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Observer Design for Torque Balancing on a DI Engine

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ABSTRACT

Torque balancing for diesel engines is important to eliminate generated vibrations and to correct injected quantity disparities between cylinders. The vibration phenomenon is important at low engine speed and at idling. To estimate torque production from each cylinders, the instantaneous engine speed from the crankshaft is used. Currently, an engine speed measurement every 45° crank angle is sufficient to estimate torque balance and to correct it in an adaptive manner by controlling the mass injected into each cylinder.

The contribution of this article is to propose a new approach of estimation of the indicated torque of a DI engine based on a nonstationary linear model of the system. On this model, we design a linear observer to estimate the indicated torque produced by each cylinder. In order to test it, this model has been implemented on a HiL platform and tested on simulation and with experimental data .

INTRODUCTION

Torque balancing for diesel engines is a major feature of modern engine controllers for two main reasons:

- Diesel injection systems introduce a bias in injected masses between cylinders (relative errors can be up to 20%). Torque production is a monotonic function of the fuel mass injected into each cylinder.
- This torque variation is thus a periodic time function and generates a vibration mode in the engine.

This phenomenon is significant at low engine speed and at idling conditions. For diagnostic purposes, the correct combustion must be monitored during engine operation. In case of misfiring, unburned gases are generated and the engine does not meet the legal limitation for HC emissions. We tried to estimate the torque produced by combustion while we only measuring the instantaneous angle speed on the crankshaft. Currently, an engine speed measurement every 45° of crankshaft rotation is sufficient to estimate torque balance and to correct it in an adaptive manner by controlling the mass injected into each cylinder.

This problem has been studied before, but the different approaches where frequential approaches or founded on the derived measures [7]. The main advantage of our approach is to use a state-space model and control techniques to solve this problem.

We describe the various torques (combustion, load, friction, ...) arising in the dynamics. This allows us to observe the instantaneous indicated torque. The objective of our work is to build an accurate observer of torque balance in transient mode (according to torque and engine speed). In Section 2, we describe the experimental setup. We explain the engine dynamics in Section 3 and the approximation we made to obtain a linear equation in Section 4. We then introduce the notion of residue in Section 5. In Section 6 , we describe our observer design and gives details of the results obtained.

EXPERIMENTAL SETUP FOR CONTROL

The synthesis of the state-model based observer-controller was first validated on a 4 cylinder DI engine model. This reference model relies on the Chmela combustion formula (heat production is proportional to the amount of injected fuel and to the burning velocity)[1]. This model includes combustion and a transformation into indicated torque and effective torque. The transmission is modeled as in [1]. A more realistic model with nonlinearities and imperfections of the flywheel will be integrated in future work. The observer is fed with instantaneous engine speed measurements coming from the reference model and signals from the test bench.

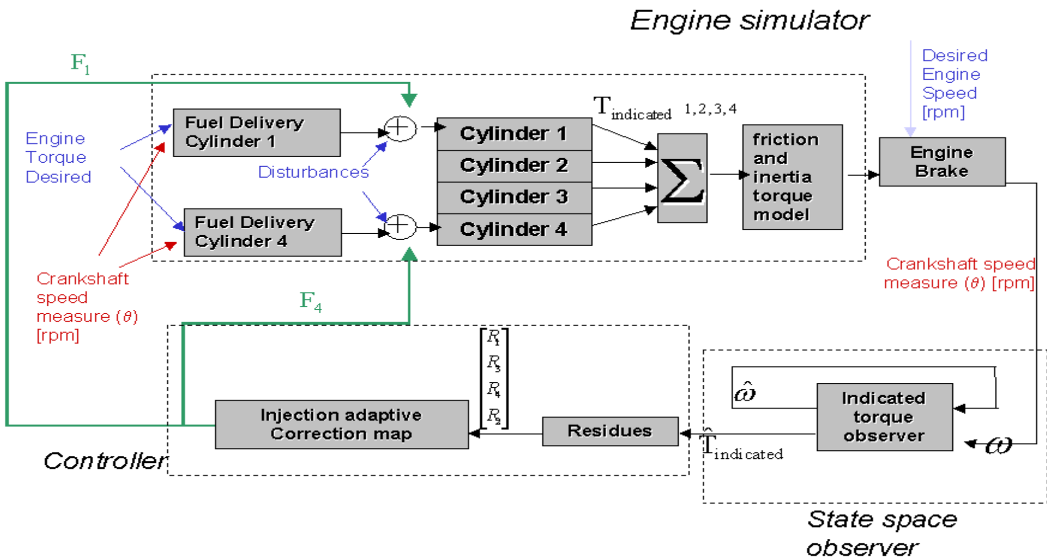

Figure 1 : GLOBAL SCHEME

The complete simulation was implemented in real time with the xPCTarget Hardware in the Loop platform and will finally be implemented on the motor bench through a fast prototyping system.

The synopsis of the complete simulation is described in Figure 1. We implemented the simulation model (reference model + observer + controller) on a real time simulator based on xPCTarget [2]. Real time is not achieved due to the complexity of the reference model. With a 1 GHz Pentium based computer, 1 second of engine real time is simulated in 30 seconds. Nevertheless, this allows extensive simulation and parameter sensivity studies: pole placement calibration with a robustness study with noise on engine speed measurements, change of load…

This HiL platform was transformed to a fast prototyping system by adding specific input/output boards. The same code can be implemented in the control system and on the test bench.

ENGINE DYNAMICS

The crankshaft velocity is the only measured quantity and we want to reconstruct the combustion torque. Following [3] we express the dynamics. With the kinetic energy theorem, the torque balance on the crankshaft is expressed as

$$\frac{d}{d\alpha}\left(\frac{1}{2}J(\alpha)\dot{\alpha}^2\right)=T_{comb}(\alpha)-T_{load}(\alpha)-T_{fric}(\alpha)$$

Let $T_{mass}=T_{comb}-T_{load}-T_{fric}$ denote the mass torque, the combustion torque is also referred to as indicated torque. We consider that the load torque T_{load} is a low frequency signal (we assumed that the variation of the load torque between two TDC is small, we made a linear interpolation to have the value between two TDC) and that friction torque is a quadratic in $\frac{d\alpha}{dt}$ [4]. We can split the mass torque into two terms

$$T_{mass}=J\ddot{\alpha}+\frac{1}{2}\frac{dJ}{d\alpha}\dot{\alpha}^2$$

The first term represents the oscillating masses and the second one stands for the rotating masses. We used a model of two masses connected by a single rod as in [5]. The total rod mass m_{rod} is

- An oscillating part $m_{rod,osc}=m_{rod}\frac{l_{osc}}{l}$
- A rotating part $m_{rod,rot}=m_{rod}\frac{l_{rot}}{l}$

The two lengths l_{rot} and l_{osc} with $l=l_{rot}+l_{osc}$ defined by the position of the center of gravity of the connecting rod. The oscillating mass is

$$m_{osc} = m_{piston} + m_{rod}\,\frac{l_{osc}}{l}$$

and the rotational mass on each cylinder is:

$$\frac{m_{osc}}{CYL} = \frac{m_{crank}}{CYL} + m_{rod}\,\frac{l_{rot}}{l}$$

The crankshaft mass is deduced from the moment of inertia $m_{crank} = J_{crank} r^2$. Straightforward calculations yield

$$T_{mass} = J\ddot{\alpha} + \frac{1}{2}\frac{dJ}{d\alpha}\dot{\alpha}^2$$

with

- $J(\alpha) = m_{rot} r^2 + m_{osc} \sum_{j=1}^{CYL} (\frac{ds_j}{d\alpha})^2$
- $\frac{dJ}{d\alpha} = 2 m_{osc} \sum_{j=1}^{CYL} (\frac{ds_j}{d\alpha})(\frac{d^2 s_j}{d\alpha^2})$

DISCRETE NONSTATIONARY LINEAR APPROXIMATION

The torque balance is an angle-dependent differential equation with time-derivatives

$$J(\alpha)\ddot{\alpha} = T_{comb}(\alpha) - T_{load}{}^*(\alpha) - f(\alpha)(\dot{\alpha})^2$$

with

- $f(\alpha) = \frac{1}{2}\frac{dJ}{d\alpha}$
- $T_{load}^* = T_{load} + T_{fric}$

where T_{load}^* refers to the extended torque. We can now reformulate the second derivative of the crankshaft angle to have

$$\dot{\alpha}\frac{d\dot{\alpha}}{d\alpha} = \frac{1}{J(\alpha)}(T_{comb}(\alpha) - T_{load}{}^*(\alpha) - f(\alpha)(\dot{\alpha})^2)$$

Using a first order approximation on the integration of the previous equation, we can break the dependence on time and on crankshaft angle and only save the square of the crankshaft angle speed as a nonlinearity.

$$\dot{\alpha}_{n+1}^2 - \dot{\alpha}_n^2 \approx \frac{2\Delta\alpha}{J(n)}(T_{comb}(n) - T_{load}{}^*(n) - f(n)\dot{\alpha}_n^2)$$

We use $\Delta\alpha = 6°$ as angular step. The model is then discretized with respect to angle steps instead of time steps. Using the square of the crankshaft angle speed $\dot{\alpha}^2$ as the first state variable x_1, we get the time varying linear equation

$$x_1(n+1) = (1 - \frac{2\Delta\alpha}{J(n)} f(n)) x_1(n) + \frac{2\Delta\alpha}{J(n)} x_2(n)$$

$$\text{where } \begin{cases} x_1(n) = \dot{\alpha}_n^2 \\ x_2(n) = T_{comb}(n) - T_{load}^*(n) \end{cases}$$

This formulation of the problem leads to classical methods of estimation for the indicated torque.

NOTION OF RESIDUES AND USE FOR TORQUE BALANCING

The notion of residue will provide us with an accurate representation of the balance. It is the relative error of the effective work. It is expressed as

$$R_j = \frac{\frac{CYL}{4\pi}\int_{\alpha_i - \frac{2\pi}{CYL}}^{\alpha_i + \frac{2\pi}{CYL}} T_{ind}\,dt - \frac{1}{4\pi}\int_0^{4\pi} T_{ind}\,dt}{\frac{1}{4\pi}\int_0^{4\pi} T_{ind}\,dt}$$

where the angle on the crankshaft α_i represents the point between the TDC and the BDC of cylinder i. By construction

$$\sum_{j=1}^{CYL} R_j = 0$$

The residues are a convenient measurement of the stroke balance. When the system is perfectly balanced the residues are 0. A positive residue in cylinder *i* reveals that this cylinder generates more torque than the average of the 4 cylinders. This imbalance can be corrected by injecting less fuel into cylinder *i*. If convergence time is not an issue, a simple integral controller will correct cylinder i.

OBSERVATION OF THE INDICATED TORQUE

Our model has a two dimensional state and one output (crankshaft speed). So far, we have exhibited only one evolution equation. It must be complemented with another equation reflecting the evolution of the indicated torque.

As a first try we considered that the variation of the indicated torque during one angular step is small.

PREDICTION MODEL

We used as a second equation

$$\frac{dx_2}{d\alpha} = 0$$

Between two angular steps we obtained

$$\begin{cases} x_1(n+1) & = & (1-\frac{2\Delta\alpha}{J(n)} f(n))x_1(n) + \frac{2\Delta\alpha}{J(n)} x_2(n) \\ x_2(n+1) & = & x_2(n) \end{cases}$$

The output is $x_1(n)$. We have a nonstationary linear system which can be expressed as

$$\begin{cases} X_{k+1} = A_k X_k \\ Y_k = CX_k \end{cases}$$

To solve the observation problem, we perform a gain scheduling. Even if each system (A_k, C) is stable, the nonstationary system may not be stable. Using the same point of view as [6] with the lifting technique, we realized that the dynamics are periodic. So we had just to check stability for $\prod_{k=1}^{T}(A_k - L_k C)$. For this case, one can numerically investigate the convergence of the system. With an appropriate gain choice, the eigenvalues of $\prod_{k=1}^{T}(A_k - L_k C)$ can be set to $\left[0 \quad 10^{-4}\right]$

which proves the convergence of the observer. A question of practical interest is the design of the gain scheduling that would give improved convergence. This is a point we are focusing on.

SIMULATION RESULTS

The next figures display :

- ref : reference
- pp1 : result of the first observer with pole placement

The simulation is at 1000 rpm with an BMEP of 2 bar. In the simulation, we introduced offsets on the mass injected in each cylinder.

- Cylinder 1: 20% of the reference mass
- Cylinder 2: 0% of the reference mass
- Cylinder 3: -20% of the reference mass
- Cylinder 4: 5% of the reference mass

Some noise was added to reproduce disturbances due to the flywheel imperfection.

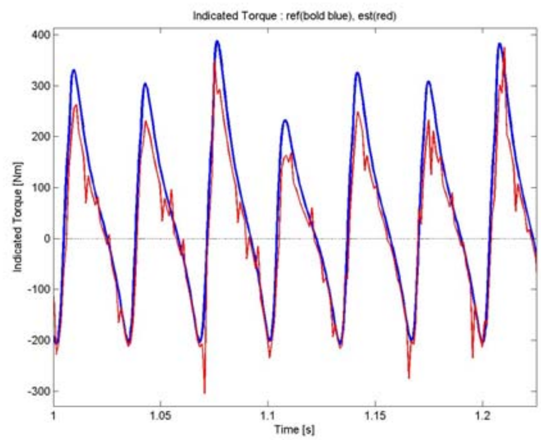

Figure 2 : Indicated torque (simulation)

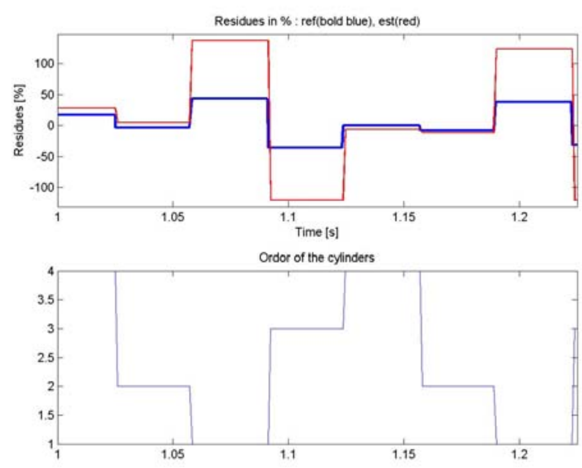

Figure 3 : Residues (simulation)

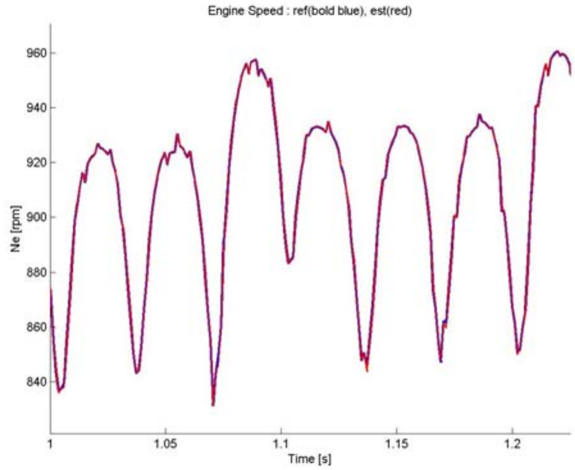

Figure 4 : Engine speed (simulation)

RESULTS FROM THE BENCH

The next figures display the result of the estimator on experimental data. We reconstruct the indicated torque from the bench with the in-cylinder pressure and we test the observer on the flywheel velocity measurement. The setting point is not the same as the simulation one.

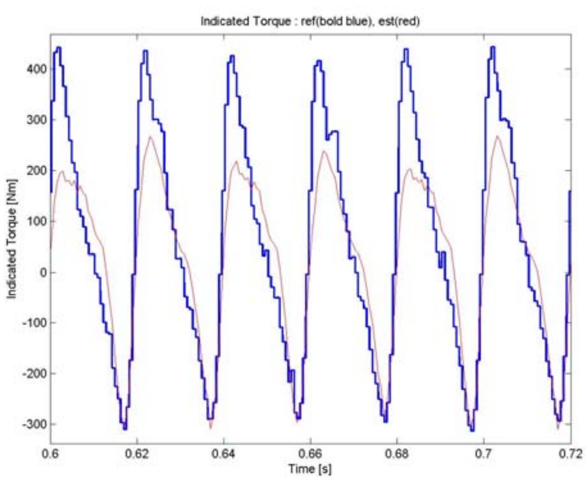

Figure 5 : Indicated torque (bench)

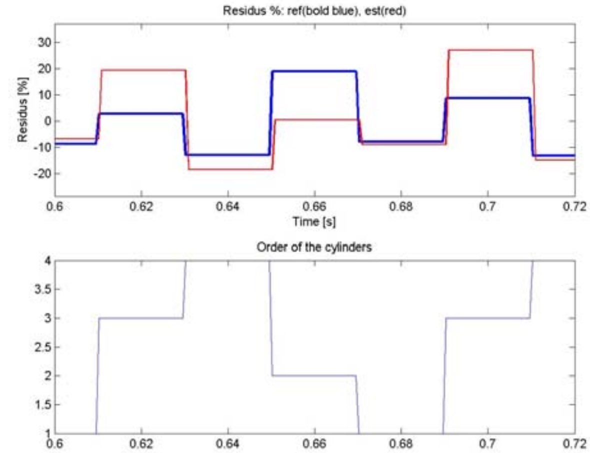

Figure 6 : Residues (bench)

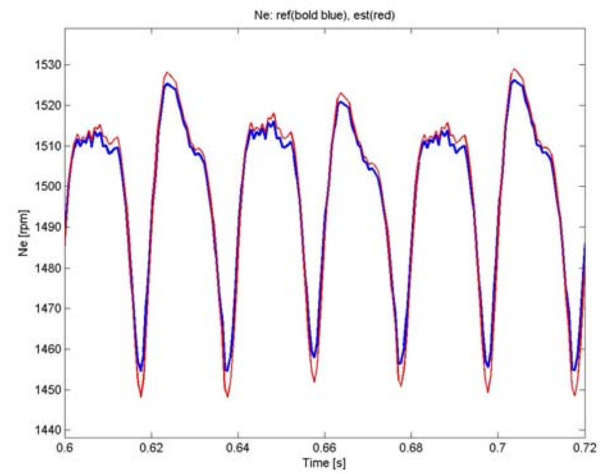

Figure 7 : Engine speed (bench)

COMMENTS

Simulation and bench results are consistent. The simulation and the result on the bench are satisfactory. Qualitatively, the residues are well estimated. For example, Figure 6 clearly shows that cylinder 1 had too much fuel injected. On the other hand the quantitative prediction is not accurate. Fortunately, in order to reduce the residues to zero, only the qualitative information matters: simple PID would work on this fully actuated system provided the sign of gains are known, i.e. the sign of residues. The idea behind this observer is the simplest possible as we introduced no knowledge about the system. Using a physics based second equation, we are now aiming at increasing the quantitative quality of the observation.

CONCLUSION AND FUTURE WORK

This article deals with a simple observer design for torque balancing on DI engine.

After explaining the engine dynamics in Section 3 and the model we use in Section 4, we explored a way to describe the evolution of the indicated torque and showed the results on simulation and on the bench (Section 6). So far we have had some interesting qualitative results on the estimation of the torque balance, though their quantitative accuracy can be improved.

Future directions are as follows. For the control design method, we have to evaluate the strengths and weaknesses of a method based on harmonic decomposition. Moreover, we have to take into account the flying wheel imperfections in an adaptive algorithm. For physics modeling, we need a more accurate model of transmission and its nonlinearities. This part will be required for less stiff transmission of vehicle. To take full advantage of such nonlinear time-varying models, an appropriate observer design will also have to be developed.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

Symbol / Units / Physical Variables

BMEP/ [bar] / Break Mean Effective Pressure
BTC/ / Bottom-Top Center
CYL / / Number of cylinders
J / [kg m²] / Moment of inertia
J_{crank} / [kg m²] / Moment of inertia of the crankshaft
l / [m] / Connecting rod length
m_{crank} / [kg] / Crankshaft mass
m_{osc} / [kg] / Oscillating mass
m_{rod} / [kg] / Rod mass
$m_{rod,osc}$ / [kg] / Oscillating rod mass
$m_{rod,rot}$ / [kg] / Rotating rod mass
r / [m] / Radius of the crankshaft
R_j / / Residue of cylinder number j
s / [m] / Piston stroke
s_j / [m] / Piston stroke of cylinder number j
TDC/ / Top-Down Center
T_{comb} / [Nm] / Combustion torque
T_{fric} / [Nm] / Friction torque
T_{ind} / [Nm] / Indicated torque
T_{load} / [Nm] / Load torque
T^*_{load} / [Nm] / Extended load torque