EXHAUST MANIFOLD PRESSURE ESTIMATION ON A TURBOCHARGED SI-ENGINE WITH WASTEGATE

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Abstract: In turbocharged engines with wastegate the exhaust pressure can change rapidly. A method to estimate the exhaust manifold pressure is presented for diagnosis of wastegate and turbocharger on spark-ignited engines. It does not require any extra sensors in the exhaust system after the calibration. A non-linear model is developed of the exhaust pressure. Estimates of the exhaust manifold pressure relies on information from an air-to-cylinder observer and a static map of stationary exhaust pressure. The exhaust manifold pressure estimator is validated using a series of wastegate steps on a turbocharged SAAB 2.3 dm^3 SI-engine. The exhaust pressure estimation is designed for steady-state conditions and the validation shows that it works well and converges within 1 to 4 seconds.

Keywords: engine modeling, estimator, sensor fusion

1. INTRODUCTION

Knowledge of the exhaust manifold pressure on a turbocharged SI-engine with wastegate is useful for diagnosis of the wastegate, the turbine, and the exhaust system. The wastegate controls the power to the turbine and prevents engine and turbocharger from destruction by reducing the pressure in the exhaust manifold. Therefore it is crucial for turbine safety to diagnose the wastegate. One method to diagnose wastegate operation is to use the exhaust manifold pressure. This is not normally measured due to the high temperatures in the exhaust system and the extra cost of an additional sensor. Observers are therefore desirable for the exhaust manifold pressure. For naturally aspirated (NA) SI-engines observers for pressure and temperature in the exhaust manifold have been proposed in (Maloney and Olin, 1998) with good results. In NA engines the exhaust pressure is generated by the exhaust system which acts as a simple flow restriction. On turbocharged engines with wastegate this restriction consists of three parts. The exhaust system which acts as a restriction and produces back pressure. The turbine which also acts as a restriction and finally the wastegate which shunts varying amounts of the exhaust gases past the restricting turbine. Changes in valve position therefore influences the flow restriction and the exhaust side of the engine can not be modeled as a constant flow restriction. Unfortunately the position of the wastegate is not measurable, which further complicates the situation.

To estimate the absolute exhaust manifold pressure the information from a mean value air-to-cylinder estimator (Andersson and Eriksson, 2001) is used together with an additional map of the stationary exhaust pressure. The estimated exhaust pressure can then for example be used for diagnosis of the wastegate, diagnosis of the turbine, or checking the back-pressure caused by the exhaust system. The developed estimator is nonlinear and model-based. No additional sensors in the exhaust system are needed by the estimator after calibration. The only sensors used are air mass flow, pressure and temperature after the throttle, which are available on many production engines. The

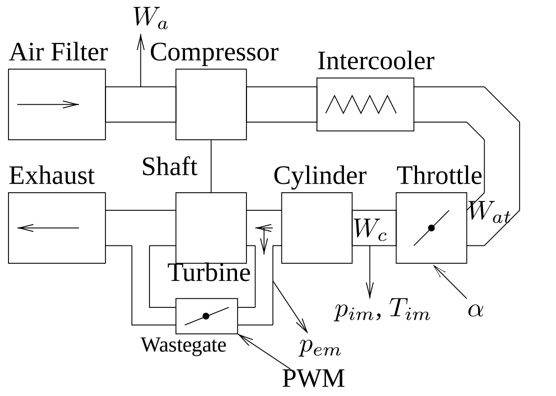

Fig. 1. Sensors and actuators on the engine. The only measured air mass flow is before the compressor, W_a.

estimated exhaust pressure is only valid under steady-state conditions since the air-to-cylinder have to settle and a simplified static intake manifold model used.

1.1 *System Overview*

In Figure 1 the components of the engine and the sensors is shown. The air flows through the air-filter and is then measured by a hot-film air mass sensor W_a. It is then compressed and cooled by the intercooler. The air flow into the intake manifold is restricted by the throttle which is operated by setting the angle of the throttle plate α. Air mass flow past the throttle and into the intake manifold is W_{at}. In the intake manifold there is one pressure sensor p_{im}, and one temperature sensor T_{im}.

From the intake manifold the air mass flow to the cylinders is W_c and it can only be measured stationary by W_a. The mass of air that can fill the cylinder depends on, among others, the amount of residual gases in the cylinder. The later is governed by the exhaust manifold pressure p_{em}, which in turn depends on the wastegate position. A closed wastegate increases the exhaust manifold pressure, and results in more residual gases and a smaller mass of air can fill the cylinder.

The wastegate is controlled by a pulse width modulated (PWM) signal.

2. MODELING

Air mass to cylinder is influenced by the exhaust pressure and a model that describes the air mass to cylinder flow is therefore first described. Later the exhaust manifold pressure variations due to wastegate operation are modeled using the air to cylinder information. Finally a brief summary of the exhaust manifold pressure estimation is given.

Here the intake manifold pressure dynamics is neglected when the exhaust manifold pressure is estimated. For a description of the symbols used, please see the nomenclature at the end.

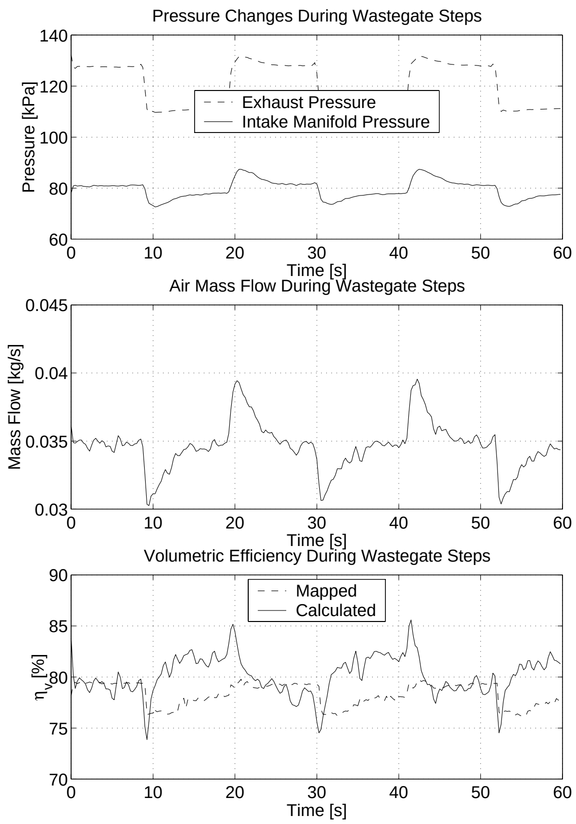

Fig. 2. *Top:* When the wastegate is opened the exhaust pressure drops but the air mass flow is maintained constant by a controller (*center*). *Bottom:* At stationary conditions with closed wastegate (time 8, 28, and 50) the mapped volumetric efficiency agrees with the current volumetric efficiency $\eta_{\text{vol}} = \frac{W_{at} R_c T_{im} n_r}{p_{im} V_d N}$. When the wastegate is open (time 17, 30, and 60) the volumetric efficiency does not match the mapped value.

2.1 *Air-to-cylinder Model*

A standard method to model air to cylinder flow is to map the volumetric efficiency of the engine under stationary conditions (Heywood, 1988; Taylor, 1994). In the turbocharged engine the exhaust manifold pressure varies with the setting of the wastegate which affects the volumetric efficiency since it is a function of the pressure ratio $\frac{p_{em}}{p_{im}}$. This is supported by measurements, see Figure 2.

Changes in exhaust manifold pressure therefore influences the air mass that can enter the cylinder. Volumetric efficiency estimates the air mass to cylinder well if the exhaust manifold pressure is the same as during the engine mapping. However if the exhaust pressure is not the same as during the mapping there will be an offset, called m_Δ, in estimated air compared to the actual air mass to the cylinder per combustion. Stationary this offset is

$$m_\Delta = \eta_{\text{vol}}\left(N, p_{im}\right) \frac{p_{im} V_d}{R_c T_{im}} - W_{at} \frac{n_r}{N} \tag{1}$$

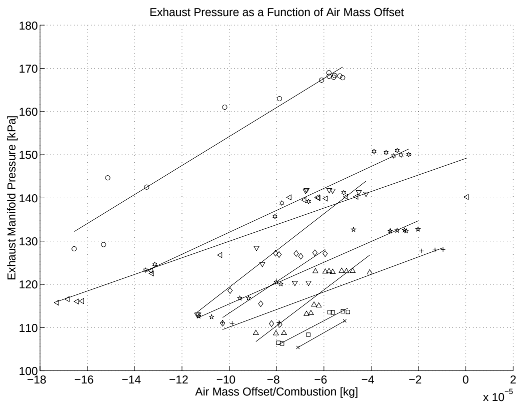

Fig. 3. Measured exhaust pressure with different wastegate settings when the engine was running at stationary conditions. The exhaust pressure is linear as a function of the air mass offset m. The slope of the fitted first order function varies slightly with engine operating point.

During intake manifold pressure transients the estimated m_Δ is not valid due to the change of mass inside the intake manifold. The air mass offset m_Δ is estimated by the estimator (Andersson and Eriksson, 2001) and can be used to produce an estimate of the change in exhaust manifold pressure, p_{em_Δ}, compared to the pressure conditions during the mapping of volumetric efficiency.

2.2 *Exhaust Pressure Model*

In Figure 3 the exhaust pressure is plotted under stationary conditions as a function of air mass offset given by Equation (1). Exhaust manifold pressure is nearly linear with the air mass offset. The developed exhaust pressure model is motivated by considering a simplified process for the gas exchange.

During the gas exchange, fresh gases are mixed with residual gases. If heat transfer is neglected, the internal energy of the mixture is conserved according to the first law of thermodynamics. A standard assumption (Heywood, 1988) is to assume constant specific heat c_v for unburned and burned mixture. In this case the burned mixture is studied at exhaust valve closing when it has expanded to the pressure in the exhaust manifold. Therefore c_v and the molecular weight will differ approximately 10% between burned and unburned mixture (Heywood, 1988). With these assumptions the gas constant R_c is also regarded as constant.

$$mc_vT_1 = m_{af}c_vT_{af} + m_rc_vT_r \tag{2}$$

If the gases are assumed to be ideal the total in cylinder mass m and the residual gas mass m_r can be obtained.

$$m = \frac{p_c\left(V_c + V_d\right)}{R_cT_1} \tag{3}$$

$$m_r = \frac{p_{em}V_r}{R_cT_r} \tag{4}$$

Since no heat transfer is assumed T_{af} in Equation (2) and the temperature in the intake manifold is measured it is used instead, $T_{af} = T_{im}$. When the in cylinder mass is calculated in Equation (3) the pressure at intake valve closing is needed. The modeled engine is not equipped with a tuned intake system therefore the intake manifold pressure is used instead, $p_c = p_{im}$. Exhaust manifold pressure can be calculated by inserting Equations (3, 4) into Equation (2), given the volume of the residual gases V_r and their temperature T_r.

$$m_{af} = m_a + m_f = m_a \underbrace{\left(1 + \frac{1}{\lambda\left(\frac{A}{F}\right)_s}\right)}_{k}$$

$$\begin{aligned} p_{em} &= \frac{V_c + V_d}{V_r}p_{im} - \frac{R_c}{V_r}m_{af}T_{im} = \\ &\underbrace{\frac{V_c + V_d}{V_r}\cdot p_{im} - \eta_{\text{vol}}p_{im}V_dk}_{p_{em_{\text{map}}}(N,p_{im})} - \underbrace{\frac{R_c}{V_r}m_\Delta T_{im}k}_{p_{em_\Delta}(m_\Delta,R_c(\lambda),T_{im},k)} \end{aligned} \tag{5}$$

The important second order effects, such as heat transfer, and valve overlap etc. are taken into account by maps. In Equation (5) $k = 1 + \frac{1}{\lambda\left(\frac{A}{F}\right)_s}$, and $\frac{V_c+V_d}{V_r}$ are constant since the engine does not have variable valve timing. In $p_{em_{\text{map}}}\left(N, p_{im}\right)$ the second term

$$\eta_{\text{vol}}p_{im}V_d \underbrace{\left(1 + \frac{1}{\lambda\left(\frac{A}{F}\right)_s}\right)}_{k}$$

can be regarded as independent of λ. Volumetric efficiency is quasi-statically a product where one factor (Heywood, 1988) is

$$\frac{1}{1 + \frac{1}{\lambda\left(\frac{A}{F}\right)_s}} = \frac{1}{k}$$

which cancels the dependency of the air/fuel ratio. The static part of the exhaust manifold pressure, $p_{em_{\text{map}}}\left(N, p_{im}\right)$, is determined during engine mapping. Gas constant R_c is also a function of air/fuel ratio.The amount of residual gases m_r is small compared to the mass of air and fuel m_{af} and therefore the gas constant for the unburned mixture is used and the impact of residual gases are neglected. Air/fuel ratio influences the gas constant in the following way

$$\begin{aligned} R_c\left(\lambda\right) &= \frac{\tilde{R}_c}{\mathcal{M}} = \frac{\tilde{R}_c}{\frac{m_f+m_a}{n_f+n_a}} = \\ &\frac{\tilde{R}_c\left(1 + \lambda\left(\frac{A}{F}\right)_s\right)}{\mathcal{M}_f + \lambda\mathcal{M}_a\left(\frac{A}{F}\right)_s} \end{aligned} \tag{6}$$

Isoocatane was used to approximate the fuel data in Equation (6) and $R_c\left(\lambda\right)$ was inserted into Equation (5). If there is no sensor data available of the air/fuel ratio it can be approximated using the relationship

$$\lambda = \frac{W_{at}}{\left(\frac{A}{F}\right)_s \dot{m}_f}$$

Fuel mass flow is calculated using the injector approximation:

$$\dot{m}_f = \frac{N}{n_r} n_{\text{cyl}} K_{\text{inj}} \left(t_{\text{inj}} - t_0\right)$$

The injector pulsewidth t_{inj} is available in most ECUs.

The resulting exhaust manifold pressure model has only one parameter and that is the volume of the residual gases V_r it was estimated using a least-square method on measured engine data.

2.3 *Summary of Exhaust Pressure Calculation Process*

First the air mass offset m_Δ, Equation (7b), is calculated. The air fuel ratio dependency is captured by Equation (7c) and is inserted into the final equation 7d.

$$R_c(\lambda) = \frac{\tilde{R}_c \left(1 + \lambda \left(\frac{A}{F}\right)_s\right)}{\mathcal{M}_{fuel} + \lambda \mathcal{M}_{air} \left(\frac{A}{F}\right)_s} \tag{7a}$$

$$m_\Delta = \eta_{\text{vol}}(N, p_{im}) \frac{p_{im} V_d}{R_c(\lambda) T_{im}} - W_{at} \frac{n_r}{N} \tag{7b}$$

$$k = 1 + \frac{1}{\lambda \left(\frac{A}{F}\right)_s} \tag{7c}$$

$$p_{em} = p_{em_{\text{map}}}(N, p_{im}) - \frac{R_c(\lambda)}{V_r} m_\Delta T_{im} k \tag{7d}$$

3. MEASUREMENT SETUP

The measurements were performed on a 2.3 dm^3 turbocharged SAAB spark ignition engine with wastegate and drive-by-wire system. The engine is connected to an asynchronous Dynas 220 NT dynamometer, which is operated at constant speed mode. The dynamometer is controlled by a PC and the engine is controlled by a research engine management system called Trionic 7. The engine management unit was connected to a PC in the control room using a CAN-bus. From the control room it is possible to control the throttle and the wastegate. The later was also manually operated with a handle.

The engine is equipped with additional pressure sensors Kristall 4293A2 and Kristall 4293A5 before the throttle, in the intake manifold, and in the exhaust manifold before the turbine. There are also extra temperature sensors of PT200 type, Heraeus ECO-TS200s, in the intake manifold and between the intercooler and the throttle and in the exhaust manifold close to the turbine.

All measurements were performed with an HPE 1415A, which is a VXI-instrument. Engine mapping was performed with a sampling frequency of 10 Hz and the signals were low-pass filtered at 5 Hz to avoid aliasing. The engine mapping was performed from 1000 RPM up to 4800 RPM in steps of approximately 500 RPM. The lower limit was due to severe vibrations at higher loads. The engine was run 25 seconds in each work point before a 5 second sampling was started. The median of the sampled data was then stored in 114 points.

Step response experiments where performed with the same instrument HPE 1415A and a sampling frequency of 1 kHz was used. Anti-alias filters were disabled due to the damping and delay introduced by the filter.

4. VALIDATION OF ESTIMATOR

In the validation process measured engine data was used. The test case will be described thoroughly in the next section. The exhaust pressure is estimated using the static map of the exhaust manifold pressure and air mass offset information m_Δ.

The estimator is validated using measurements of the exhaust pressure while wastegate valve was manually operated. In the engine management system, a controller tried to maintain constant air mass flow through the throttle. Since the power to the compressor is reduced when the wastegate is opened the throttle controller will open the throttle to compensate for the lowered air mass flow. Throttle angle will therefore not be constant during the test, which influences the air mass flow through the throttle and the air dynamics introduces a small deviation in the estimated air mass offset m_Δ until the system has settled.

4.1 *Estimated Exhaust Pressure*

Measurements have been taken for a number of engine speeds between 1600 and 3100 RPM. In each measurement the engine speed was held constant and the wastegate was initially controlled by the ECU. The wastegate was opened and held constant for approximately 10 seconds and then closed. In Figure 4 the results of the estimation can be seen with the use of a map and the combined map and air mass offset information. The fit is within approximately 6 % for the estimated absolute exhaust manifold pressure.

To show the dynamic behavior of the exhaust manifold pressure estimator two operating points was chosen, one at low engine speed and load and one at higher engine speed and higher load. To reduce noise the signals used in the computation of Equation (5) have been low pass filtered and so have the measured exhaust manifold pressure been to reduce engine pumping fluctuations. The low speed and load case is shown in Figure 5 and the higher speed and load is shown in Figure 6 where different settings of the wastegate was used. It takes a few seconds for the estimated exhaust manifold pressure to converge since stationary conditions have been assumed to calculated the air mass offset m_Δ.

The delay in Figure 5 and Figure 6 is caused by the low pass filtering of e.g. the air mass flow signal.

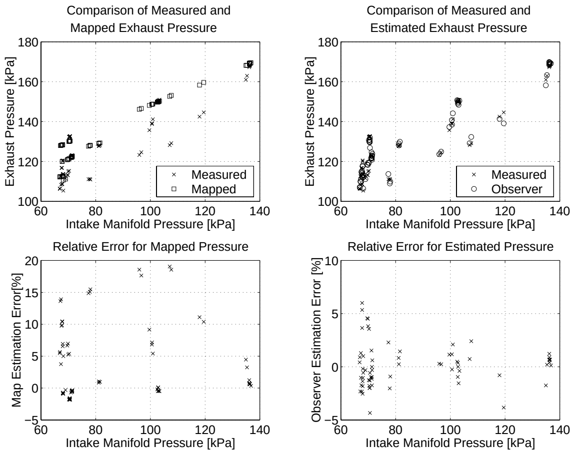

Fig. 4. *Left column:* When the static map of the exhaust pressure is used the exhaust pressure is overestimated as the wastegate is opened. *Right column:* The air mass offset information decreases the estimation error when the wastegate is opened.

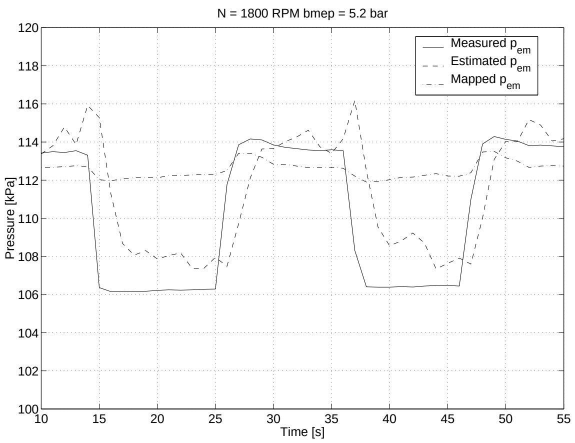

Fig. 5. Estimated exhaust manifold pressure during a wastegate step at low engine speed and load. Steady-state performance is within a few percent.

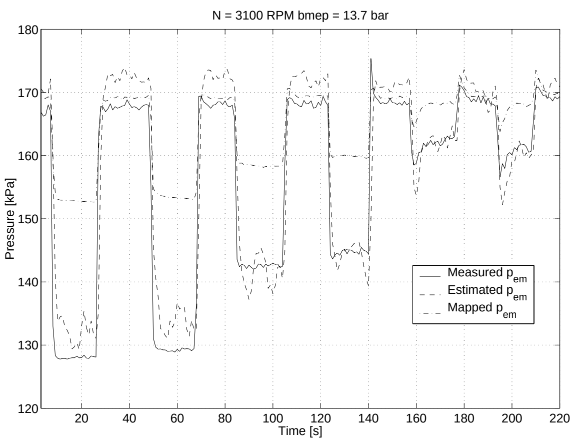

Fig. 6. Estimated exhaust manifold pressure during a series of wastegate steps at medium engine speed and higher load.

Transients in estimated exhaust pressure is due to intake manifold filling/emptying where the estimated in cylinder air mass offset m_Δ is not correct.

4.2 *Parameter Sensitivity*

If the absolute exhaust pressure is estimated there will obviously be an offset if the mapped pressure is incorrect. Since the estimated exhaust manifold pressure is a sum of a mapped value and an estimated offset p_{em_Δ}. The later depends on m_Δ which is calculated in Equation (1), measured intake manifold temperature, air/fuel ratio, and estimated volume of the residual gases V_r. The m_Δ is affected by errors in the measured air mass flow into the manifold W_{at}, p_{im}, and volumetric efficiency η_{vol}.

Residual gas volume V_r is assumed to be constant but in reality it is not since there are dynamic effects in the gas exchange such as the inertia of the gases.

5. CONCLUSIONS

On turbocharged spark-ignition engines with wastegate the exhaust pressure can not be estimated using a static map of N and p_{im}. This since the wastegate valve can change position during normal operation. The non-linear model based air-to-cylinder observer estimates the in cylinder mass offset m_Δ and together with the static map of the exhaust manifold pressure a better estimate of the exhaust manifold pressure can be made. The proposed model based estimator captures well the changes in exhaust pressure when the wastegate is opened and closed. Which can be used in the diagnosis system, e.g. to check the back pressure caused by the exhaust system.

ACKNOWLEDGMENTS

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6. REFERENCES

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NOMENCLATURE

Symbol	**Description**
p_{im}	Intake manifold pressure
p_{em}	Exhaust manifold pressure
p_c	In cylinder pressure at intake valve closing
T_{im}	Intake manifold temperature
T_1	Temperature of charge (air, fuel, and residual gases) at start of compression
T_{af}	Temperature of air/fuel charge at start of compression
T_r	Temperature of residual gases
η_{vol}	Volumetric efficiency
α	Throttle angle
W_a	Measured air mass flow
W_{at}	Air mass flow through throttle
W_c	Air mass flow to cylinder
m	In cylinder mass at inlet valve closing
m_f	Mass of fuel the cylinder
m_a	Mass of air the cylinder
m_{af}	Mass of air and fuel in the cylinder
$m_\triangle$	Air mass to cylinder offset, calculated using mapped volumetric efficiency
m_r	Residual gas mass
$\mathcal{M}_c$	Molecular weight of mixture in the cylinder at inlet valve closing
$\mathcal{M}_a$	Molecular weight of air
$\mathcal{M}_f$	Molecular weight of fuel
n_a	Number of moles of air
n_f	Number of moles of fuel
c_v	Specific heat at constant volume
γ	Ratio of specific heats
R_c	Specific in cylinder gas constant at intake valve closing
$\tilde{R}$	Gas constant, 8.31 $[\frac{J}{\text{mole}\cdot K}]$
r_c	Compression ratio of the engine
λ	Normalized air/fuel ratio
$\left(\frac{A}{F}\right)_s$	Stoichiometric air/fuel ratio
k	Scaling factor to calculate air and fuel mass given air mass, $k = 1 + \frac{1}{\lambda\left(\frac{A}{F}\right)_s}$
V_d	Displacement volume
V_c	Clearance volume
V_r	Volume of residual gases
n_r	Number of revolutions per cycle
N	Engine speed i revolutions per second
t_{inj}	Time in seconds where the injector is open
K_{inj}	Maximal delivered fuel mass per second
t_0	Time in seconds for the injector needle lift
t_{inj}	Engine speed i revolutions per second