaust manifold into the cylinder head, a significant benefit on the fuel consumption is noticeable by restrictive cooling power usage.

3.4. Transient investigation

As described in the simulation tools section, the engine model is coupled with a vehicle model within VeLoDyn. The main focus of the transient simulation is to figure out the parameters impacting the catalyst light-off. Four configurations
are presented dealing with exhaust manifold length (long or short) and water-cooling (uncooled, at 70% water-cooled and 100% water-cooled). The long exhaust manifold is 30% longer than the short exhaust manifold, which was used for the steady-state simulation. No obvious difference on the catalyst light-off is withdrawn of Fig. 8, but after zooming on the first hundred seconds in Fig. 9, the length of the exhaust manifold has a significant impact on the catalyst light-off on the contrary of the water-cooling. Indeed, with a short exhaust manifold – cooled or uncooled – the catalyst light-off is reached in a time between 20 and 30 seconds while a long exhaust manifold needs twice as long. The water-cooling does not impact the heat increase of the catalyst – through the exhaust temperature in Fig. 8 and Fig. 9 – although the water gasket isolates the exhaust manifold instead of producing a higher heat transfer through the material. It was expected that this isolation would improve the light-off keeping a higher quantity of enthalpy going through the catalyst converter, however as explained before the position of the catalyst converter (depending on the exhaust manifold geometry) has much more impact on the light-off than the water-cooling. This is the reason why the variant with 50% cooling was not investigated.

4. DIESEL SIMULATION

4.1. Motivation

Increased heat withdrawal between exhaust valve and turbine inlet enables higher temperatures at the exhaust valve while keeping the inlet turbine temperature constant. This new degree of freedom opens two main options for the Diesel engine:

1. Increase of rated power with unchanged turbo system which would represent a downsizing measure.

2. Maintaining rated power and optimize the turbo system to the new conditions to improve part load behavior of the engine.

In the following, the latter way is examined.

A smaller turbo system promises higher low end torque resulting in better drivability. Additionally a higher boost pressure under part load conditions allows a higher EGR-rate which results in lower NOx emissions. In order to examine the NOx reduction potential of the cooled exhaust manifold the following steps were performed: First compressor and turbine size were adapted to the new conditions at rated power. In a second step the newly sized turbo system will be examined in the NEDC with the help of transient simulation. For comparability reasons, the EGR rate was increased until the fuel efficiency in the NEDC was matched with the baseline setup.

4.2. Steady state turbocharger assessment for rated power

The basic idea for the application of the water-cooled exhaust manifold is the reduction of the air/fuel ratio and with that the fresh air mass flow at rated power without running into exhaust temperature limitations. Decreasing the air/fuel ratio on the other hand causes increased particulate emissions. This limitation will also be considered in the following assessment.

Here, the analytically examined engine is an inline, single stage turbo charged 2.0l common rail engine with a rated power of 105kW. Since the aim is to keep the rated power constant, the open question is, how exhaust gas temperature and particulate emissions change with decreased boost, fresh air mass flow, and air/fuel ratio (Table 1).

According to Fig. 10, for the case of the theoretical minimum air/fuel ratio of $\lambda=1.1$, the exhaust gas temperature would rise from 730°C to 850°C without cooling. The smoke number would increase from 1.0 FSN to 5.5 FSN without changes of the combustion system. This is not necessarily a problem, since today’s Diesel engines are all equipped with a Diesel particulate filter (DPF). A rough estimation of the balance point (equilibrium of soot input and soot oxidation) of the DPF with the higher soot load reveals that a smoke number of 5.5 FSN
leads to an DPF overload despite of thermal soot oxidation in the examined particular operating point.

As Fig. 11 reports, the combustion system needs to be improved in a way, that with the fixed boundary conditions of the DPF in that example \( V = 4l, cpsi = 300, T_{\text{DPFin}} = 600^\circ \text{C} \), the smoke number would decrease below 4.0 FSN at \( \lambda = 1.1 \). This could be achieved by the typical measures of increasing rail pressure and/or optimizing nozzle and combustion chamber geometry.

With the fixed minimum air/fuel ratio at \( \lambda = 1.1 \) the turbocharger system can be matched to the new conditions at rated power. For the new desired air/fuel ratio the boost pressure and with that the fresh air mass flow can be reduced by 27% . Since compressor inlet temperature and pressure are not affected by the exhaust manifold changes, the actual physical mass flow and the reduced mass flow are, according to Equation 4, diminished by the same factor.

\[
\dot{m}_{\text{C,red}} = \dot{m}_{\text{C,phys}} \sqrt{\frac{p_i}{T_i}} \Rightarrow \dot{m}_{\text{C,red}} = \dot{m}_{\text{C,phys}} (4)
\]

Fig. 12 depicts the change of the compressor operating point in the flow rate and efficiency map, which moves, as expected, towards smaller mass flows along the engine air consumption line.

\[
\text{Fig. 11 Assessment of DPF balance point for rich Diesel combustion at rated power}
\]

\[
\text{Fig. 12 Shift of the compressor operating point}
\]

Things look more difficult on the turbine side. According to the turbocharger main Equation (5), the pressure ratio of the turbine decreases from 2.2 for the base line setup to 1.56 for the reduced boost pressure. The reduced mass flow in the turbine map is effected by the changed turbine inlet conditions.

\[
\frac{p_2}{p_1} = \frac{1}{\left(\frac{\dot{m}_T}{\dot{m}_V}\right) c_{\text{V,T}} T_i \eta_{\text{ATL}}} \left(1 - \left(\frac{p_4}{p_3}\right)^\frac{\gamma - 1}{\gamma} \right)^{\frac{\gamma - 1}{\gamma} \frac{\gamma}{\gamma - 1}}
\]

\[
\frac{p_1}{p_i} = 1 - \left(\frac{\dot{m}_T}{\dot{m}_V}\right) c_{\text{V,T}} T_i \eta_{\text{ATL}}
\]

(5)
With the now known pressure ratio the reduced mass flow, and with that the turbine operating point in the turbine map can be calculated.

\[ \dot{m}_{T,\text{red}} = \dot{m}_{T,\text{phys}} \frac{\sqrt{T_3}}{P_3} \]  

(6)

According to Eq. (6) the reduced mass flow of the base line setup amounts to \( \dot{m}_{T,\text{red}} = 57.3 \) kg/s/kPa while the reduced mass flow with the cooled exhaust manifold is with \( \dot{m}_{T,\text{red}} = 60.9 \) kg/s/kPa higher. This leads to the surprising conclusion that the cooled exhaust manifold is not a measure for the size reduction of the turbocharger system with the given boundary condition of equal exhaust gas temperatures, since the reduction of the exhaust gas pressure overcompensates the reduction of the physical mass flow through the turbine, as shown in Fig. 13. Despite of the reported effects, there seems to be still potential for part load operation improvement through the option of decreasing the compressor size. On the turbine size the turbine either has to be increased or the VTG turbine will be decreased but supplemented by an additional waste-gate for higher mass flows. Both options are investigated in the next chapter.

\[ \text{Fig. 13 Shift of the turbine operating point} \]

4.3. Transient investigation

Since the cooled exhaust manifold leads to the need of resizing compressor and turbine, this will be quantized here for the options with and without waste gate. On the compressor side a map was picked with roughly 15% less flow rate. Fig. 14 compares the operating points of the full load curve for the base line configuration with the location of the operation point for rated power for the smaller compressor with and without waste gate on the turbine side. As estimated in the previous chapter, the boost pressure is diminished with the new decreased air/fuel ratio from 2.5 bar down to 1.75 bar. With that the necessary compressor power reduces naturally. The chosen compressor shows also a reasonable good efficiency for rated power and has plenty altitude reserve. Full load operating points with air/fuel ratio for smaller engine speeds show sufficient distance to the surge line.

\[ \text{Fig. 14 Compressor match at rated power} \]

Things are again a little more complicated with the turbine. For comparability reasons the turbine efficiency map was not changed, while the mass flow map was scaled with a constant factor. For the case of cooled manifold without added waste gate the mass flow map of the turbine had to be increased by 30% in order to achieve the same turbine efficiency with the same VTG position at rated power (see Fig. 15). In case of an additional waste gate the mass flow through the turbine is considerably (20%) smaller in comparison to the base line turbine. However, the same compressor pressure ratio could be achieved, because of the higher turbine pressure ratio.

The newly matched compressor and turbine maps were now used for the simulation of part load conditions in the NEDC. For demonstrating positive effects on pollutant emissions, part load points were recalibrated in terms of raising the boost pressure and EGR set point until the BSFC values of the base engine were reached. The CO2 neutral increase of EGR rate could basically be achieved through better efficiencies of the compressor for both cases and also better efficiencies on the turbine side with the added waste gate in part load operation. Additionally a small contribution of fuel savings between 1% and 2% in the NEDC was simulated through reduced friction losses. The heat energy put into the coolant water, accelerated the heat-up process of the lubricant. The increased EGR rate leads to a decreased NOx pollution. Fig. 16 reports 2% NOx reduction for the bigger turbine and a potential of 22% NOx reduction for the smaller turbine with waste gate.
As for the gasoline engine the additional heat input of the water-cooled exhaust manifold into the cooling system has to be considered and will lead to the need of resizing the radiators. The impact of the water-cooled exhaust manifold to the heat-up behavior of the Diesel exhaust gas aftertreatment system was not part of the investigation. During the warm-up phase in any cycle the higher heat capacity of the water-cooled exhaust manifold will have temperature disadvantages for DOC and, if applicable, for NOx aftertreatment devices as LNT or SCR.

**REFERENCES**


**DEFINITIONS, ACRONYMS, ABBREVIATIONS**

**NEDC**
New European Driving Cycle

**VeLoDyn**
Vehicle Longitudinal Dynamics
VTG
Variable Turbine Geometry

$m_f$
Fuel mass flow

$h_f$
Fuel enthalpy

$m_a$
Air mass flow

$h_a$
Air enthalpy

$h_e$
Exhaust gas enthalpy