SAE International

Optimal Idle Speed Control of a Light Duty Turbodiesel Engine With the Aim of Minimizing Fuel Consumption

2011-01-0713
Published
04/12/2011

Kamyar Nikzadfar and Amir Hossein Shamekhi
Khaje Nasir Univ. of Technology

doi: 10.4271/2011-01-0713

ABSTRACT

Idle Control of Internal Combustion Engines (ICEs) is an important mode of Engine Management Systems (EMS) due to its effects on Fuel Consumption (FC) and pollutions produced in urban traffics. In this paper an optimal controller is used to control the idle speed with respect to minimization of FC in transient situations. In order to design an optimal controller a state space representation of engine in demanded. Subspace identification method is used to derive a proper state space model of engine around idle speed state. The required input/output data is generated using a mean value model. A finite time optimal controller is designed with the aim of minimizing both fuel consumption and speed fluctuations. The influence of finite time value on gross fuel consumption is then studied under various weighting factors. It is shown that the fuel consumption is sensitive to the controlling time duration for finite time LQR.

INTRODUCTION

Increasing demands for more efficient engines and stricter standard limits on exhaust gas pollutions require more accurate control on engine operating parameters. First generations of internal combustion engines used simple mechanical control instruments such as governors to regulate their operational status. Invention of semiconductors which leads to advent of microcontrollers made it possible to implement more complicated control strategies on mechanical systems, one of which is internal combustion engine. On the other hand new engines are provided with more actuators which in turn increase the number of manipulated parameters. These all have made it possible to decrease both pollution and fuel consumption of engines.

New generation of diesel engines are provided with EMS which controls the injection of fuel, boost pressure and many other operational parameters. Automotive Idle Speed Controller (ISC) is an important function in engine control field. Engine high nonlinearities in low speeds makes designing of this controller complicated. About 30% of FC during a city driving cycle is spent at idle mode [1]. ISC should be able to keep engine speed in a narrow band close to designed idle speed while exerting external load such as A/C load, alternator load and steering hydraulic pump load. ISC should maintain engine speed in a limited range; lowering speed would cause instability in engine operation while increasing speed from desired speed would increase FC.

A complete review on ISC has been done by Zhengmao [2]. Different methods of modeling, identification and design of ISC have been reviewed in his paper. Many control methods have been used to control idle speed. The first generation of diesel engines used simple mechanical controller mechanisms named governor to regulate their speed. Nowadays with the aid of microcontrollers PID controllers are used due to their robustness and simplicity. However more complex control methods such as adaptive control, optimal control and robust control have been investigated to regulate engine speed in idle mode, most of them were implemented on spark ignition engines. [2] Discrete time control has been used to control a turbocharged diesel engine. In this case the discrete time controller is designed using reference governor for the ISC system. [4] Also an adaptive control algorithm has been used to control a heavy duty diesel engine [5]. Javiar and et.al. used an automatic tuning method for calibrating of a PID controller to increase the performance of a PID base ISC controller for a diesel engine [6]. In their research they used a close-loop identification method to estimate the parameters of engine based on a first order system with time delay and used the estimated parameters to tune the PID controllers. Mills used a multivariable LQG control algorithm for regulating the idle speed of a diesel engine, in his research a nonlinear engine model was used to design and test the controller [7]. Also Han and et.al. used a non-linear model of a diesel engine and linearized it around idle speed to design an optimal controller for regulating the engine speed. They estimated the model parameters using nonlinear identification method based on numerous tests done on the engine. They could successfully control the engine [8].

In this paper a linear state space identification method is used based on input/output data which are generated from a mean value model of a turbocharged engine. 4 states are used to develop the model. A finite time LQR controller is used then to control the engine speed with aiming at minimizing the fuel economy.

DIESEL ENGINE DYNAMIC MODELING

In order to identify the engine dynamics a comprehensive model of engine is needed. The model should also be accurate enough to predict engine behavior under operational conditions. Many researches have been done to simulate the engine dynamic behavior. In order to simulate the dynamic performance of engine, Mean Value Model (MVM) is utilized. To model the engine performance required information such as turbocharger maps, torque generation efficiency, geometric and dynamic characteristics of engine are collected from the test results obtained by Hendricks and et al. [9]. The model contains inlet and exhaust manifolds, turbocharger, torque generation, internal friction and crank shaft dynamic sub models. Figure 1 illustrates the numbering sequence of engine. A brief description of engine modeling procedure is mentioned in following paragraphs. The variables in following equations are introduced in definition section of paper.

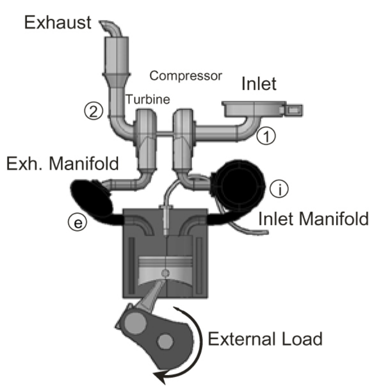

Fig. 1. Schematics of engine and numbering sequence

The test data used in this paper is obtained from the experimental data published by Hendricks and et.al, the engine properties are as follows [9]:

Table 1. Engine Properties

Cylinder Bore	$76/5\ mm$
Stroke	$86/4\ mm$
Displacement volume	$1588\ CC$
Compression ratio	$23:1$
Max. Power	$50\ kW\ @\ 4000\ rpm$
Engine rotational part inertia	$0/58\ kg.m^2$
Turbine diameter	$22\ mm$
Turbine shaft inertia	$0/9\times10^{-5}\ kg.m^2$

TURBOCHARGER MODEL

In order to compress the inlet air, mechanical energy is fed to compressor; the required mechanical torque is calculated using the following formula:

$$T_c = \frac{\dot{m}_c C_{pa} T_1}{\eta_c \omega_{tc}} \left[\left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right] \tag{1}$$

Compressor model is able to predict the required torque, flow rate and temperature of inlet air. The performance maps of compressor are usually published by turbocharger manufacturer and are used as a 2D lookup table. These maps are applied in the model using the Hendricks and et al. test data [9]. The turbine generated torque is also calculated as follows:

$$T_t = \frac{\dot{m}_t C_{p,e} T_3 \eta_t}{\omega_{tc}} \left[1 - \left(\frac{P_4}{P_3} \right)^{\frac{\gamma_e - 1}{\gamma_e}} \right] \tag{2}$$

Exhaust gas mass flow rate through the turbine is a function of turbine pressure ratio and turbine shaft speed. Also the turbine efficiency is a function of air flow Mach number and turbine shaft speed. These functions are also used in the model as 2D lookup tables. Furthermore, the dynamic model of turbocharger shaft is used to interconnect the turbine sub model and compressor and is formulated using following equation:

$$T_t - T_c = I_{tc} \dot{\omega}_{tc} \tag{3}$$

Where I_{tc} is the rotational inertia of interconnecting shaft.

INLET AND EXHAUST MANIFOLDS MODELS

In turbocharged engines, the manifolds are intermediate volumes placed between compressor and inlet ports or turbine and exhaust ports. The air flow of turbine and compressor has been calculated before. The engine air flow rate is calculated using the volumetric efficiency as follows:

$$\dot{m}_{en}(t) = \rho_2(t)\eta_v\left(p_2,\omega_{en}\right)\frac{V_d}{N}\frac{\omega_e(t)}{2\pi} \quad (4)$$

The volumetric efficiency is a function of inlet manifold pressure and engine RPM.

$$\eta_v = \eta_{v,p}(P_2).\eta_{v,\omega}(\omega) \quad (5)$$

The results of a test are considered to investigate the effects of engine speed on volumetric efficiency [9]. The inlet manifold pressure plays an important role on volumetric efficiency; there are many models which predict the volumetric efficiency as a function of inlet pressure. Inlet manifold pressure is calculated using the differential equation 6 which includes the compressor air flow rate and engine suction flow rate [10]:

$$\dot{P}_2 + \frac{\eta_v V_d N}{120 V_{im}} P_2 = \dot{m}_c \frac{RT_2}{V_{im}} \quad (6)$$

Also temperature and pressure in formula 6 are calculated using appropriate thermodynamic relations.

TORQUE GENERATION, INTERNAL FRICTION AND CRANKSHAFT DYNAMIC MODELS

In order to model the engine accurately, the thermal efficiency of engine should be available. Engine torque generation depends on some of the engine operational parameters. It is shown that the diesel engine thermal efficiency is mainly a function of three major parameters:

$$\eta_{ind} = f\left(N,\lambda,\zeta\right) \quad , \quad \frac{1}{\lambda} = \phi = \frac{\left(F/A\right)_{actual}}{f_s} \quad (7)$$

The way which thermal efficiency is related to engine speed and equivalence ratio is derived using the experimental data. This function is applied into the model as a 2D lookup table. The effect of injection timing ζ on torque generation is modeled as a second order function [11]. The result torque is calculated using energy equations. The generated torque, external load and internal friction torque together accelerate or decelerate the crankshaft. The following dynamic equation is used to calculate the engine speed.

$$T_{ind} - T_f - T_l = I_{cr}.\dot{\omega}_{cr} \quad (8)$$

In the above formula T_f represents the internal friction torque of engine moving parts and also pumping losses of engine. The pumping loss is calculated considering the difference between exhaust gas pressure and inlet pressure while the friction part is calculated as a function of both temperature and speed of engine [12].

MODEL VALIDATION

After calibrating of the model using steady state responses, it was tested by a step unloading of 10 N.m at 10th sec and subsequent loading of 10 N.m in 30th sec.

As figure 2 illustrates, the engine speed increases due to unloading in 10th sec and decreases to initial speed again. The trend of variations shows good similarity between experimental results and model ones.

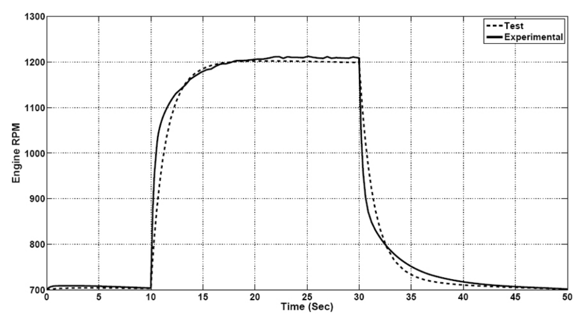

Fig. 2. Comparison of speed variation in model and experiment due to a step unloading and loading of 10 N.m

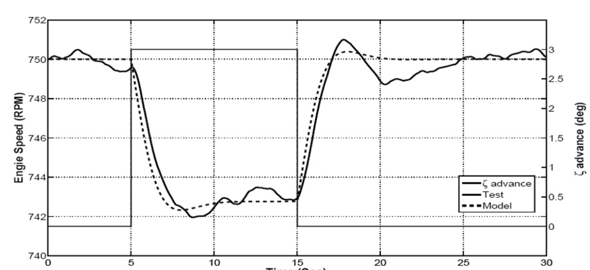

Fig. 3. Comparison of speed variation in model and experiment due to a step change in injection angle

An important manipulated variable is Injection timing ζ which will be discussed later. It will cause engine speed to vary. A test is done on the engine to see how the engine would response to ζ variation. In 5th second the injection angle is retarded 3 crank angles from MBT and is returned to MBT condition in

15th second. The low domain variations in speed are results of cycle by cycle variations in torque generation which are now more apparent. It also shows that the injection timing maybe used to compensate for low speed disturbances. However using this manipulated variable will decrease the fuel economy of engine while running in idle mode.

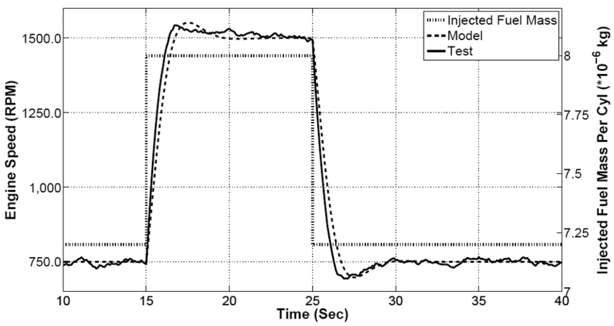

Fig. 4. Comparison of speed variation in model and experiment due to a step change in injected fuel mass

The other parameter which severely affects the engine speed is mass of fuel injected per cycle for every cylinder. The comparison between the model and experimental data show that model is able to predict the engine behavior accurately enough to be used for designing the controller.

DIESEL ENGINE IDENTIFICATION

SUBSPACE SYSTEM IDENTIFICATION

System identification is a method to derive a mathematical model for dynamic systems based on I/O data. Different methods of dynamic system identification are classified to linear and nonlinear methods. Recently fuzzy and neural networks have also been employed to identify dynamic systems [13, 14]. In this paper a linear approach is utilized to identify a state space model based on subspace method. Subspace identification is an efficient computational method to derive a state space model out of I/O data [15]. Below a full description of subspace identification method is considered regarding to Keyvaani and et al. paper on subspace identification approach [16]:

Subspace identification is a method for identification of multivariable LTI systems. This method provides a good alternative to classical nonlinear optimization based prediction error methods. Subspace identification methods for LTI systems can basically be classified into two different groups: The first group consists of the methods that aim at recovering the column space of extended observability matrix and use the shift-invariant structure of this matrix to estimate the matrices A and C; this group consists of so called MOESP methods. The second method aim at approximating the state sequence of the system and use this approximate state in a second step to estimate the system matrices; the method that constitute this group are the N4SID method, the CVA methods and the orthogonal decomposition method. The acronym N4SID stands for "Numerical algorithms for Subspace State Space System IDentification". The algorithm determines the order of the system directly from the singular value decomposition of the oblique projection. The interested reader are referred to [14, 15] to gain detail information about this identification method.

SYSTEM IDENTIFICATION USING MATLAB

In this paper Matlab® Identification Toolbox is used to identify the linear model based of N4SID method. As discussed earlier in modeling section, turbocharged diesel engine has four main dynamics processes related to inlet manifold, exhaust manifold, turbocharger inertia and crackshaft inertia. The corresponding states are P_i, P_o, N and N_{tc} respectively, where $P_i = P_2$ and $P_e = P_3$. The process to identify the engine state space model is done in two main steps:

1. Identification of an intermediate state space model with 3 inputs of $u = \begin{bmatrix} T_l & m_f & \xi \end{bmatrix}^T$ and 4 outputs of $y = \begin{bmatrix} N & N_{tc} & P_i & P_o \end{bmatrix}^T$ based on I/O data. The order of the system is obtained by the singular value decomposition method.

2. Performing a linear transformation to transform the intermediate model to desired model with states of $x = \begin{bmatrix} N & N_{tc} & P_i & P_o \end{bmatrix}^T$

I/O data used in identification is generated from the MVM developed before. The generated data are selected so that they are able to take influences of all the inputs into account. Random white noise with appropriate intensity is used to generate the demanded data. In identification procedure, 8 distinct generated data sets are used simultaneously to model engine behavior accurately, one of which is illustrated below:

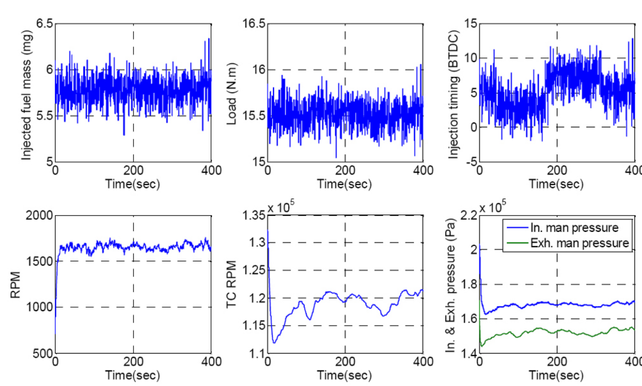

Fig. 5. A set of input-output data

Using the N4SID scheme, a discrete state space equation results in following form with sampling time of $T = 0.1\ Sec$:

$$\begin{cases} x'(t+T)_{8\times 1} = A'_{4\times 4}\, x'(t) + B'_{4\times 3}\, u(t)_{3\times 1} \\ y(t)_{4\times 1} = C'_{4\times 8}\, x'(t) + D'_{4\times 3}\, u(t)\ , D' = 0 \end{cases} \tag{9}$$

In which $u = \begin{bmatrix} T_l & m_f & \xi \end{bmatrix}^T$ and $y = \begin{bmatrix} N & N_{tc} & P_i & P_o \end{bmatrix}^T$. This state space equation has non-meaningful states due to subspace identification. A linear transformation is applied to transform the equation 16 to a system with meaningful states as follow:

$$\begin{cases} y(t+T)_{4\times 1} = A_{d\,4\times 4}\, y(t) + B_{d\,4\times 3}\, u(t)_{3\times 1} \\ N(t)_{4\times 1} = C_{4\times 4}\, y(t) \end{cases} \tag{10}$$

Since $D' = 0$ in equation 16, state equation and output equation can be merged. Consequently A and B matrices are computed: $A_d = C'A'C'^{-1}$, $B_d = C'B'$ furthermore output is simply the first state of system and also $D = 0$. In order to derive a continues state space model, the following matrix functions are used:

$$\begin{aligned} A_d &= e^{AT} \\ B_d &= \left(e^{AT} - I\right)A^{-1}B \Rightarrow B = A\left(A_d - I\right)^{-1} B_d \end{aligned} \tag{11}$$

In which A_d and B_d are state and input matrices of discrete state space system respectively and T is sampling time of discrete system. In order to describe the engine state space in more meaningful manner, the input signals are expanded to disturbance signal and real input signals; load is really a disturbance while injected mass and injection timing are two real inputs to engine. So equation 9 will alter to following form:

$$\begin{cases} y(t+T) = Ay(t) + Bu(t) = Ay(t) + B_u u_{new}(t) + Ev(t) \\ N(t) = Cy(t) \end{cases} \tag{12}$$

Where $v(t)$ is disturbance (load) and $u_{new}(t)$ is input vector which contains injection mass and injection timing. The matrices in equation 12 will calculated to be:

$$A = \begin{bmatrix} -0.3679 & 0.0042 & -0.0043 & 0.0008 \\ -2.9647 & 0.0524 & -0.1661 & 0.1534 \\ -0.9883 & 0.0456 & -0.2064 & 0.2115 \\ 7.0695 & 0.0073 & 0.0369 & -0.1088 \end{bmatrix}$$

$$B_u = \begin{bmatrix} 272.6 & -9.3 \\ -241.8 & -128 \\ -2757.1 & -94.4 \\ -830.6 & 157.7 \end{bmatrix}, E = \begin{bmatrix} -52.8 \\ 330.5 \\ 957.9 \\ 109.5 \end{bmatrix}$$

$$C = \begin{bmatrix} 1 & 0 & 0 & 0 \end{bmatrix}$$

Where A is state dynamic matrix, B_u is input matrix, E is disturbance matrix and C is output matrix.

STATE SPACE MODEL VALIDATION

The state space model is verified by appropriate stimulus, a test procedures is done to verify the state space model. The stimuli and result of test are illustrated in figures 6 and 7. The results show an adequate agreement between state space and nonlinear model. The test results show that engine speed prediction have a RMS error of 3.46%, 6.57%, 3.74% and 2.65% for engine RPM, turbocharger RPM, inlet and exhaust pressure respectively for step sequence.

CONTROLLER DESIGN

LQR FINITE TIME CONTROLLER

The optimal control is best suited for cases which convert energy to other forms, the major effect of optimal control is its effect on systems in transient behavior. Usually optimal control compromises between output performance and effort energy. The weighting factors are used to balance between input values and output performance. In optimization of system can be considered for a finite time or and infinite time span. The finite time LQR method is an optimal controlling approach based on minimization of a performance index in a finite time as follows, where $u(t)$ is the controlling effort and r is the reference value of states.

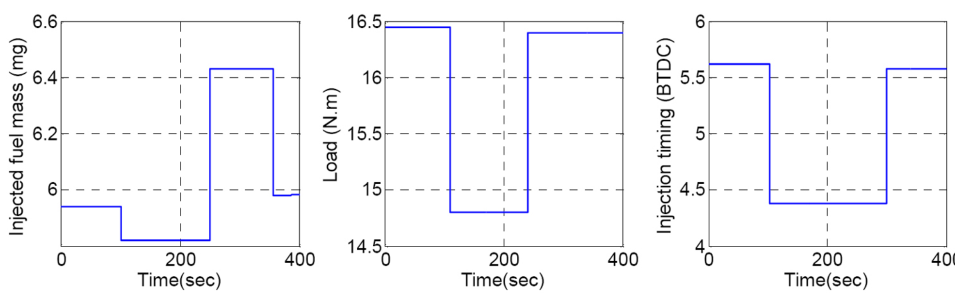

Fig. 6. Step sequences stimuli for verification test

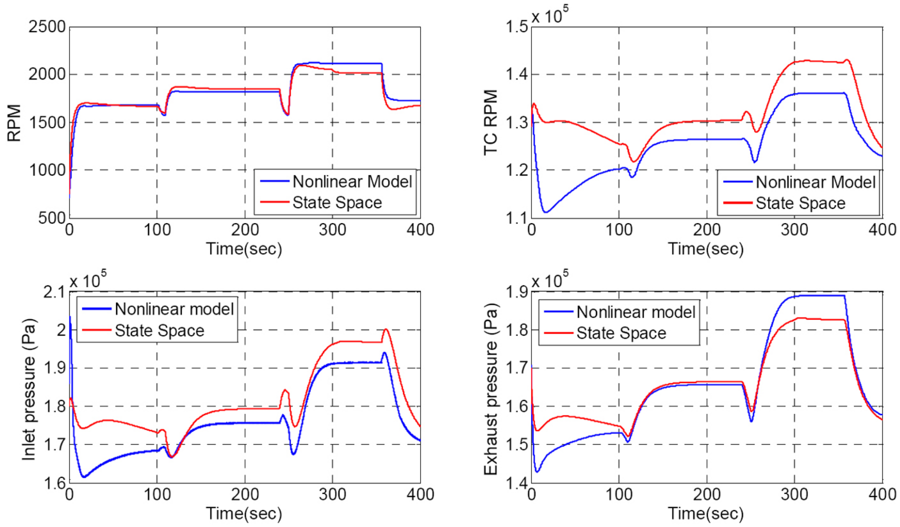

Fig. 7. Result of verification for step sequences input

$$J = \frac{1}{2}\left[x(t_f) - r\right]^T H\left[x(t_f) - r\right] + \frac{1}{2}\int_0^{t_f} \left\{\left[x(t) - r\right]^T Q\left[x(t) - r\right] + u^T(t)Ru(t)\right\} \quad (13)$$

The main problem is to find a control effort $u(t)$ in such a way to minimize the above performance index. Based on Hamiltonian method the appropriate controlling effort is calculated as follow:

$$u(t) = -R^{-1}B_u{}^T K(t)x(t) - R^{-1}B_u{}^T s(t) \quad (14)$$

In which $K(t)$ and $s(t)$ are simply the solution of following differential equations:

$$\dot{K}(t) = -K(t)A - A^T K(t) - Q + K(t)B_u R^{-1}B_u{}^T K(t)$$
$$K(t_f) = H \quad (15)$$

$$\dot{s}(t) = -\left[A^T - K(t)B_u R^{-1}B_u{}^T\right]s(t) + Qr$$
$$s(t_f) = -Hr \quad (16)$$

Matlab® is used to solve the above differential equations, depend on selection of Q, H and R different controlling scenarios are resulted, some of which are illustrated and discussed follow. Different loading sequences are tested to check the controller performance. On the other hand different time spans are simulated to check the effect of finite time on engine fuel consumption.

Since only the engine speed is to be regulated in 750 RPM, Q and H matrices takes into account only the first state. The weighting and set point matrices are defined as follows:

$$Q = \begin{bmatrix} q_{rpm} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}, H = \begin{bmatrix} h_{rpm} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}, R = \begin{bmatrix} r_f & 0 \\ 0 & r_{IT} \end{bmatrix}, r = \begin{bmatrix} 750 \\ 0 \\ 0 \\ 0 \end{bmatrix} \quad (17)$$

The system response for sinusoidal external loads is considered follow, engine speed, TC speed, inlet and exhaust pressures and corresponding injection fuel mass and injection timing values are illustrated in figures 9 and 10 for different weighting combinations. Also the external load is illustrated in figure 8.

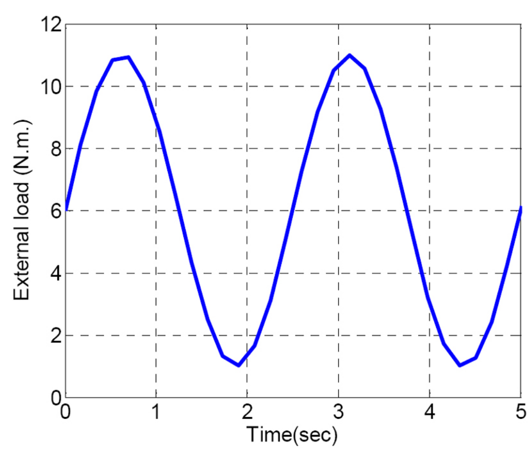

Fig. 8. The external sinusoidal load

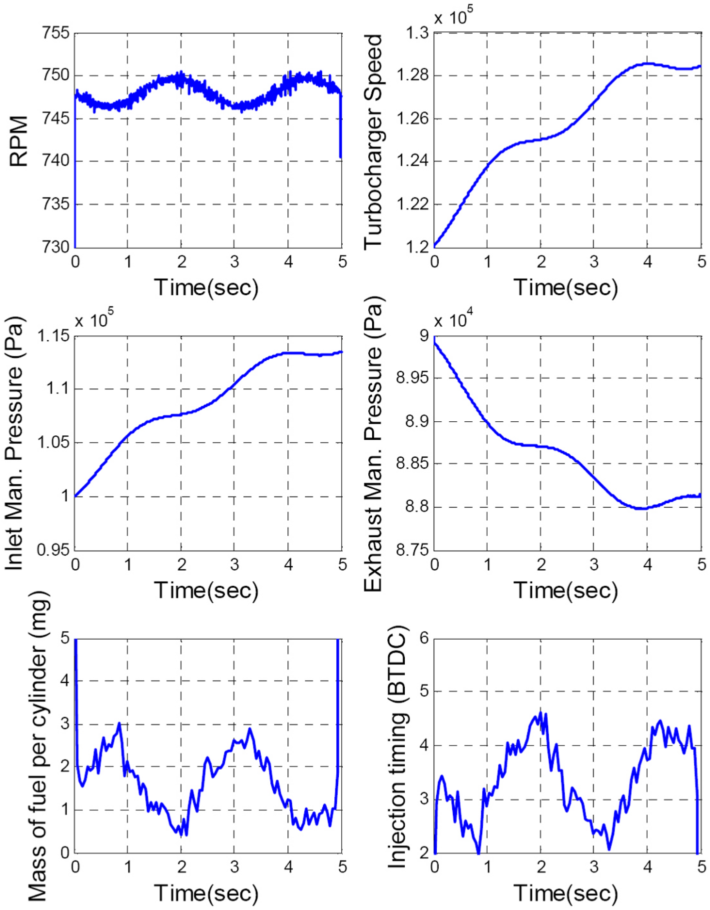

Fig. 9. The controlled states and corresponding optimal inputs for q_{rpm} = 1, $_{hrpm}$ = 6, r_f = 3, r_{IT} = 0.1

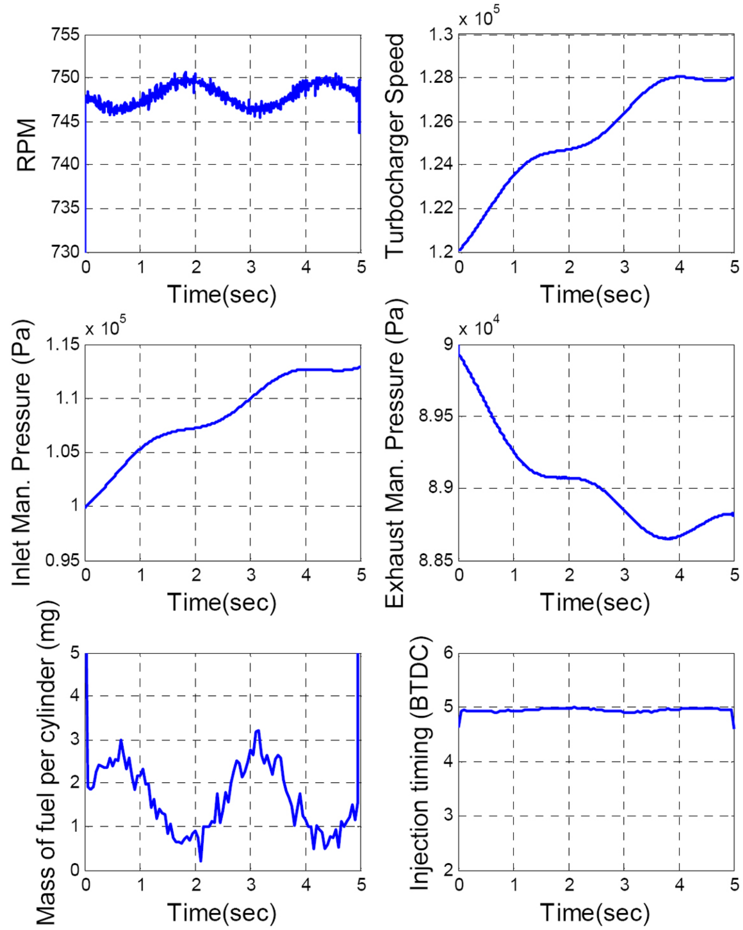

Fig. 10. The controlled states and corresponding optimal inputs for q_{rpm} = 1, $_{hrpm}$ = 1, r_f = 3, r_{IT} = 3

As illustrated in figures 9 and 10, increasing r_{IT} will lower the deviation of injection timing, on the other hand varying controller parameter does not affect inlet manifold pressure and TC speed, but it affects exhaust manifold pressure to some extend.

SENSITIVITY ANALYSIS

After designing the controller, the controller parameter should be tuned regard design demands. The parameters to tune are weighting parameters in equation 17, e.g. q_{rpm}, h_{rpm}, r_f, r_{IT} which stand for speed transient, speed target, fuel injection in transient and injection timing in transient respectively. On the other hand the other important parameter which should be taken into account is the value of final time t_f in equation 13 which stands for the value of control time duration. The q_{rpm}, r_f, r_{IT} parameters are tuned based on importance of each parameter. Increasing r_f will decrease fuel consumption while increasing q_{rpm} decreases the speed fluctuations in expense of fuel economy. Increasing r_{IT} lowers the deviation of injection timing relative to optimal injection timing which is turn would decrease fuel consumption.

At last the following value are assigned to weighting factors, q_{rpm} = 1, r_f = 3, r_{IT} = 2. The time span for LQR finite time controller and h_{rpm} are analyzed to find their optimum value based on engine fuel consumption. In order to make an optimum tuning the variation of fuel consumption is plotted versus finite time value for different h_{rpm} s. A sinusoidal input of 2Hz frequency is used to examine the controller.

As illustrated in figure 11, the effect of h_{rpm} on fuel consumption is not important for finite time values of more than 5 sec, also it has been shown that the fuel consumption converge to a fix value of 2 mg/cyl for time values of more than 5 sec. It is also obvious that fuel consumption increase with increasing h_{rpm} in low time spans. It's mostly due to effects of h_{rpm} on speed regulation. Decreasing h_{rpm} will increase the settling time. At last h_{rpm} = 2 and t_f=5 sec is selected due to their optimal effects of fuel consumption.

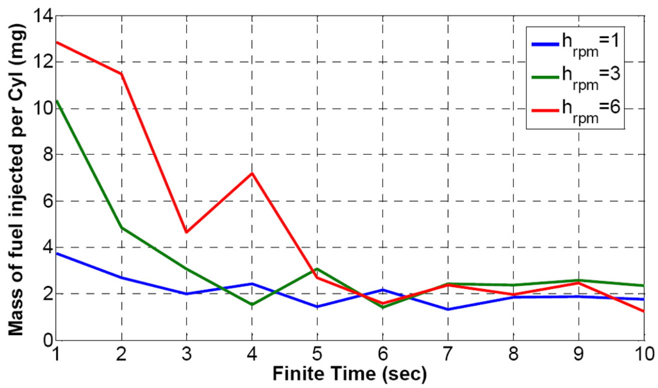

Fig. 11. Controller parameter tuning based on fuel consumption optimization

In order to verify the controller design with the optimal parameters a step load is exerted to engine as follows:

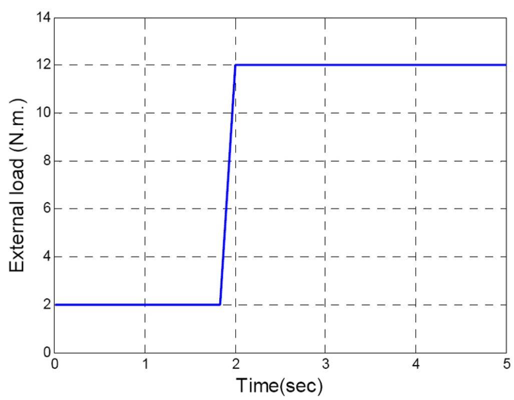

Fig. 12. The external step load

The result of controller is illustrated in figure 13.

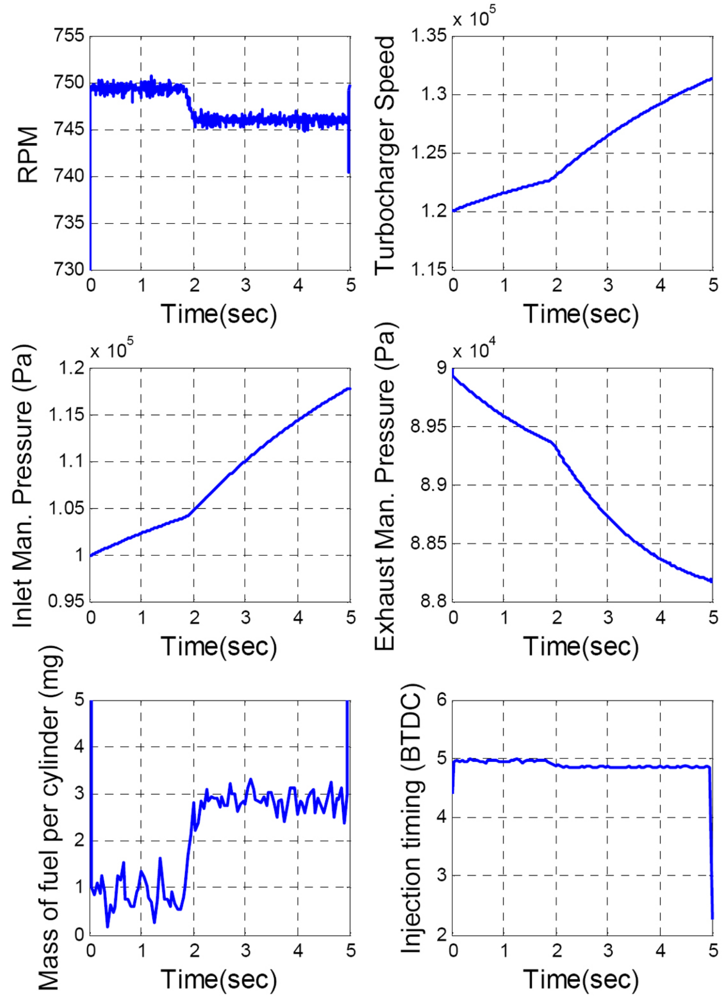

Fig. 13. The controlled states and corresponding optimal inputs for the step load

CONCLUSION

A new approach for developing state space representation of engine is used instead of normal linearization methods which needs differentiable equations. The testing procedure for identification procedure is minimized by introduction of a mean value model instead of real engine. A LQR finite controller the applied to the model to regulate the speed of engine to desired idle speed, variation of weighing values are studied, it is apparent that increasing fuel consumption weight result is decreasing fuel consumption while increasing speed variation, on the other hand the final time weighting value has not significant effect on control procedure. Some deviation from desired speed is seen in the controlled value (RPM), it's mostly because the influences of load on system dynamics as seen in Eq. 12. the variation in speed decreases by increasing the RPM weighting value. Also it is shown that t_f is an effective parameter on fuel consumption (controller target), furthermore the effects of h_{rpm} on fuel consumption is significant for low t_f (final control time).

DEFINITIONS

Notation	Description	unit
B	Cylinder bore	m
$C_{p,a}$	Heat capacity of inlet air	$kJ/kg.K$
$C_{p,e}$	Heat capacity of exhaust gas	$kJ/kg.K$
$(F/A)_a$	Actual fuel to air ratio	
$fmep_f$	Friction mean effective pressure due to wet friction	Pa
f_s	Stochiometric fuel to air ratio	
h_{rpm}	Weighting factor for speed regulation	

Notation	Description	unit
I_{cr}	Crankshaft and engine rotating part inertia	$kg.m^2$
I_{tc}	Turbocharger shaft and blades rotational inertia	$kg.m^2$
$\dot{m}_c$	Mass rate of air through compressor	*kg/sec*
$\dot{m}_{en}$	The passing air mass flow rate through engine	*kg/sec*
$\dot{m}_{ex}$	Exhaust gas mass flow rate	*kg/sec*
m_f	Mass of injected fuel per cylinder	*mg*
$\dot{m}_t$	Mass rate of gas through turbine	*kg/sec*
N	Engine speed	*RPM*
N_{tc}	Turbocharger speed	*RPM*
P_1	Pressure of air before compressor	*Pa*
P_4	Pressure of exhaust gas after turbine	*Pa*
P_e	Exhaust manifold pressure	*Pa*
P_i	Inlet manifold pressure	*Pa*
q_{in}	Combustion specific heat value	*kJ/kg*
q_{rpm}	Weighting factor for speed fluctuations	
r_c	Compression ratio	
r_{IT}	Weighting factor for Injection Timing	
r_f	Weighting factor for mass of injected fuel	
S	Piston stroke	*m*
t_f	LQR Controlling time duration (finite time LQR)	
T_1	Temperature of air before compressor	*K*
T_4	Temperature of exhaust gas after turbine	*K*
T_c	Compressor demanded torque	*N.m*
T_f	Friction torque	*N.m*
T_{ind}	Engine indicated torque	*N.m*
T_l	Load torque	*N.m*
T_t	Turbine generated torque	*N.m*
V_c	Combustion chamber volume	m^3
V_d	Engine displacement volume	
V_{im}	Inlet manifold volume	m^3
Γ_d	Observabilty matrix	
$\Pi_{e,max}$	Maximum ratio of exhaust pressure to inlet	
ϕ	Fuel to air equivalence ratio	
γ_a	Air specific heat ratio	
γ_e	Exhaust gas specific heat ratio	
η_c	Compressor efficiency	
η_{ind}	Engine indicated efficiency	
η_t	Turbine efficiency	
η_v	Volumetric efficiency of engine	
$\eta_{v,\omega}$	Volumetric efficiency (speed related part)	
λ	Air to fuel equivalence ratio	
ω_e	Engine speed	*Rad/sec*
ω_{tc}	Turbocharger speed	*Rad/sec*
ξ,	Injection timing	*BTDC°*
ξ_0	Injection timing correspond to MBT	*BTDC°*

ABBREVIATIONS

I/O
Input-Output

CVA
Canonical Variable Analysis

EMS
Engine Management System

FC
Fuel Consumption

ICE
Internal Combustion Engine

ISC
Idle Speed Controller

LQG
Linear Quadratic Gaussians

LQR
Linear Quadratic Regulators

LTI
Linear Time Invariant

MBT
Maximum Brake Torque

MOESP
Multivariable Output Error State Space

MVM
Mean Value Model

N4SID
Numerical algorithms for Subspace State Space System IDentification

RMS
Root Mean Square

SVFC
State Vector Feedback Control

TC
TurboCharger

VGT
Variable Geometry Turbine

REFERENCES

1. Jurgen, R., "Automobile Electronics Handbook. New York", McGraw-Hill, 1995.

2. Ye, Zhengmao, "Modeling, Identification, Design and Implementation of Nonlinear Automotive Idle Speed Control Systems-An Overview", IEEE Transactions on Systems, Man and Cybernetics, November 2007

3. Vachtsevanos, G., Farinwata, S., and Kang, H., "A Systematic Design Method for Fuzzy Logic Control with Application to Automotive Idle Speed Control," in Proc. 31st Conf. Decision Control, Tucson, AZ, Dec. 1992.

4. Kolmanovsky, E.Gilbert, and Cook, J., "Reference Governors for Supplemental Torque Source Control in Turbocharged Diesel Engines" in Proc. American Control Conference, Albuquerque, NM, vol. 1, pp. 652-656, 1997

5. Memering, D. and Meckl, P., "Comparison of Adaptive Control Technique Applied to Diesel Engine Idle Speed Regulation," ASME Journal of Dynamic Systems Measurement and Control, vol. 124, pp. 682-688, Dec. 2000.

6. Villegas, J., Genc, U., Karagiorgis, S., Tafner, R. et al., "Auto-Tuning of PID Controllers for Calibration of Idle Speed Controller Maps in Diesel Engines," SAE Technical Paper 2010-01-0330, 2010, doi:10.4271/2010-01-0330.

7. Mills, J.S., "Multivariable Control of Engine Idle Speed," SAE Technical Paper 970611, 1997, doi: 10.4271/970611.

8. Han, M., Loh, R.N.K., Wang, L., Lee, A. et al., "Optimal Idle Speed Control of an Automotive Engine," SAE Technical Paper 981059, 1998, doi: 10.4271/981059.

9. Jensen, J.-P., Kristensen, A.F., Sorenson, S.C., Houbak, N. et al., "Mean Value Modeling of a Small Turbocharged Diesel Engine," SAE Technical Paper 910070, 1991, doi: 10.4271/910070.

10. J Kao, M.H., Moskwa, J. J., "Turbocharged Diesel Engine Modeling for Nonlinear Engine Control and State Estimation" ASME Journal of Dynamic Systems Measurement and Control, Vol 117, March 1995.

11. Gerhardt, J., Hönninger, H., and Bischof, H., "A New Approach to Functional and Software Structure for Engine Management Systems - BOSCH ME7," SAE Technical Paper 980801, 1998, doi: 10.4271/980801.

12. Guzzella, L., Onder, C.H, "Introduction to Modeling and Control of ICE systems" Springer, 2004

13. Nelles, O., "Nonlinear System Identification: From Classical Approaches to Neural Networks and Fuzzy Models", Springer Berlin Heidelberg, 2000

14. Ljung, L., "System Identification: Theory for Users", Prentice Hall PTR, 1999

15. Overschee, P., Moor, B., "Subspace Identification for Linear Systems: Theory-Implementation-Applications", Kluwer Academic Publishers, 1996

16. Keyvaani, H., Mohammadi, A and Pariz, N., "Identification of Modal Series Model of Nonlinear Systems Based on Subspace Algorithms", Journal of Applied Sciences, Volume 10, papers 909-914, 2010

The Engineering Meetings Board has approved this paper for publication. It has successfully completed SAE's peer review process under the supervision of the session organizer. This process requires a minimum of three (3) reviews by industry experts.

ISSN 0148-7191

Positions and opinions advanced in this paper are those of the author(s) and not necessarily those of SAE. The author is solely responsible for the content of the paper.

SAE Customer Service:
Tel: 877-606-7323 (inside USA and Canada)
Tel: 724-776-4970 (outside USA)
Fax: 724-776-0790
Email: CustomerService@sae.org
SAE Web Address: http://www.sae.org
Printed in USA

SAE International®