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Air-path control for a prototype PCCI diesel engine

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Abstract— **The tightening in diesel engines regulations for emissions control forced the automotive industry to face the challenge coming from these constraints. This led to an increase in the overall engine complexity, in both, mechanical and software domains. For what concerns this paper the increased complexity means to deal with and to manage the cross coupled and nonlinear behavior of the engine air-path. Here we propose a control system able to manage under steady-state conditions the airflow through the engine with the purpose of ensuring a smooth Premixed Charge Compression Ignition combustion behavior using the most simple and economical structure possible. The objective was to design a controller capable to handle the three air-path devices of Exhaust Gas Recirculation, Variable Geometry Turbochargers and *flap* valve all together. Modeling of the system and control design are the two main areas on which we focused our attention in order to retrieve simplicity and system economy. All the trials needed to design the air-path controller were carried out on the mono-dimensional simulation software GT-Power by Gamma Technologies that has proven to be very effective in the simulation domain. The model was created by FPT Industrial and provided to us together with the engine. Based on that model we derived all our computations, in terms of system identification and validation, that led us to a feed-forward plus feedback control structure capable to cover the whole engine operating area. Simulations show very encouraging results also in presence of high disturbances in the outputs.**

I. INTRODUCTION

Engine emissions are a common topic in the present climate debate due to their impact on the environment. The emission levels have been reduced constantly over the last two decades and the whole automotive industry was forced by governments to fulfill the law requirements. Technologies like Exhaust Gas Recirculation (EGR) and Variable Geometry Turbochargers (VGT) have been developed and improved by years in order to meet these requirements.

Recirculating exhaust gas back into the cylinder, through the EGR valve, allows to reduce Nitrogen Oxides (NO_x) formation. On the other hand particle matter (soot) formation increases. A compromise between NO_x and soot emissions has to be found. VGTs are commonly employed in compression ignition engines since they make possible to expand the usable flow rate range in practical applications while maintaining a high level of efficiency. A back pressure valve or *flap* valve can be placed after the turbine exhaust to regulate the pressure in the exhaust manifold affecting the expansion rate and thus the pressure difference over the turbine and as consequence the EGR air mass flow. This is also used after engine cold start to increase the temperature of the exhaust gas entering the aftertreatment systems to reduce the light-off of the DOC (diesel oxidation catalyst).

Consequently, the airflow path can be controlled by means of the three valves: EGR, VGT and *flap*. A coordinated control is necessary since the EGR, VGT and *flap* actuation effects are coupled and affect in a complex manner the EGR fraction and the air entering the cylinder and as a consequence the whole engine functioning. Furthermore the characteristics of the actuators are highly non-linear and depending on the working condition of the engine. The development of a controller able to handle both steady-state and transient conditions plays a primary role into engine functioning and management.

The aim of this work is to develop a controller able to manage in steady-state conditions the engine airflow, with the purpose to enhance the behavior of a prototype 3-liter diesel engine modified in order to optimize Premixed Charge Compression Ignition (PCCI) combustion mode. More details about this work can be found in [1].

Control of High Pressure (HP) EGR combined with VGT and *flap* is a topic that in the past years called the attention and the interest of researchers. This is mainly due to the challenging nature of this problem: it is multi-variable and highly nonlinear, with sign reversal as well as non-minimum phase behaviors. Furthermore the behavior of the system depends on the operating point. This led to an increase in the overall engine complexity in both mechanical and software domains. In fact conventional control methodologies such as PIDs had to give way to new control strategies such as predictive control, [2] and [3], neural networks [4], fuzzy logic [5] and H_∞ [6]. The drawbacks of these new control techniques are basically their complexity and difficulty, as they are not intuitive, in tuning the parameters. Currently the model based approach is the most studied and used as the advantages deriving from the disposal of an embedded model in the controller fit in with these new control strategies. Examples of model-based designs can be found in [7], [8] and [9].

II. PROBLEM DEFINITION

The engine in exam is a 3-liter, 4 cylinders prototype endowed with a HP-EGR, VGT and *flap* derived from the F1C Euro VI engine of FPT Industrial by means of some modification needed to adapt the engine to the PCCI combustion. In particular the compression ratio is reduced from the standard 17.5:1 to the lower 14.6:1.

The PCCI combustion falls into the Low Temperature Combustion (LTC) methods and allows to reach a high level

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of air-fuel mixing. In application fuel is injected directly into the combustion chamber during the compression phase well before the Top Dead Center (TDC). This EGR-diluted system makes possible, if fuel is injected early enough and mixing rates are sufficient, to achieve a premixed charge before the Start Of Combustion (SOC), to simultaneously reduce NO_x and soot emissions. Compared to conventional diesel combustion, a single early injection strategy in a charge with high EGR rates, extends the ignition delay due to injection in a colder environment and increase fuel penetration due to a lower in-cylinder pressure. This enhances the premixing between the fuel and the surrounding charge. The high EGR rates promote a slower combustion with respect to the conventional one, reducing the peak temperatures responsible of NO_x emissions, whereas the premixing reduces the formation of soot to a great extent. However, some drawbacks relative to over-penetration and wall wetting can be experienced (leading to high engine-out HC values) together with a higher fuel consumption due to a slow combustion and maybe to a not optimally phased combustion with respect to TDC. EGR rate becomes of relevant importance in PCCI combustion as it can be used to control the SOC since a lower oxygen content affects ignition delay. Other drawbacks of this combustion process compared to a conventional combustion are noise and higher in-cylinder pressure gradient that means higher mechanical stress for the engine and a higher cycle to cycle and cylinder to cylinder variability due to the high rate of EGR.

The aim of this work is to develop a controller able to manage in steady-state conditions the engine airflow. Control inputs are the opening of the EGR valve, u_{EGR}, VGT valve u_{VGT} and *flap* valve u_{FLAP}. The principal gas flows for a diesel engine are the airflow through the compressor W_a and the exhaust gas recirculation flow W_{EGR}. These are to be controlled to desired levels and their dynamics are important for the overall control objectives. The target for the designed controller in this work are the intake manifold pressure p_{im} and mass air flow W_a. These two variables are directly measured while the EGR mass flow has to be calculated. The layout of the engine with the control variable is shown in Fig. 1.

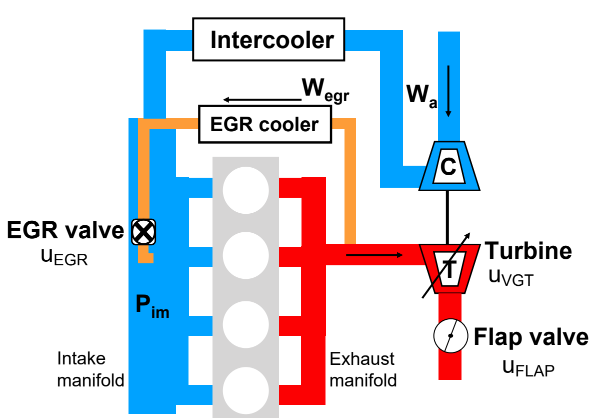

Fig. 1. Engine layout with control variables.

III. PLANT IDENTIFICATION

In order to avoid to run a large amount of extremely expensive tests on the real engine, a preliminary detailed simulation model was built using GT-Power Engine Simulation Software. This simulation model was obtained on the base of few tests on the real engine available in the test bed, described in [10], at Politecnico di Torino and it was subsequently successfully validated. For the sake of brevity the development and the results of this preliminary identification phase are not reported here. The simulation model provided by GT-Power was then used to carry on a large amount of tests and all the computation needed to identify a simplified engine model.

Different approaches to model the engine behavior are presented in the literature often based on mean value techniques. The approach used in [11] describes the engine behavior through physical relationships. The result of this modeling approach is a very good accuracy paid with algebraic complexity, high order of the model resulting in a high computational cost. In order to lower the order of the model with a good choice of the system states some of those equations can be neglected. This leads to the so called “third order engine model” that has been presented and used in many papers such as [12], [13] and [14]. But in our specific case the third order model cannot be implemented because it requires to add a fourth equation needed to model the *flap* valve. This extra equation causes a sharp increase in the model complexity.

The highly non-linear and coupled behavior of the system composed by EGR, VGT and *flap* valves is extremely difficult to express relying on equation describing their physics. For this reason black-box model estimation was used to build the engine model. Other examples of black-box models application can be found in [3], [15] and [16]. The relevant feature for the final choice of the model was a suitable balance between low order and model error. The whole operating space of the engine was divided in four sub-regions and for each sub-region a specific model was estimated in order to better represent the overall behavior of the engine. These four sub-regions correspond to a fixed engine speed of 2500 rpm and to Brake Mean Effective Pressure (BMEP) ranges as shown in table I; they identify the areas of low, medium-low, medium-high and high load in which the engine can operate. The data used in the estimation process were obtained from simulation tests carried on the GT-Power simulation model. To perturb the system around the four sub-regions working points, square waves were used

TABLE I
ENGINE WORKING REGIONS

Load	BMEP (bar)
low	$0 \div 2.5$
medium-low	$2.5 \div 5$
medium-high	$5 \div 7.5$
high	$7.5 \div 10$

TABLE II

SQUARE WAVES CHARACTERISTICS USED IN ESTIMATION

		Load			
		high	med-high	med-low	low
EGR	ϕ	17.6÷19.6	25÷35	23.6÷ 26.6	11.9÷12.9
	τ	24	23	18	15
VGT	(%)	30÷40	54÷64	88÷98	51÷61
	τ	20	20	16	12
FLAP	ϕ	29.4÷32.4	30.5÷32.5	25.1÷27.1	14÷15
	τ	23	22	21	14

as actuation command for the EGR, VGT, and *flap* valves. The characteristics of the actuation signals are shown in table II where ϕ stands for the diameter of the valve in millimeter (mm) and τ for the period of the signal in seconds (s). It is of a relevant importance that this is a non-minimum phase system with cross coupled behavior affected by a high level of uncertainty.

Model estimation was carried out with Matlab System Identification Toolbox. Through GT-Power the four chosen point of functioning were perturbed acting simultaneously on the inputs of EGR, VGT and *flap*. Input variations were chosen in order to induce the intake manifold pressure and the air mass flow to remain in the linear region close to the central position of the original point, thus avoiding to stimulate non-linearities.

Experimental data from GT-Power were processed in order to remove mean values and trends. From these datasets transfer functions and state-space models were derived. The overall best compromise between model order and model error was offered by the second order models. State-space models have been preferred to transfer functions for their homogeneity as they presents the same denominator, that corresponds to the same dynamic, in every channel. The estimation of transfer function, even if it provides good results, does not ensure the same denominator in all the channels. A better estimation result would have been obtained using hybrid models in terms of order. In this way, through a puzzle, transfer functions would have been preferred to state-space model. But this would have meant the need of specifically tuned controller for each channel for each of the four models. All the above lead us to the use of second order state-space models.

As an example, the estimated state space model for the medium-low load is shown in (1):

$$\dot{\Delta x} = \begin{bmatrix} -1.2261 & 0.6047 \\ 2.5161 & -3.0775 \end{bmatrix} \Delta x +$$
$$+ \begin{bmatrix} 0.0156 & -0.7379 & -0.0265 \\ -0.1080 & 0.6502 & 0.2726 \end{bmatrix} \begin{bmatrix} \Delta u_{EGR} \\ \Delta u_{VGT} \\ \Delta u_{FLAP} \end{bmatrix} \quad (1)$$
$$\begin{bmatrix} \Delta p_{im} \\ \Delta W_a \end{bmatrix} = \begin{bmatrix} 0.3026 & -0.0183 \\ 0.0013 & 0.0101 \end{bmatrix} \Delta x$$

This model showed an overall fitting with the experimental data of 88.12% and 84.32% with respect to the intake manifold pressure and the intake air mass flow respectively.

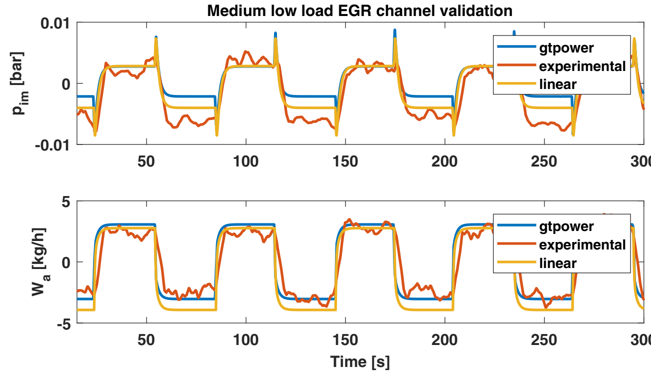

Fig. 2. Validation of the linear model EGR channel.

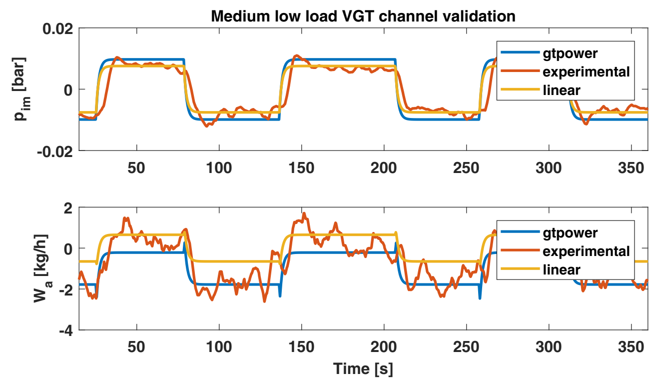

Fig. 3. Validation of the linear model VGT channel.

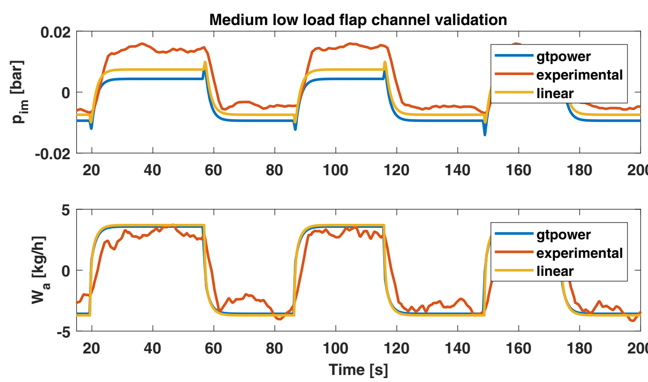

Fig. 4. Validation of the linear model *flap* channel.

Similar results, here not reported for the sake of brevity, were obtained for the three other working regions.

All the identified linear models were validated channel by channel in comparison with the GT-Power simulation model and with test data acquired on the real engine, using the same input signals. An example, concerning the medium-low load working point, is shown in Fig. from 2 to 4 where the intake manifold pressure p_{im} (upper figure) and intake air mass flow W_a (lower figure) are represented. The data from the GT-Power simulation are shown in blue, in red the ones measured on the actual engine in the test bench and in yellow the data from the linear identified models. In Fig. 2 and 4 the pressure spikes in the intake manifold are well evident for the GT-Power model and as a consequence in the estimated model.

This is due to the way of modeling the valves in GT-Power and due to the clean data coming from the simulations these peaks are clearly distinguishable. In real tests the engine does not show a smooth and steady behavior so the outputs oscillate. This makes necessary the use of a filter to clean the data around a medium value. In doing so the peaks get smothered. The pressure channel is well matched for the three actuations. Of these three channels, the one from u_{FLAP} to p_{im}, Fig. 4, is the most uncertain due to the direct influence on the turbine that is modeled through a discrete map. In addition, as the GT-Power model, the linear model speeds up a little the transients in this channel especially when the *flap* valve is opened. By looking at the channel from u_{VGT} to W_a in Fig. 3 the oscillatory behavior, even when a steady state is supposed, is evident and here the linear model suffers from this uncertainty. However the almost negligible influence of the VGT over the air mass flow makes this result acceptable. For the other two Fig. 2 and 4 the air mass flow is well matched with the real data even if the transient is a little faster than the real one for the channel from u_{FLAP} to W_a.

IV. CONTROL DESIGN

The control goal is to ensure a regular combustion behavior so that the engine, running in steady state condition, is able to show a regular and smooth response. Two are the main reasons for this control objective: i) the need of a controller able to manage the engine during the tests for the search of the optimal working conditions; ii) the potential of this type of engine, exploiting PCCI combustion, to run at fixed point. A smooth engine behavior in steady state conditions is achieved controlling the combustion process through the correct actuation of the valves in the air-path. This makes crystal clear the use of the valves actuation signals as inputs u for the identified models.

The model outputs y are the intake manifold pressure and the intake air mass flow. This choice stems from the simplicity in retrieving these variables in a series production engine as they are commonly accessible from the ECU. This leads to a simpler engine as no physical sensors have to be added to the engine and to a simpler control system since no objects like observers have to be designed and implemented as the data come out of the ECU.

The control structure is based on a classical closed loop feedback controller with the add of a feed-forward action. This feed-forward action is carried out by means of maps that, given the actual values of engine RPM and BMEP, provide the equilibrium condition opening values for each one of the three actuators. Those maps were built using optimized data obtained from the experimental campaign on the real engine.

As shown in Fig. 5, where ICE stays for Internal Combustion Engine, for the EGR channel only feed-forward action was used while for the VGT and *flap* channels the feed-forward action was added to a classical feedback control loop. The use of only feed-forward control for the EGR valve is justified as PCCI combustion mode requires a high amount

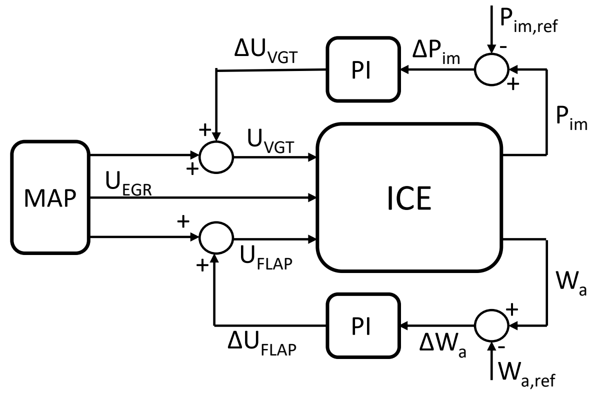

Fig. 5. Control scheme layout.

of recirculated gas. This fact limits the operating range of the valve around the almost fully open position as its main task is to guarantee the high amount of EGR. On the other hand the feedback control action is entrusted to two PI compensators, one for the VGT and one for the *flap*. To free from the cross-coupling effect that would have resulted in a multi-variable control system the control strategy used was to address each valve actuation to the major influenced output. This means to use the VGT valve to control the intake manifold pressure and the *flap* to control the air mass flow rate. An example of decoupled control can be found in [17]. In contrast to this, multivariable control examples can be found in [18] and [19].

The two compensators have been designed specifically for each channel and for each one of the four working regions, for a total of eight compensators. For all the feedback controllers the steady state requirement of having a zero error following a step input was chosen and from the time requirements a cutting frequency around 10 rad/s was chosen.

All the above plus the need of a low order control system were translated in the following PI structure for the compensators:

$$C(s) = \frac{K(1+\tau_n s)}{s}$$

The numerical values of the gain and the zero for the eight compensators are shown in table III.

V. CONTROL RESULTS

The controlled system performances were checked in simulation by means of the GT-Power model, able to reproduce the cross coupling effect of the system channels. Moreover, discrete random perturbation signals were added one for

TABLE III
K AND τ_n VALUES

Channel		Load high	med-high	med-low	low
P_{im}	K	-0.08178	-0.23976	-0.8312	-2
	τ_n	4.928	9.895	1.289	0.666
W_a	K	92.151	51.024	201.72	35.798
	τ_n	2.139	1.999	4.734	2.894

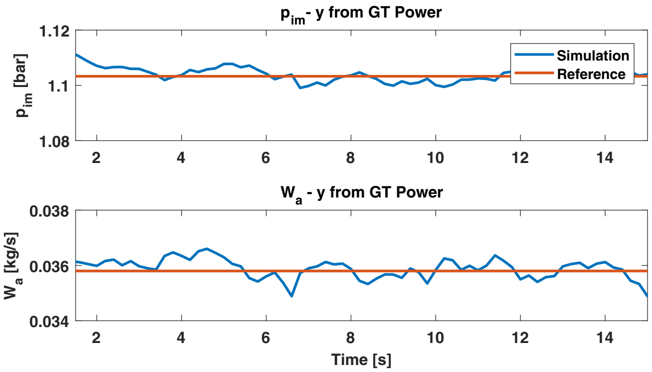

Fig. 6. Outputs y from GT-Power with disturbance injection and quantization interval in medium-low load conditions.

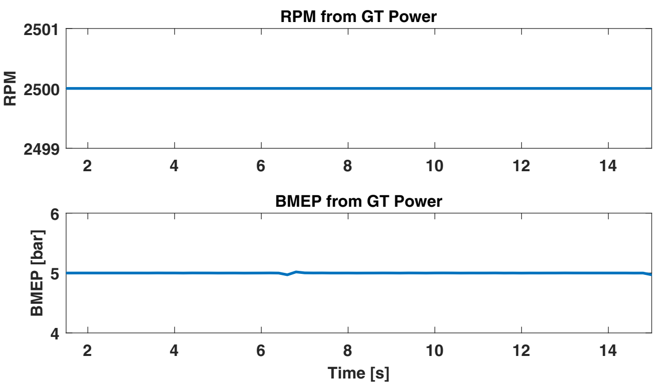

Fig. 7. Working point acquired from GT-Power with disturbance injection and quantization interval in medium-low load conditions.

each GT-power output to simulate measurements noise: the variance of the noise for p_{im} and W_a, respectively $5 \cdot 10^{-4}$ and $4 \cdot 10^{-6}$, was chosen according to the worst condition recorded in the test bench experiments. To further stress the controller behavior a quantization interval has been added in order to better simulate the real working environment in which only discrete opening values of the actuators are possible. A quantization interval of $0.5\ mm$ was chosen for the EGR and *flap* valves while for the VGT the interval was of 5%.

The results for the medium-low load condition test are shown in Fig. 6: the controller is able to maintain the outputs close to the reference value with minor oscillations. Fig. 7 shows the engine working point during the simulation. The working condition of 2500 rpm with 5 bar of BMEP is successfully maintained during the simulation. In Fig. 8 results for medium-high load conditions are reported. What is visible in Fig. 8 in the first part of the simulation until 6 seconds are the effects of the starting transient of the GT-Power model. This is much more evident in this case, rather than the previous in Fig. 6, because the start up transient is more aggressive due to the higher loads. After the settling down of the start up transient, the controller is able to track the references and to maintain the imposed working conditions as Fig. 9 shows. Similar results were obtained for the two other working regions.

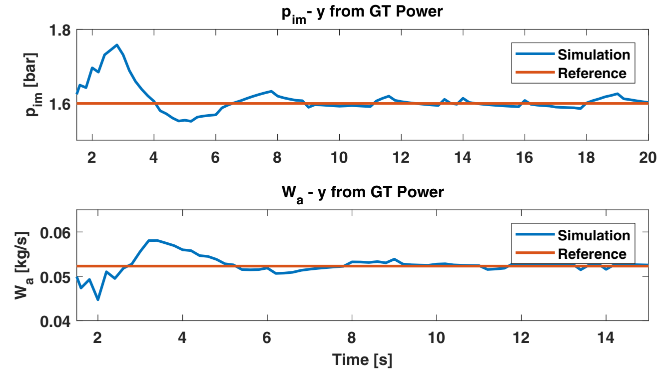

Fig. 8. Outputs y from GT-Power with disturbance injection and quantization interval in medium-high load conditions.

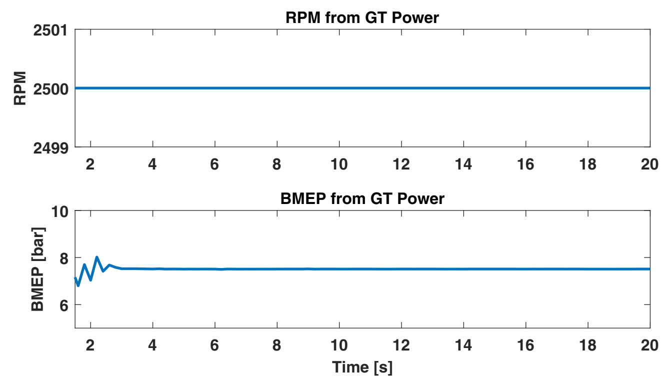

Fig. 9. Working point acquired from GT-Power with disturbance injection and quantization interval in medium-high load conditions.

Even if the controller was designed to deal with steady-state condition, also the switching action between the four engine sub-regions, performed through logic switches, was simulated. As shown in Fig. 10 the control system is capable to follow correctly slow transients moving between the sub-regions. Fig. 11 illustrates the RPM and BMEP profiles that were set in GT-Power and used for the transient simulation. First the BMEP was varied from an high load conditions corresponding to 10 bar to a medium-high load condition of 8 bar. In the same time instant in which the BMEP profile reaches 8 bar the engine speed starts increasing from 2500 to 3000 rpm. These two transient profiles are not intended for representing sharp variations. They represent relaxed and progressive transients like the ones usually performed in test benches while mapping the engine. Between the two channels, the one regarding the intake air mass flow, W_a, controlled through the *flap*, is the most critical one due to the presence of time lag between the valve actuation and the reaction of the output. This because any actuation of the *flap* valve affects the exhaust manifold and the turbo group with its own lag and then the air mass flow is affected. All this suggests that in order to significantly improve the control performance the lag problem as to be properly considered also trough a modeling procedure that contains the lag phenomena. Furthermore it is visible, from 10 seconds ahead

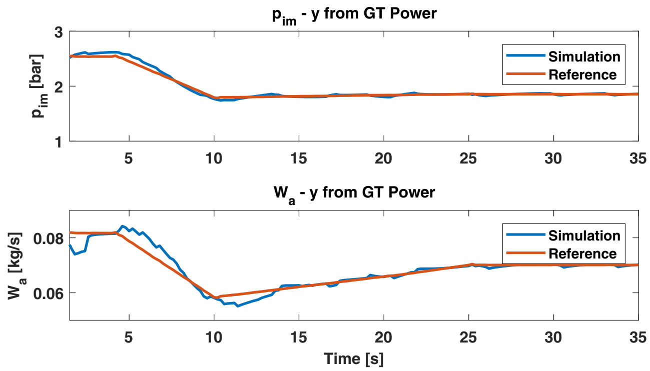

Fig. 10. GT-Power output during transient simulation.

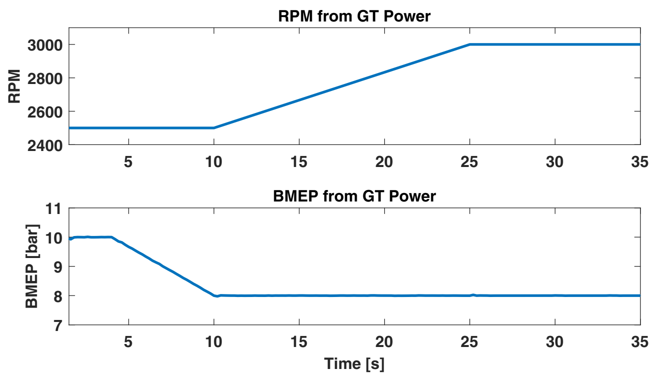

Fig. 11. RPM and BMEP transient profiles.

in Fig. 10, that the engine speed variation produces a greater change in the intake air mass flow rather than the intake manifold pressure. All the oscillations around the reference values are due to the quantization interval of each actuator.

VI. CONCLUSIONS

The objective of this work was to design a control system able to manage in steady-state conditions the engine airflow with the purpose of ensure a regular PCCI combustion behavior through the most simple and economical structure possible. As outcome we verified that both engine models, the detailed GT-Power one and the identified linear one, approximate the real engine behavior in a very good way, although some differences arose from the discretization in the elements described by maps, i.e. compressor and turbine, in the GT-Power model. Furthermore, despite the high content of uncertainty and disturbances, whether external or internal, the control system, based on the feed-forward plus feedback architecture, is capable to maintain the reference target value for the two outputs of intake manifold pressure and intake mass air-flow. There is no loss in these performances if a quantization interval is added for each valve, so it is clear that disturbances and uncertainty are well handled by the control system showing its strength in dealing with them. Further improvements in performance are possible focusing on modeling and sub-region controller optimization e.g. further dividing the engine operating space.

VII. ACKNOWLEDGEMENT

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