Abstract

Emission requirements for diesel engines are becoming increasingly strict, leading to the increase of engine architecture complexity. This evolution requires a more systematic approach in the development of control systems than presently adopted, in order to achieve improved performances and reduction of times and costs in design, implementation and calibration. To this end, large efforts have been devoted in recent years to the application of advanced Model-Based MIMO control systems.

In the present paper a new MIMO nonlinear feedback control is proposed, based on an innovative data-driven method, which allows to design the control directly from the experimental data acquired on the plant to be controlled. Thus, the proposed control design does not need the intermediate step of a reliable plant model identification, as required by Model-Based methods. In this way, significant advantages over Model-Based methods can be achieved in terms of times and costs in design and deployment as well as in terms of control performances. The method is applied to the control design for the air and charging systems, using experimental data measured on a four cylinder diesel engine with single stage turbocharger. The performances of the designed controller are evaluated on an accurate nonlinear engine model, showing significant reductions of up to 2.7 times for the intake manifold pressure, up to 2.7 times for the oxygen concentration tracking errors and about 4 times in controller design and calibration efforts with respect to a decoupled-gain-scheduled PID controller typically applied for the air charging system control of diesel engines.

Introduction

In the last decades, emission requirements for diesel engines are becoming increasingly strict and at the same time, there is the necessity to improve the fuel economy. This trade-off leads to the increase of engine architecture complexity, due to the coupling of several hardware components with many interactions and highly nonlinear behaviors. In this context, the design of effective control strategies plays a crucial role, not only to fully exploit the benefits coming from the enhanced engine architectures but also to improve the engine performances and reduce the overall cost of design, deployment and calibration time.

The air and charge subsystem of a diesel engine is critical for reduction of the exhaust emissions, such as nitrogen oxides (NOx), particulate matter (PM) and CO2. On the other side, the air-path process is a quite complex system with strong couplings, actuator constraints, and fast dynamics. Furthermore, it is highly nonlinear, based on thermo dynamical phenomena. The nonlinear characteristics of the system and the actuators interactions highly sharpen the difficulties in achieving an effective control design. In addition, for better trade-off of NOx and PM emissions it is essential to decouple the air flow control (EGR or Oxygen control) and the turbocharger control (boost pressure control). In this perspective, the traditional engine control design methods, as gain scheduled PID decoupled controllers, represent a suboptimal solution, requiring substantial calibration efforts and not properly addressing the control issues. Due to the increasingly stricter emission standards, it is very hard to meet the standards without consideration of coupling between the actuators. In order to achieve more systematic development of engine control systems, allowing the combination of improved performances and reduction of times and costs in design, implementation and...
calibration, large efforts have been devoted in recent years to the application of advanced Model-Based MIMO control systems, see e.g. references \[1, 2, 3, 4, 8, 9, 10]\).

In the present paper a new MIMO nonlinear feedback control is proposed, based on the STC® (Self Tuning Control)\(^1\) data-driven method, which allows the control design directly from the experimental data acquired on the plant to be controlled, without requiring the identification of a plant model, see references \[5, 6]\). On the contrary, Model-Based methods require first to identify a reliable engine model to be used for applying the selected control method (e.g. MPC, H\(_\infty\),...), see Figure 1.

Figure 1. Model-Based vs. STC® design

Significant advantages over Model-Based methods can be achieved by STC® (Self Tuning Control) method, in terms of times and costs in design and deployment as well as in terms of control performances. In fact, derivation of engine models suitably trading between reliability and complexity, as required by Model-Based design, is time consuming, while it is avoided in STC® design. Moreover, in order to avoid too high real time computational burden, Model-Based controls are typically designed using approximate model linearization in different operating points. Indeed, STC® allows the design of MIMO nonlinear controllers suitably trading between their real time computational burden and control performances, without assuming any preassigned parametric form of involved nonlinearities.

The next sections will describe the application of STC® method for the design and validation of a MIMO controller, named STC/AC, of the Air Charging system of a GM diesel engine.

Plant Description

The system under investigation is a 1.6L four cylinder Diesel Engine with single stage turbocharger. The engine is equipped with a high-pressure EGR loop, a VGT turbine and a throttle valve. A graphical representation of the system is given in Figure 2.

Figure 2. Sketch of the diesel engine system

That system can be modeled as a sixth-order mean value model based on the gas dynamics at intake and exhaust manifolds and on the turbocharger speed variation:

\[
\dot{p}_t = \frac{RT_i}{V_i} (W_{itv} + W_{egr} - W_{e,in})
\]

\[
\dot{p}_x = \frac{RT_x}{V_x} (W_{e,in} + W_f - W_{egr} - W_t)
\]

\[
\dot{p}_{tc} = \frac{RT_{tc}}{V_{int}} (W_c - W_{itv})
\]

\[
\dot{F}_i = \frac{RT_i}{p_i V_i} (W_{egr} (F_x - F_i) - W_{itv} F_i)
\]

\[
\dot{F}_x = \frac{RT_x}{p_x V_x} \left((W_{e,in} + W_f) (F_e - F_x)\right)
\]

\[
\dot{N}_t = \frac{1}{N_t} \left(p_t - p_c\right)
\]

with the residual gas concentration at engine outlet expressed as:

\[
F_e = F_i \frac{W_{ei}}{W_{ei} + W_f} + W_f \frac{(\lambda)}{W_{ei} + W_f} + 1
\]

the engine mass flow rate calculated by the speed density equation:

\[
W_{e,in} = \eta_{v0} \frac{V_d N_t}{120RT_i} p_i
\]

1. STC® is a registered trademark of Modelway srl
and the compressor mass flow rate expressed as function of the turbine speed and the throttle upstream pressure:

\[ W_c = f(N_t, p_{ic}). \]

The turbine power can be expressed as:

\[ P_t = W_t c_p \eta_t T_x \left[ 1 - \left(\frac{p_{a}}{p_x}\right)^{\gamma - 1} \right], \]

and a similar expression can be used for the compressor:

\[ P_c = W_c c_p \frac{1}{\eta_c} T_c \left[ 1 - \left(\frac{p_{ic}}{p_x}\right)^{\gamma - 1} \right], \]

where we have assumed that the pressures upstream the compressor and downstream the turbine are close to the ambient pressure.

Instead, the flows through the EGR valve, the throttle valve and the turbine can be characterized by using the orifice equations:

\[ W_{itv} = C_d A_{itv} \frac{P_{itv,us}}{\sqrt{R_{itv,us}}} \xi \left(\frac{P_t}{p_{ic}}\right) \]
\[ W_{egr} = C_d A_{egr} \frac{P_{x}}{\sqrt{R_x}} \xi \left(\frac{P_t}{p_x}\right) \]
\[ W_t = C_d A_{vgt} \frac{P_{exh}}{\sqrt{R_{egr,lp}}} \xi \left(\frac{P_{exh}}{p_x}\right) \]

with the flow effective areas \( C_d A_{itv}, C_d A_{egr} \) and \( C_d A_{vgt} \) that depend on the positions of the corresponding actuators \( x_{itv}, x_{egr} \) and \( x_{vgt} \), and the pressure correction factors given by:

\[ \xi \left(\frac{P_{us}}{p_{ds}}\right) = \begin{cases} \left(\frac{2\gamma}{\gamma - 1}\right) \left(\frac{p_{t}}{p_{r}} - \frac{p_{r}}{p_{r}}\right) \frac{p_r}{p_{ds}} > 0.5292 \\ \left(\frac{2\gamma}{\gamma + 1}\right) \frac{p_r}{p_{ds}} \frac{p_{r}^{\gamma - 1}}{2(\gamma - 1)} \frac{p_r}{p_{ds}} < 0.5292 \end{cases} \]

where \( p_r \) is the pressure ratio between the upstream, \( p_{us} \), and the downstream pressure, \( p_{ds} \), across the orifice. The oxygen concentration \( O_2i \) is not described directly from the above six-order model, but it can be easily computed from the intake residual gas concentration as \( O_2i = 21(1 - F_i) \).

It must be remarked that these equations are not used for the STC/AC design. The above equations are reported here to evidence that the actual plant, even if only approximately described by these equations, is strongly nonlinear and characterized by interactions between the VGT, EGR and Throttle actuators.

**STC/AC Design**

As discussed in the previous section, the plant is a multiple input and multiple output (MIMO) nonlinear system, and each input affects the behavior of both the outputs, as depicted in the block diagram in Figure 3.

The control problem is formulated as a reference tracking design: find a controller that drives the EGR and VGT actuators in order to deliver the intake manifold pressure \( (p_i) \) and oxygen concentration \( (O_2i) \) as near as possible to desired reference values. The closed loop system configuration is represented in Figure 4:

In the figure above, the vector \( u \) indicates the control command \( (u_1; \text{turbine VGT position } u_{vgt}; u_2; \text{EGR position } u_{egr}) \), the vector \( v \) indicates the exogenous input \( (v_1; \text{engine speed } N_e; v_2; \text{injected fuel } W_f) \), the vector \( y \) indicates the output of the engine \( (y_1; \text{intake manifold pressure } p_i; y_2; \text{oxygen concentration } O_2i) \) and the vector \( r \) indicates the references \( r_1 \) and \( r_2 \) that the controlled engine outputs \( y_1 \) and \( y_2 \) must track. The STC/AC design and implementation process can be summarized as follows.

In the off-line design phase, a predictor \( \hat{y}(t + 1) = (\hat{y_1}(t + 1), \hat{y_2}(t + 1)) \) of the output \( y(t + 1) = (y_1(t + 1), y_2(t + 1)) \) is computed from experimental engine data of the form:

\[ \hat{y}(t + 1) = f(u(t), q(t)) \]  

where:

\[ q(t) = (y_1(t), \ldots, y_1(t - n), y_2(t), \ldots, y_2(t - n), v_1(t), \ldots, v_1(t - n), v_2(t), \ldots, v_2(t - n)) \]

\( n \) is the predictor memory and \( f \) is a vector-valued polynomial function. These variables are obtained from experimental data by means of convex optimization algorithm described in [6].
The online STC/AC implementation is performed by computing, at each time \( t \), a command \( u^*(t) = (u_1^*(t), u_2^*(t)) \) is looked for, such that the predicted model output \( \hat{y}(t + 1) \) is as close as possible to the desired reference value \( r(t + 1) = (r_1(t + 1), r_2(t + 1)) \). This command is obtained solving the following optimization problem:

\[
  u^*(t) = \arg \min_{u \in U} \| (z, q(t), r(t + 1)) \|_w^2 + \mu \| z \|_2^2
\]

where \( \| \cdot \| \) is the vector 2-norm, \( U \) is the input domain and \( w, \mu \geq 0 \) are two design parameters: \( w \) determines the trade-off between the two output tracking precisions; \( \mu \) determines the trade-off between tracking precision and command activity.

The data used for the control design have been acquired from typical tests used to characterize the engine dynamics and to evaluate control performances. In particular the following tests have been used:

1. NEDC cycle: New European Driving Cycle, is an homologation cycle for light-duty vehicles, used in Europe, for determining the emissions levels and fuel consumption until Euro6. It is composed by an urban driving part and an extra urban part.
2. WLTP cycle: Worldwide harmonized Light vehicles Test Procedures. It is a new standard homologation cycle that will replace the NEDC. It is more dynamic than the NEDC, covering a wide range of driving situations from urban traffic to motorway.
3. Dynamic tests: non standard tests performed in order to excite a wide range of the system dynamics;
4. Transient tests: transient between fixed engine operating points (engine speed and injected fuel).

The measurements have been acquired with a sample time of 100 ms, and are composed of the input-output signals reported in Table 1 and Table 2.

<table>
<thead>
<tr>
<th>Name</th>
<th>Variable</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>u1</td>
<td>( \mu_{\text{VGT}} )</td>
<td>[%]</td>
<td>Turbine VGT position</td>
</tr>
<tr>
<td>u2</td>
<td>( \mu_{\text{EGR}} )</td>
<td>[%]</td>
<td>EGR valve position</td>
</tr>
<tr>
<td>v1</td>
<td>( N_e )</td>
<td>[rpm]</td>
<td>Engine speed</td>
</tr>
<tr>
<td>v2</td>
<td>( W_f )</td>
<td>[mm/str]</td>
<td>Injected fuel</td>
</tr>
</tbody>
</table>

Table 2. Output signals of the engine model

<table>
<thead>
<tr>
<th>Name</th>
<th>Variable</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>y1</td>
<td>( p_i )</td>
<td>[kPa]</td>
<td>Intake manifold pressure</td>
</tr>
<tr>
<td>y2</td>
<td>( O_2_{i} )</td>
<td>[%]</td>
<td>Intake manifold oxygen concentration</td>
</tr>
</tbody>
</table>

Data have been divided into “design dataset” (data used to design the control) and “validation dataset” (data used to verify the control) as shown in Table 3:

<table>
<thead>
<tr>
<th>Test</th>
<th>Design</th>
<th>Validation</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEDC Cycle</td>
<td>0 &lt; t &lt; 200, 800 &lt; t &lt; 1200</td>
<td>200 &lt; t &lt; 800</td>
</tr>
<tr>
<td>WLTP Cycle</td>
<td>0 &lt; t &lt; 200, 1000 &lt; t &lt; 1800</td>
<td>200 &lt; t &lt; 1000</td>
</tr>
<tr>
<td>Dynamic1</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Dynamic2</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Dynamic3</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Dynamic4</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Dynamic5</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Transient1500</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Transient2000</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Transient2500</td>
<td>X</td>
<td></td>
</tr>
</tbody>
</table>

An STC/AC has been designed and implemented according the above described procedure, having filter memory \( n = 3 \), as obtained from the algorithm used for designing the predictor (1). The values of the two design parameters in (3) have been set to \( w = 20, \mu = 0.04 \), giving a most acceptable trade-offs between the two output tracking precisions and command activity.

The computing time for the online implementation of the STC/AC is the one needed to solve the optimization problem (3). The average time taken by a Matlab m-function implementation on a laptop with an i7 3Ghz processor and 16 MB RAM is about 1.6e-3 s (average computed over 5000 runs of a Monte Carlo simulation). The average time taken by a compiled Simulink mex implementation on the same laptop is about 1.4e-4 s (average computed over 1000 runs of a Monte Carlo simulation). Even shorter times are expected for an implementation of the control algorithm on a real-time CPU.

STC/AC Performances Verification

A first verification of the performances achieved by the STC/AC has been performed on a engine model identified as described below. Indeed the STC/AC is designed in a Matlab/Simulink® environment still ready to be implemented in the ECU (Engine Control Unit) and to be tested on the test cell and vehicle.

Model Identification

In order to face the complexity of the modeling problem due to strong nonlinearities and complex input-output correlations, the NOSEM® (NOnlinear Set-membership Modeling) method has been used. This data-driven identification technology does not need deep and detailed first principle laws investigations (see Figure 5).
The main features of the NOSEM® method are:

1. It allows one to handle systems whose physical description is not well known or too complex;
2. The model accuracy can be evaluated by means of tight error bounds;
3. The search for a parametric form of the involved nonlinearities is not required;
4. It allows significant reduction in times and costs of model identification.

For more details see reference [7]. In the following the identified engine nonlinear model is named NOSEM/AC (NOSEM/Air Charging).

The NOSEM/AC model has been designed from the experimental data acquired in the same tests used to design the STC/AC. However, a different organization of these data has been considered for model identification and cross-validation, see Table 4.

<table>
<thead>
<tr>
<th>Test</th>
<th>Design [s]</th>
<th>Validation [s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEDC Cycle</td>
<td>0 &lt; t &lt; 400</td>
<td>400 &lt; t &lt; 1000</td>
</tr>
<tr>
<td></td>
<td>1000 &lt; t &lt; 1200</td>
<td></td>
</tr>
<tr>
<td>WLTP Cycle</td>
<td>0 &lt; t &lt; 600</td>
<td>600 &lt; t &lt; 1400</td>
</tr>
<tr>
<td></td>
<td>1400 &lt; t &lt; 1800</td>
<td></td>
</tr>
<tr>
<td>Dynamic1</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Dynamic2</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Dynamic3</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Dynamic4</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Dynamic5</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Transient1500</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Transient2000</td>
<td>X</td>
<td></td>
</tr>
<tr>
<td>Transient2500</td>
<td>X</td>
<td></td>
</tr>
</tbody>
</table>

The NOSEM/AC computes the outputs $y_1$ and $y_2$ as non linear functions of the past values of inputs and outputs of the forms:

\[
y_1(t+1) = g_1(y_1(t), ..., y_1(t-4), u_1(t), ..., u_1(t-5), u_2(t), \]
\[
  ..., u_2(t-5), v_1(t), ..., v_1(t-5), v_2(t), ..., v_2(t-5))
\]
\[
y_2(t+1) = g_2(y_2(t), ..., y_2(t-4), u_1(t), ..., u_1(t-5), \]
\[
  u_2(t), ..., u_2(t-5), v_1(t), ..., v_1(t-5), v_2(t), ..., v_2(t-5))
\]

The functions $g_1$ and $g_2$ are identified from experimental data as described in [2].

The performances of the identified NOSEM/AC are shown in Table 5, where the mean, the standard deviation (std) and the 90% confidence interval (CI90) of percent modeling errors are calculated on the validation dataset described in Table 4.

<table>
<thead>
<tr>
<th>Test</th>
<th>Output</th>
<th>Mean</th>
<th>Std</th>
<th>CI90</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEDC Cycle</td>
<td>$p_i$</td>
<td>-0.74%</td>
<td>0.83%</td>
<td>±2%</td>
</tr>
<tr>
<td>NEDC Cycle</td>
<td>$O_2$</td>
<td>0.09%</td>
<td>0.65%</td>
<td>±1%</td>
</tr>
</tbody>
</table>
| WLTP Cycle | $p_i$  | -0.77% | 1.22% | ±2.25%
| WLTP Cycle | $O_2$  | -0.07% | 1.23% | ±1.5% |
| Dynamic1   | $p_i$  | -1.67% | 2.83% | ±5%   |
| Dynamic1   | $O_2$  | 0.32%  | 2.52% | ±2.5% |
| Dynamic5   | $p_i$  | -1.77% | 2.64% | ±5%   |
| Dynamic5   | $O_2$  | 0.29%  | 1.95% | ±2.75%|
| Transient1500 | $p_i$ | -2.06% | 1.03% | ±3.25%|
| Transient1500 | $O_2$ | 0.17%  | 1.53% | ±1.75%|
| Transient2500 | $p_i$ | -2.42% | 2.58% | ±5.5% |
| Transient2500 | $O_2$ | 0.18%  | 0.90% | ±1.25%|

It may be worth noting that the NOSEM/AC model achieves significantly better model accuracies over a previously developed mean value model, based on the equations reported in the section Plant Description. In particular:

- for the homologation NEDC and WLTP cycles, the 90% confidence intervals CI90 of the modeling errors are reduced by up to 3.5 times for the intake manifold pressure and up to 12 times for the oxygen concentration.
- for the other data set, the 90% confidence intervals CI90 of the modeling errors are reduced by up to 8 times for the intake manifold pressure and up to 6 times for the oxygen concentration.

For the homologation cycles (NEDC and WLTP) are now reported the figures showing the modeled outputs $p_i$ and $O_2_i$ compared with measures. For confidentiality reasons the values of y axis of each figure are expressed as percent values with respect to maximum value.
In this section the performance achieved on the NOSEM/AC model by the designed STC/AC controller are presented and are compared with those achieved by a decoupled-gain-scheduled PID controller typically applied for the air charging system control of diesel engines. More precisely, this PID controller consists of a feedback control plus a feedforward part. The closed loop regulators are PIDs gain scheduled and the feedforward is implemented through steady state maps. The controllers gains, also, are described by map functions of engine operating points. The coupling between the subsystems is not considered and in certain areas, to avoid interactions, only one controller is active, while the other actuators are controlled only by the feedforward part.

Controllers performance are evaluated considering the percent tracking errors between the reference signals and the corresponding close-loop outputs. Table 6 reports the results obtained by the STC/AC controller. In particular, the mean, the standard deviation (std) and the 90% confidence interval (CI90) reported are computed on the validation datasets described in Table 3.

<table>
<thead>
<tr>
<th>Test</th>
<th>Output</th>
<th>Mean</th>
<th>Std</th>
<th>CI90</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEDC Cycle</td>
<td>p,i</td>
<td>1.09%</td>
<td>1.49%</td>
<td>±3%</td>
</tr>
<tr>
<td>NEDC Cycle</td>
<td>O2,i</td>
<td>0.32%</td>
<td>1.23%</td>
<td>±2%</td>
</tr>
<tr>
<td>WLTP Cycle</td>
<td>p,i</td>
<td>0.89%</td>
<td>2.73%</td>
<td>±4.5%</td>
</tr>
<tr>
<td>WLTP Cycle</td>
<td>O2,i</td>
<td>0.25%</td>
<td>1.98%</td>
<td>±3.5%</td>
</tr>
<tr>
<td>Dynamic2</td>
<td>p,i</td>
<td>-1.29%</td>
<td>2.83%</td>
<td>±5%</td>
</tr>
<tr>
<td>Dynamic2</td>
<td>O2,i</td>
<td>-0.53%</td>
<td>2.32%</td>
<td>±3.5%</td>
</tr>
<tr>
<td>Transient2000</td>
<td>p,i</td>
<td>-1.71%</td>
<td>2.46%</td>
<td>±4.5%</td>
</tr>
<tr>
<td>Transient2000</td>
<td>O2,i</td>
<td>-0.57%</td>
<td>2.12%</td>
<td>±3.5%</td>
</tr>
<tr>
<td>Transient2500</td>
<td>p,i</td>
<td>0.83%</td>
<td>1.29%</td>
<td>±2%</td>
</tr>
<tr>
<td>Transient2500</td>
<td>O2,i</td>
<td>0.38%</td>
<td>1.27%</td>
<td>±1.5%</td>
</tr>
</tbody>
</table>

In comparison with the PID controller, the STC/AC has provided significant improvements in control tracking performance, that can be summarized as follows:

- on average over the considered five testing data, the 90% confidence intervals are almost halved for both the intake manifold pressure and for the oxygen concentration.
- the maximum of the 90% confidence intervals of the intake manifold pressure tracking errors achieved by the STC/AC and PID controllers are 5% and 12%, respectively.
- the maximum of the 90% confidence intervals of the oxygen concentration tracking errors achieved by the STC/AC and PID controllers are 3.5% and 12%, respectively.

The reasons of such improvements have to be ascribed to two main reasons now briefly described. On one hand, the STC method allows to systematically consider the nonlinearities and couplings typical of a nonlinear MIMO system. On the other hand the PID approach doesn't take in account the strong couplings among the input and output signals and the system nonlinearities are approximated by linearization in several working points. Moreover, from the calibration point of view, decoupled-gain-scheduled PID controller requires a large calibration effort, needed by the large number of PID controllers to be tuned. Using STC approach, only the two design parameters w and μ have to be tuned, in order to allow an appropriate trade-off between the two output tracking precisions and command activity. Thus, in general, the control calibration efforts evaluated to be about three-four times less than required by the conventional decoupled-gain-scheduled PID controller.

As example of the STC/AC validation test performed, Figure 8 and Figure 9 present the homologation cycles (NEDC and WLTP, respectively), showing the controlled outputs p_i and O2_i compared with the references. For confidentiality reasons the values of y axis of each figure are expressed as percent values with respect to maximum value.
In order to assess the control actions toward the engine VGT and EGR actuators, in Figure 10 and Figure 11 the percentage values of the STC/AC commands are presented.

**Summary/Conclusions**

In the paper the innovative MIMO nonlinear data-driven STC method is applied to the control of the air and charging systems of a four cylinder diesel engine with single stage turbocharger. The STC control is directly designed from the experimental data typically used the engine dynamics characterization and for control performance evaluation. Thus, the STC method, avoiding the step of engine model identification, achieves significant reduction of times and costs in design, deployment and calibration of the controller with respect to the traditional engine control design methods, as well as to the advanced model-based methods investigated in recent years. Moreover, the STC method systematically accounts for the nonlinearities and coupling of nonlinear MIMO systems, not requiring parametrized approximation of nonlinearities (e.g. linearization on different working points), thus allowing to achieve improved performances over traditional control systems. Control performances, measured as errors between the two controlled output variables and the corresponding tracking references, have been evaluated on a accurate engine model, and compared to a decoupled-gain-scheduled PID controller. In particular, on average over the validation data, the 90% confidence intervals of the tracking errors are almost halved for both the intake manifold pressure and for the oxygen concentration, while the maximum have been reduced by about three times.

Furthermore, the STC/AC allow to easily tune the outputs tracking errors and the command activities. Indeed, the STC approach, having only two tuning parameters, requires a control calibration effort which is 3-4 times less than that required by the conventional decoupled-gain-scheduled PID approach.

It can be noted that no robustness investigation of the designed control system vs. ambient temperature and pressure has been performed, since the available experimental data used for the STC design have been measured at constant values for these variables. However, the STC method may allow a systematic robust design, if experimental data for these variables are measured in the ranges of interest.

As indicated by the results reported above, the innovative STC method presented in this paper appears to allow the systematic design of efficient MIMO nonlinear control systems as required in many engine problems, with significant reduction in development times. In order to consolidate these results, future work is planned to extensively test the developed controller for the air charging system on test cell and vehicle and to evaluate the scalability of the method to support more complex engine architecture, e.g. dual loop EGR engines.

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**References**


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Definitions

\( x_{th} \) - Intake throttle valve position.
\( x_{egr} \) - EGR valve position.
\( x_{vgt} \) - Turbine VGT position.
\( p \) - Intake Manifold pressure.
\( O_2 \) - Intake manifold Oxygen concentration.
\( p_s \) - Exhaust manifold pressure (upstream turbine).
\( p_{ic} \) - Upstream compressor pressure.
\( F_i \) - Residual gas fraction at intake manifold.
\( F_s \) - Residual gas fraction at exhaust manifold.
\( F_e \) - Residual gas concentration at engine outlet.
\( N_t \) - Turbocharger speed.
\( W_{egr} \) - EGR mass flow rate.
\( W_{th} \) - Throttle valve mass flow rate.
\( W_{vgt} \) - VGT mass flow rate.
\( P_t \) - Turbine power.
\( P_c \) - Compressor power.
\( T_{egr} \) - EGR gas temperature.
\( T_i \) - Intake manifold gas temperature.
\( T_s \) - Exhaust manifold gas temperature.
\( T_{ic} \) - Air temperature after intercooler.
\( W_{e, in} \) - Engine in mass flow rate.
\( V_i \) - Intake manifold volume.
\( V_{int} \) - Intercooler volume.
\( V_s \) - Exhaust manifold volume.
\( \gamma \) - Ratio of specific heats.

\( R \) - Universal gas constant.
\( J_t \) - Turbocharger moment of inertia.
\( N_e \) - Engine speed.
\( c_p \) - Specific heat capacity of air at constant pressure.
AFRs - Air fuel ratio.
\( \eta_v \) - Volumetric efficiency.
\( V_d \) - Displacement
\( p_a \) - Ambient pressure
\( W_c \) - Compressor mass flow rate
\( W_f \) - Fuel mass flow rate