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On–line identification of fuel dynamics for a model-based injection control

Roberto Cipollone and Carlo Villante
University of L'Aquila, Italy

M. Sughayyer
Palestine Polytechnic University - PALESTINE

ABSTRACT

Literature showed quite clearly that the efficiency of Air to Fuel Ratio (AFR) control for Spark Ignition (SI) Internal Combustion Engines (ICE) strongly depends on its capacity to deal with the fuel-flow phenomena inside intake manifolds.

Moreover, engine performances (such as power output, specific fuel consumption, and exhaust gas emissions) are directly related to the efficiency of the combustion process, which, on its turn, can be affected substantially by the air/fuel ratio variations related to the fuel-film dynamics.

In this work a comprehensive model-based air/fuel ratio control technique is proposed: this is based on a dynamical model of the air dynamics inside inlet manifolds and on the online identification of the fuel-film parameters. Here the identification procedure is illustrated in detail and validated basing on experimental data regarding a single-cylinder engine. In order to demonstrate the method validity, in fact, a single-cylinder research engine (type AVL 5401) equipped with a port-fuel injection system has been experimented using a dynamic test bench.

The first-order model of Aquino has been used as a basis to form the required algorithm for the identification process. The required input variables are the amount of fuel injected and that of the air inducted in the cylinder. This latter value is not obtained (as usual) by a look-up table stored in the ECU, but is calculated within each engine cycle by a mathematical model of the air dynamics developed by the same authors. The model used is named Model Of Interconnected Capacities (MOIC) and has been previously presented and separately validated.

INTRODUCTION

SI engines performances (power output, specific fuel consumption, and exhaust gas emissions) directly depend on the efficiency of the combustion process, which can be affected substantially by the AFR variations.

The efficiency of the AFR control system strongly depends on its capacity to deal with the fuel-flow phenomena, which affects the fuel transport processes inside the intake manifold (fuel vaporization dynamics and transport processes).

Those present a particular complexity being strongly influenced by engine speed and load variation, as well as by injection timing, injection duration, spray characteristics, flow field in the intake pipe, intake port design, air and wall temperatures.

Some of the injected fuel, in fact, will impinge on the port walls, on valve stem, and on the backside of the intake valve forming the fuel-film, causing a divergence between the injected mass of fuel and that inducted within the cylinder. A compensation action is therefore necessary.

As underlined by many studies such as (Aquino, 1981; Anatone, et al., 1998; Cipollone, and Di Dionisio, 1997; Turin, et al., 1994; Moyne, and Maroteaux, 1997; Stanton, and Rutland, 1998, Gambino, et al., 1994), wall wetting, evaporation off the wall, and liquid along the wall are all likely to be important within AFR control strategies.

Over the last few years, many authors proposed the substitution of the map-based control (conventionally implemented by OEM ECUs) with a more intelligent control based on a engine model running on the ECU (model- based AFR control) and simulating air, as well as fuel-film dynamics.

In order to limit the computational complexity of those models, the fuel-film behavior is usually described making use of various semi-empirical parameters.

Many of the current AFR control strategies make use of pre-identified fuel-film parameters (Anatone, et al., 1998; Chang, et al., 1993, Jensen, et al., 1997) stored in ECU maps: the correspondent models in this case have a small accuracy especially in transient conditions. Many other authors have, therefore, introduced adaptive control strategies (Raymond, and Hans, 1994; Ault, et al., 1993; Dingli, et al., 1996) guaranteeing highly acceptable results with reference to small engine transients only.

To overcome these difficulties, new conception control systems rely on the on-line identification of the fuel-film parameters. A recent study (Simons, et al., 1998) has introduced a scheme for self-tuning regulators which performs the identification exciting the fuel path dynamics at a constant operation speed. This approach, anyway, is still far from being applicable within a model-based AFR control.

Taking the lead by these considerations, in this work a comprehensive model-based AFR technique is proposed, making use of an air-dynamics model and on the online identification of the fuel-film parameters.

For the air dynamics, the model used was the Model Of Interconnected Capacities (MOIC): this was developed within the last years by the same authors and was previously presented and separately validated (Cipollone, and Sughayyer, 2001). Here only the solving algorithm of the model is reported and briefly discussed.

The on-line identification procedure for the fuel film parameters is, instead illustrated and reported in detail. Its efficiency was also validated basing on experimental data regarding a single-cylinder engine. In order to demonstrate the method validity, in fact, a single-cylinder research engine (type AVL 5401) equipped with a port-fuel injection system has been experimented using a dynamic test bench.

The proposed method allows AFR control system to be self-tuned to any transient situation without the necessity of fixed operating point reference or limiting the control system just to small fuel-path excitations.

The identification procedure was applied to the Aquino compensator (Aquino, 1981), which is the most popular and widely-used model for fuel-film dynamics. The model equations have been solved on the basis of successive cycles and rearranged to allow self-correction of the fuel-film parameters using a very simple algorithm regardless of the operating point.

The input/output scheme of the identification process is similar to that used in (Simons, et al., 1998), which considers as inputs the amount of fuel injected and that inducted in the cylinder: the first is derived by the injection durations stored in the ECU; the second is calculated trough the MOIC, making use of the output signal of the lambda sensors.

SCIENTIFIC BACKGROUND

Port fuel injection (PFI) systems are very common among fuel injection systems due to its fast response and to the high controllability of injected quantity of fuel. In PFI engines, liquid fuel is injected with low pressure (2.5-3.0 bar) into the relatively low-pressure atmosphere through an electronic controlled injector. Because of the low atomization of the fuel spray, some of the liquid fuel adheres to the port wall and the backside of the inlet valve. Consequently, this spray-wall interaction leads to uncontrolled mixture preparation. The AFR control efficiency of PFI engines can be, therefore, improved significantly by optimizing the mixture formation process in their intake manifolds.

In the most of the PFI systems, liquid fuel is injected in the intake port directly aiming to the backside of the intake valve. A portion of the injected fuel stays airborne, but a significant part is deposited on the port walls, generating a fuel film. The fuel quantity that adheres to the intake port surfaces depends on many factors and parameters, as discussed in many previous works (Aquino, and Fozo, 1988; Bourke, and Evers, 1994; Simons, et al., 1998; Simons, et al., 2000; Cho, et al., 2000; Almkvist, et al., 1995; Senda, et al., 1999; Anatone, and Carpellucci, 1999): intake port design; flow field in the intake duct; spray characteristics; injection timing relative to the intake valve open timing; temperatures of air and surfaces; engine speed and load.

The transport of the fuel into the engine cylinder occurs as vapor, droplets and liquid streams from the wall fuel film. As a consequence, fuel dynamics is slower than air dynamics: therefore, especially under transient engine operation, the deposition or discharge of the fuel from the film leads to distortions in the AFR of the inducted mixture. Moreover, especially at part-load, when the intake valve opens, backflow of the exhaust gases from the cylinder may shatter into the intake port the fuel accumulated on valve backside, enhancing fuel vaporization and redistribution on the port walls. These effects still remain uncompensated, leading to significant errors on AFR (Simons, et al., 2000).

The influence of engine temperature on AFR excursions during transient operations, is one of the most widely studied aspects. (Bourke, and Evers, 1994), for example, showed that the fuel film is more stable at the higher temperatures: during throttle ramps it doesn’t increase significantly because the hotter manifold allows the evaporation of the fuel. They also discussed the effect of temperature on AFR excursion due to throttle ramps: at the higher temperatures, the initial lean excursion is much lower and the rich excursion is slightly reduced; the duration of AFR excursions also is much lower at higher temperatures. Moreover, many studies showed that the mass of the fuel-film on the intake walls in cold engine conditions (coolant at 30 °C) is about 4 times that experienced in hot engine conditions (coolant at 80°C).

Other researchers, such as (Aquino, and Fozo, 1998), studied the effects of the mean size of the droplets of the injected fuel: they showed that droplet size influence the building of the fuel film but it doesn't affect the AFR excursions during throttle transients. Many other authors, at last, concentrated on the spray characteristics of the injector and the injection cone angle.

A variety of techniques, based on different physical principles, have been used to predict or measure the mass of the fuel film in the intake manifold. Many researchers proposed to correlate it with the composition of the exhaust gases. Many others used complex and detailed 3-D modeling of the injection spray. Others, on the other end, concentrated on the design of control-oriented models to be used for on-board reduction of AFR excursions during throttle ramps. These models provides adequate fuel compensation for most of the transient throttle movements, but did not completely describe the film dynamics, resulting in poor predictions in case of high speed engine transients. The approach used for this work is to overcome these difficulties, designing a comprehensive online model-based AFR controller. A model-based controller should permit transient compensations in each working conditions, being based on a full (even tough simplified) simulation of engine behavior constantly performed on board by the ECU.

FUEL DYNAMICS

Fuel-film dynamics in SI engines is very complex, especially at the microscopic level. Fortunately, however, for AFR control purposes is usually sufficient to make use of a macroscopic approach. One of the most popular models describing the behavior of the fuel-film is due to Aquino (Aquino, 1981): this simple first-order model tracks macroscopically the liquid puddle dynamics inside engines intake manifold, according to the following set of equations:

$$\frac{df}{dt} = x\frac{i}{T} - \frac{f}{\tau} \tag{1}$$

$$v = (1-x)\cdot i \tag{2}$$

$$\frac{c}{T} = \frac{v}{T} + \frac{f}{\tau} = (1-x)\frac{i}{T} + \frac{f}{\tau} \tag{3}$$

where: *f* is the mass of the fuel film; *i* is the injected fuel mass; *v* is the part of injected fuel mass flow rate that vaporizes and enters in the cylinder; *c* is the total fuel mass inducted into the cylinder; *x* is the fraction of the injected fuel that enters the film; τ is the characteristics time of the fuel that leaves the film. All the values are referred to an engine period *T*.

The film dynamics is predicted by this model in response to a step increase in engine load. Since only (*1-x*) of the fuel flows directly with the air, if the throttle is opened rapidly a lean mixture is predicted. The Aquino model has been widely used to develop fuel-metering strategies, which compensate for the fuel transport lag, and has therefore been chosen as reference for the application of the proposed identification procedure. The following procedure has been applied for the identification of the two semi-empirical Aquino parameters: *x* and τ. The Aquino compensator was chosen to make it possible to validate the procedure basing on reference data for *x* and τ. stored in the ECU maps.

THE IDENTIFICATION PROCEDURE

The model equations have to be applied to two successive cycles (named respectively *0* and *1*), then rearranged and solved to obtain the required equations, which will form the basis of the algorithm used in the identification process.

At a given engine cycle (*0*) the mass of the fuel film at the walls (f_0) can be supposed known, and Eq. 3 gives:

$$\frac{c_o}{T_o} = (1\text{-}x)\frac{i_o}{T_o} + \frac{f_o}{\tau} \tag{4}$$

The same equation states for the next cycle. Making the hypothesis that within the cycle the Aquino parameters do not change substantially, it can be written that:

$$\frac{c_1}{T_1} = (1\text{-}x)\frac{i_1}{T_1} + \frac{f_1}{\tau} \tag{5}$$

where *x* and τ are the parameters which have to be identified, while i_0, c_0, i_1, i_1 are the input values of the procedure and can be can be evaluated as explained in the following paragraph. The mass of the fuel film at the following cycle (f_1 in Eq. (5)) can be, instead, evaluated integrating Eq.1 on an engine period:

$$f_1 = f_0 + \left(x\frac{i_0}{T_0} - \frac{f_0}{\tau}\right)\cdot T_0 \tag{6}$$

Rearranging Eq. 4 to 6 *x* and τ. can be evaluated according to:

$$x = \frac{T_1(i_0 - c_0)^2 + f_0[T_0(i_1 - c_1) - T_1(i_0 - c_0)]}{f_0(i_0T_1 - i_1T_0) + i_0T_1(i_0 - c_0)} \tag{7}$$

$$\tau = \frac{f_0(i_0T_1 - i_1T_0) + i_0T_1(i_0 - c_0)}{i_0c_1 - i_1c_0} \tag{8}$$

Eq. (7) and (8) permit the identification of x and τ: f_0 is known and can be updated (for the application of the procedure to following cycles) applying Eq. (6); T_0 and T_1 are directly evaluated by the ECU; i_0, c_0, i_1, i_1 (the main input variables for the identification procedure) are evaluated according the following procedure.

EVALUATION OF THE INPUT VARIABLES FOR THE IDENTIFICATION PROCESS

The evaluation of the input variables for the identification procedure has been performed basing on the following hypotheses:

The injected mass of fuel (i) is proportional to the injection duration (ΔT_{inj}), obtainable by the ECU. By this assumption:

$$i = k_{inj} \cdot \Delta T_{inj} \tag{9}$$

The total mass of the inducted fuel (c) can be obtained by the mass of air inducted and the output signal of the lambda sensor (λ), according to:

$$c = \frac{m_{ac}}{\phi_{st} \cdot \lambda} \tag{10}$$

where m_{ac} is the air trapped inside engine cylinder, ϕ_{st} is the stoichiometric AFR and λ is the relative air fuel ratio obtained from exhaust gas. In Eq. (10) m_{ac} is referred to the control cycle (that under control in terms of AFR control), while λ is the feedback which is available after many events: compression, expansion, mixing processes inside the exhaust manifolds and transport till the sensor position. All these phenomena produce a time delay which must be taken into account; the time response of the sensor has also to be considered.

A dedicated model has been used for the evaluation of m_{ac} named MOIC. This has been previously proposed and validated by the same authors and is briefly reported in the following.

AIR DYNAMICS AT THE INTAKE MANIFOLD

In a previously published work (Cipollone, and Sughayyer, 2001) a new method of modeling air dynamics inside intake manifolds was projected to run on-line directly implemented in the ECU: it, therefore, suits the case here to form a comprehensive model based controller. The method is called Method Of Interconnected Capacities (MOIC) and takes into account the inertia of the inducted air dividing the inlet manifold in various capacities exchanging mass and energy one to another in a dynamical way.

The dynamical equations describing the model are reported below. Their full derivation from the conservation equations imposed for the manifold is reported in (Cipollone, and Sughayyer, 2001).

$$\begin{bmatrix} \dfrac{dP_j}{dt} \\ \dfrac{dT_j}{dt} \\ \dfrac{d\bar{u}_j}{dt} \end{bmatrix} = \frac{(1-y)T_j}{P_j \cdot V_j} \begin{bmatrix} \dfrac{\bar{u}_j^2 P_j}{2T_j}\dfrac{dm_j}{dt} - \dfrac{P_j}{T_j}\dfrac{dE_j}{dt} + \dfrac{\bar{u}_j^2 P_j}{2T_j}\Phi \\ \left(\dfrac{RT_j}{y-1} + \dfrac{\bar{u}_j^2}{2}\right)\dfrac{dm_j}{dt} - \dfrac{dE_j}{dt} + \bar{u}_j\Phi \\ \dfrac{R}{1-y}\Phi \end{bmatrix} \tag{11}$$

$$\frac{dm_j}{dt} = \dot{m}_{i,j} - \dot{m}_{e,j} \tag{12}$$

$$\begin{aligned} \frac{dE_j}{dt} &= \left(\frac{\gamma}{\gamma-1}RT_j + \frac{u_i^2}{2}\right)\dot{m}_{i,j} - \left(\frac{\gamma RT_e}{\gamma-1} + \frac{u_e^2}{2}\right)\dot{m}_{e,j} + \\ &+ \frac{P_j \cdot V_j}{RT_j}\frac{2fC_P|\bar{u}_j|}{D}\ (T_w - T_j\ ) \end{aligned} \tag{13}$$

$$\Phi = P_iA_i - P_eA_e - \bar{u}_j\frac{dm_j}{dt} - \frac{|u_j|}{u_j}f\frac{\bar{u}_j^2}{2}\frac{4}{D}\frac{P_j \cdot V_j}{RT_j} \tag{14}$$

being: j a suffix referred to the capacity under examination, $m_{i,j}$ and $m_{e,j}$ the inflow and the outflow masses; P, V and T the pressure, volume and temperature of the capacity; R the gas constant; E the internal energy in the capacity; h_i and h_e the specific stagnation enthalpies of the entering and exiting masses; γ the ratio of the specific heats, f the friction coefficient, CP the constant pressure specific heat; D the mean duct diameter; T_w the mean wall temperature.

In the calculation of Φ, a further very helpful simplification can be done, by which a sufficiently accurate solution can be obtained. In that, P_e has been replaced by P_i and the mean value of u_i can be considered equal to u_e.

At the first and at the last of the capacities representing the inlet manifold, two boundary conditions must be imposed to solve the problem (in terms of P and T at the inlet valve).

At the inlet section, in case of direct flow (air entering in the manifold), the application of the energy conservation for isentropic flow gives:

$$\dot{m}_t = \begin{cases} \dfrac{C_t A_t P_0}{\sqrt{RT_0}} \left(\dfrac{2\gamma}{\gamma-1}\right)^{0.5} \left(\dfrac{P}{P_0}\right)^{1/\gamma} \sqrt{1-\left(\dfrac{P}{P_0}\right)^{(\gamma-1)/\gamma}} \\ \text{(subsonic flow)} \\ \dfrac{C_t A_t P_0}{\sqrt{RT_0}} \gamma^{0.5} \left(\dfrac{2\gamma}{\gamma-1}\right)^{(\gamma+1)/(2\gamma-2)} \\ \text{(sonic flow)} \end{cases} \quad (15)$$

while for reverse flow all the kinetic energy of the air can be supposed to be dissipated (stating that $P = P_0$).

At valve section, in case of direct flow (from the manifold to the cylinder):

$$\dot{m}_t = \begin{cases} \rho C_v A_P \sqrt{\dfrac{\dfrac{2\gamma RT_c}{\gamma-1}\left(\dfrac{P}{P_c}\right)^{(\gamma-1)/\gamma}}{\left(\dfrac{A_P}{A_v}\right)^2 \left(\dfrac{P}{P_c}\right)^{2/\gamma} - 1}} \\ \text{(subsonic flow)} \\ \rho C_v \dfrac{A_v^2}{A_P} \dfrac{\sqrt{\gamma RT_0}}{K^{2/(\gamma-1)}} \\ \text{(sonic flow)} \end{cases} \quad (16)$$

where K is given by the solution of:

$$\left(\frac{A_P}{A_v}\right)^2 = \left(\frac{\gamma+1}{\gamma-1} - \frac{2}{\gamma-1}K^2\right)K^{4/(\gamma-1)} \quad (17)$$

In case of reverse flow, energy is conserved up to the throat section of the valve, while all the kinetic energy is lost after the throat section.

The second after the valve is referred to a sudden enlargement till to the wall-pipe-section where the kinetic energy can be considered lost.

The solution of MOIC properly implemented by the ECU gives a precise estimation of the inducted mass of air which can be used as input in the identification procedure for fuel film parameters.

METHOD VALIDATION

For the validation of the proposed method, a single cylinder research engine type AVL 5401 has been experimented for a wide range of random transient tests using different mixture strength levels from rich to lean. Also different engine speeds were considered in addition to the coolant temperature variation around the engine operating temperature.

The exhaust manifold was equipped with a wide-band oxygen sensor and with a fast AFR analyzer (Dual-Channel AF Recorder 4800). Pressure signals were measured by an high-accuracy fast-response sensor (micro-mechanical absolute-pressure sensors by Bosch). Temperatures by Bosch thermo-resistors. Engine speed and TDC by a hall-effect sensor. Throttle position by a precise potentiometer.

A personal computer was equipped by a National-Instrument acquisition board (NI PCI 6034E) and was used for signals acquisition. All the acquired signals were suitably processed to avoid signal disturbances.

The sensors have a better performance with reference to those equipping production systems. Anyway, the choice was to use equipments with can be reasonably supposed to be put on-board in next-generation systems: no laboratory equipment and sensors were used in the identification procedure. The system were able to perform the identification within the engine cycle at 6000 Rpm: computation load was, therefore, fully compatible with the equipment used. For on-board applications, however, it must be considered that the ECU will replace the dedicated personal computer, and the procedure should be applied to multi-cylinder engines. The computational load will therefore become much more critical.

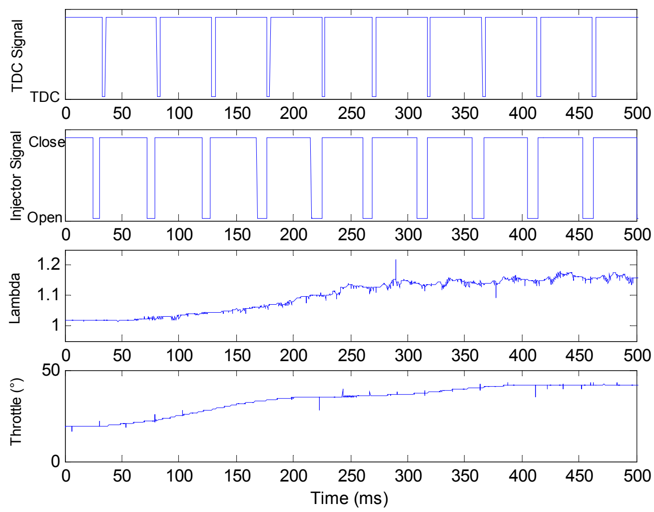

Fig. 1. Transient at 2500 rpm.

Figure 1 shows the principal four signals in an engine transient test at 2500 rpm in the lean mixture range with throttle opening ramp of 28 degrees over a period of 0.35 second, The reference values of the fuel-film parameters that have been used in the validation process were those mapped by AVL for the research engine (Table 1): these were obtained by AVL for small engine transients.

For the validation process, two test-cases were performed to validate the proposed identification procedure, respectively regarding low and medium speed transients.

Table 1: Reference data.

T_w [°C]	x	τ
54	0.33	0.51
62	0.31	0.44
72	0.30	0.39
85	0.30	0.39

FIRST CASE STUDY: LOW SPEED TRANSIENTS

Starting engine speed was set to 2000 rpm in order to simulate a typical part-load acceleration. The research engine was also loaded partially what usually happens when the vehicle accelerates from low to medium speed (e.g. during a surpass or increasing velocity on the highway). Figure 2 reports, as an example, one of the measured throttle ramps and the correspondent AFR excursions.

Table 2 reports the values identified for the fuel-film parameters by the application of the procedure previously described: the values refer to different transient tests with the research engine part-loaded. In each test a wide throttle opening, as in Fig. 2, was applied at different wall temperatures to explore its influence on the fuel-film dynamics.

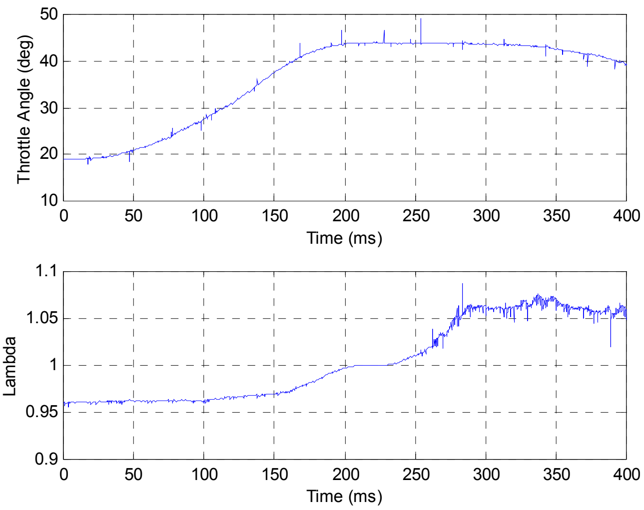

Fig. 2. Second case-study: transient at 2000 rpm.

The reported results in the previous table show that the fuel-film parameter τ is much more variable than x. The agreement by the identified values of x and the reference values is satisfactory: the same cannot be said for τ. Strong variations, in fact, occur in the predicted value for τ: these results can be explained considering that all the tests in Table 2 refer to a relatively low engine speed, in a situation where heat transfer between manifold and film and backflow are expected to influence significantly fuel vaporization from the film and film building. This situation cannot be represented completely by such a simple model such as that implemented by the Aquino compensator.

Table 2: First case-study: identified parameters.

Test #	T_w [°C]	x	τ
1	58	0.27	0.48
2	58	0.27	0.40
3	58	0.31	0.55
4	57	0.27	0.39
5	57	0.27	0.58
6	57	0.29	0.53
7	60	0.28	0.55
8	74	0.27	0.48
9	74	0.27	0.35

SECOND CASE-STUDY: MEDIUM SPEED TRANSIENTS

Starting engine speed was set to 2500 rpm to simulate another typical part-load transient. The research engine was loaded partially to simulate those accelerations from a medium speed to higher one. The throttle ramps were similar to those of the previous case, as shown in Fig. 3.

The results of the identification procedure are reported in Table 3 and demonstrate that at higher engine speeds the fuel-film parameters (especially τ) become more stable. This result is also in perfect agreement with what underlined by other authors (Bourke, and Evers, 1994).

A closer agreement with the reference data given in Table 1 is also evident.

These results for medium speed transients recommend the use of engine speed as a threshold parameter to activate the proposed identification algorithm of the fuel-film parameters in the on-board applications.

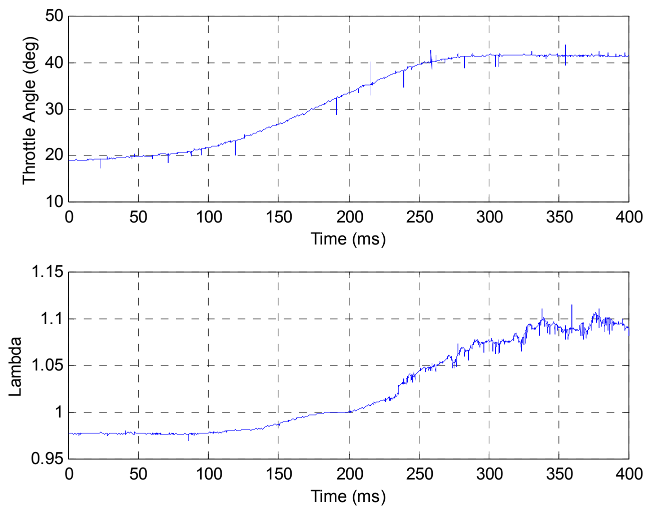

Fig. 3. Second case-study: transient at 2500 rpm.

Table 3: Second case-study: identified parameters.

Test #	T_w [°C]	x	τ
1	55	0.31	0.37
2	57	0.32	0.47
3	63	0.26	0.41
4	63	0.27	0.40
5	68	0.29	0.50
6	63	0.27	0.42

CONCLUSIONS

A new approach for identifying online the fuel-film parameters has been developed. It allows the air to fuel ratio control system to be self-updated on a wide range of transient situations. The proposed on-board procedure also would permit the avoidance of the time-consuming off-board experimentations that are usually to be setup to identify the fuel-film parameters.

Experimental validation has been performed under transient engine conditions on a single-cylinder research engine (AVL 5401).

The agreement between the results obtained from the proposed procedure and the reference data (mapped on the ECU) is quite satisfactory.

The method underlines the uncertainties in such parameters when the engine speed decreases (due to heat transfer and backflow influence) and suggests the engine speed as a threshold parameter to activate the procedure onboard.

The proposed method is under improvement to be applied to multi-cylinder engines and to be included in a model-based AFR controller.

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INDEX OF SYMBOLS

A	Cross-sectional area
C	Coefficient of discharge
c	Inducted mass of fuel, Specific heat
D	Diameter
E	Outlet section
E	Internal Energy
f	Mass of the fuel film; Friction factor
i	Injected mass of fuel, Inlet section
f	Fuel-Film Mass
V	Air-born vaporized fuel mass
N	Engine speed
P	Pressure
R	Gas constant
T	Engine period; Temperature
U	Flow velocity
V	Volume
x	Fuel-film parameter
□	Relative air to fuel ratio
□	Fuel-film parameter
□	Ratio of specific heats