Development of a Hierarchical Observer for Burned Gas Fraction in Inlet Manifold of a Turbocharged Diesel Engine

Kamyar Nikzadfar and Amir H. Shamekhi

***Abstract*—Altogether with extent of automotive role in today's life, governments are approving stringent laws to lower the permitted level of automotive emissions. An important specie of automotive engines emissions is NO_x. Different methods are developed to decrease the level of NO_x generation; one of which is exhaust gas recirculation (EGR). Since the performance of EGR is strictly depended on burned gas fraction (BGF) in inlet manifold, the precise control of BGF is of importance. Unfortunately, due to inlet manifold temperature, no economical sensor is available to measure the BGF; therefore, estimators are employed instead. In this paper, a stable observer is designed for estimation of air fraction in inlet manifold. The governing equations are in the form of linear parameter varying (LPV). Since the LPV parameters are not directly measured, a hierarchical estimator structure is developed. Lyapunov theory is employed to design the higher level estimator, while high gain estimators and open-loop estimators are developed for estimation of the lower level parameters. Experimental test results show that the higher level estimator is able to estimate the BGF with high accuracy in both transient and steady states. Furthermore, it is shown that BGF estimator is more sensitive to aspirated gas flow estimation rather than exhaust temperature and pressure.**

***Index Terms*—Burned gas fraction estimation, intake oxygen concentration, diesel engines, linear parameter varying, hierarchical estimator, dirty derivative observer.**

Nomenclature

Abbreviations

BGF	burned gas fraction
LPV	linear parameter varying
EGR	exhaust gas recirculation
BSFC	brake specific fuel consumption
UEGO	universal exhaust gas oxygen
LMI	linear matrix inequality
MRIS	model reference identification scheme
HiL	hardware in the loop
VGT	variable geometry turbine
IOC	intake oxygen concentration

Manuscript received November 30, 2017; revised April 29, 2018, July 29, 2018, and October 4, 2018; accepted October 16, 2018. Date of publication October 22, 2018; date of current version December 14, 2018. The review of this paper was coordinated by Dr. M. S. Ahmed. *(Corresponding author: Kamyar Nikzadfar.)*

K. Nikzadfar is with the Department of Mechanical Engineering, Babol Noshirvani University of Technology, Babol 47148-71167, Iran (e-mail: nikzadfar@nit.ac.ir).

A. H. Shamekhi is with the Department of Mechanical Engineering, K. N. Toosi University of Technology, Tehran 15875-4416, Iran (e-mail: shamekhi@kntu.ac.ir).

Color versions of one or more of the figures in this paper are available online at http://ieeexplore.ieee.org.

Digital Object Identifier 10.1109/TVT.2018.2877580

I. Introduction

With stringent legislation by governments to decrease emission, extensive efforts are made by automotive manufacturers to lowering the level of internal combustion engines emission as a main source of pollution. One of the most important species of diesel engine emissions is NO_x. NO_x is generated due to breakage of air Nitrogen molecules in high temperature combustion. NO_x as a main part of engine emissions is critical to change biological processes and plays an important role in various physiological and pathological processes. It also contributes to formation of tropospheric ozone. Therefore, different technologies are developed to reduce NO_x. These technologies might be categorized into two classes of in-cylinder treatment and after treatment systems. Exhaust gas recirculation (EGR) is a well-known technology which delivers the combustion products from exhaust manifold into inlet manifold through a controlled valve in order to reduce the combustion temperature. The recirculated exhaust gases decrease the level of NO_x generation by decreasing the share of combustible gases in inlet gas which will in turn decrease the temperature of combustion. The precise control of BGF in inlet manifold will significantly reduce the level of NO_x generation with negligible effects on soot generation. Although, further increase of BGF in inlet air leads to a drastic decrease of NO_x formation, too much burnt gas will increase both soot and brake specific fuel consumption (bsfc) [1]. These all make precise control of EGR an important task in engine management systems. Usually closed loop control systems are employed to control burnt gas mass fraction is inlet manifold by regulating EGR valve. In production-type diesel engines control systems, the mass air flow is usually employed as feedback variable to control EGR, however it is not a good choice for control of EGR due to weak correlation of NO_x formation with air mass flow [2]. Although the best feedback parameter is oxygen concentration in inlet manifold, it is not possible to measure the level of inlet manifold burnt mass fraction by conventional sensors due to insufficient temperature in inlet manifold; therefore, observers are employed to estimate burnt gas mass fraction. By exact measuring of recirculated exhaust gas mass flow rate, besides inducted air flow rate; open loop estimators can be employed to estimate burnt mass fraction in manifold [3]. However, in the case of diesel engines -where not all the exhaust gas is burnt gases-, open loop observers which solely operate based on EGR mass flow rate estimation fail to estimate intake oxygen concentration (IOC) -or intake burnt mass fraction- with desired accuracy in transition regime [4]. That is why the task of burnt mass fraction estimation is of high importance specially in transient operations. Inlet and exhaust manifolds, fuel injection system and engine induction system forms a dynamic system with burned gas (fresh air) fraction as states. Today, with advent of universal exhaust gas oxygen (UEGO) sensors, it is possible to measure fresh air mass fraction in exhaust manifold as the measurable output of the mentioned dynamic system [5], [6]. Unfortunately, due to nonlinearities involved in processes, the task of estimation is complicated and leads to complex nonlinear observers.

Different researchers have addressed the problem of BGF estimation in diesel engines. D'Ambrosio *et al.* developed a thermo-fluidic zero dimensional model of EGR gas flow in gas passages of diesel engine [7]. They employed pressure and temperature of downstream and upstream of EGR valve to estimate the flow through EGR valve lift based on orifice equation and discharge coefficient. The result models are applied to estimate EGR rate in both conventional and non-conventional combustion conditions. In order to promote the performance of estimation in transient modes, some researchers tried to compensate for dynamic prediction of EGR gas flow rate in transient. Kyunghan *et al.* employed correction algorithms for estimation of IOC in transient. They defined dynamic correction state for orifice valve of EGR, also they used energy conversation equations in intake manifold to consider the dynamic effects of inlet manifold pressure on estimation of IOC in transients. They determined a linear parameter varying (LPV) based on these dynamic models, after which they employed extended Kalman filter to derive the value of correction factors. They demonstrated that their estimation scheme can estimate the value of IOC in transient regimes. Diop *et al.* developed an estimation algorithm for the oxygen concentration in the intake manifold of a turbocharged diesel engine [8]. The developed algorithm employed boost pressure, fuel injection rate, engine speed and EGR valve opening for estimation. They used a first order dynamic model for estimation. The resulted estimator was slow due to engine constraints for guarantying the stability of observer. Chen and Wang developed an observer to estimate the air fractions in the integrated diesel engine and after treatment system [9]. They claimed that ignoring the after treatment systems and filters in designing air fraction estimators dictates significant errors on estimation. They developed a Luenberger-like observer and analysed it using a Lyapunov method as well as the physical meaning of system parameters. Zeng and Wang developed an estimator considering time-varying transport delays [10]. They calculated the delay time based on the continuity of fluid velocity and show that developed estimator can properly estimate air fraction in dynamic situations. Zhao and Wang developed a Luenberger-like observer for the oxygen fraction in inlet manifold based on dynamic model of air-path loop taking in to the account the existence of oxygen content in the biodiesel fuel [11]. The convergence of observer is proved using physical consideration on engine. Wang developed an estimator for estimation of air fraction in a dual loop EGR system where two paths exist for exhaust gas recirculation [5]. He used the UEGO sensor data besides a 4th order LPV model of a turbocharged engine provided with a pressure throttle for low load EGR demands for estimation purposes. The LPV model is composed of varying parameters which describe the systems, where the parameters are functions of air fraction, gas flowrates and pressure and temperature of different inlet/outlet locations. Finally, he used a Lyapunov-based approach to stabilize the estimator in whole engine operating points. The measurement/estimation of parameters used in LPV model was a restriction in state estimation. The proposed engine in Wang research was provided with pressure and temperature sensors for exhaust manifold, however the exhaust sensors are not available in every diesel engine. On the other hand, Wang employed an open-loop estimator for recirculated gas mass flowrate using a simple orifice and measuring upstream and downstream pressure and upstream temperature of recirculated mass flow rate. Since the estimator does not compensate for dynamic process, the error of estimation will arise specially in transient modes. Feilong and Pfeiffer developed a set of estimators to estimate flow rate, pressure and temperature of inlet and exhaust manifold for low pressure cooled EGR [12]. In the other research, Wang *et al.* developed a model based observer for oxygen concentration in a VGT turbocharged diesel engine [13]. They proposed a less conservative observer development method for Lipschitz nonlinear system using Ricatti equations. They also employed linear matrix inequality (LMI) to obtain the observer gains. Castilo *et al.* tried to developed an estimator for simultaneously estimate the low pressure recirculated gas mass flowrate and BGF of a dual loop EGR diesel engine [6]. They used a sliding mode observer to estimate the low pressure recirculated mass flowrate based on inlet manifold pressure and measured inlet flowrate sensor. Also they used LPV model of engine inlet/exhaust system to estimate the BGF in desired location based on robust Kalman filter and LMI approach. Lee *et al.* employed model reference identification scheme (MRIS) to estimate the exhaust gas recirculation [14]. They developed a LPV model of inlet manifold gas dynamics as a reference model for observer design and derived a stabilized update rule, based on model reference identification to estimate exhaust mass flowrate. The problems encountered in design of BGF estimation in diesel engines can be categorized into three class of 1) BGF estimation 2) unknown mass flow rate estimation and 3) hard to measure temperature and pressure estimation. However, a research which takes into account all the required estimators and study the performance of a unified estimator has not been found in the literature.

In this paper, a closed loop estimator for estimation of BGF in inlet manifold of high speed turbocharged engine with pressure throttle is developed based on standard sensors. Where the parameters are hard to measure or expensive sensors should be provided to measure the parameters, appropriate easy implementable estimation schemes are developed. Finally, a hierarchical estimation scheme is developed containing both lower level estimators for estimation of pressure, temperature and mass

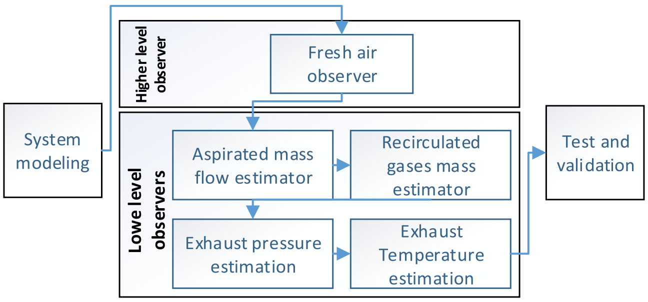

Fig. 1. The flowchart of research process.

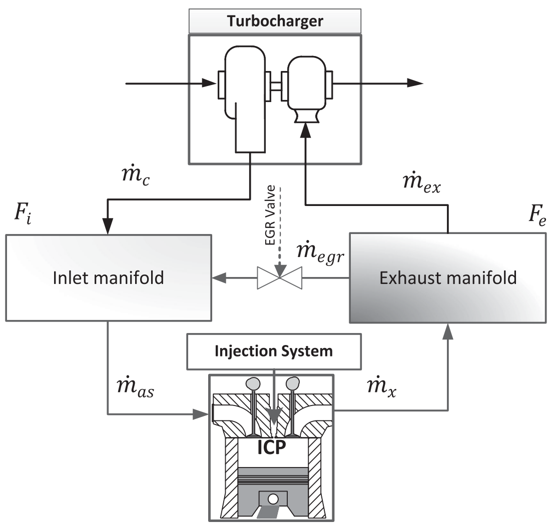

Fig. 2. Fresh and burned gas dynamic system.

flowrates and an upper level estimator for estimation of BGF in inlet manifold. Dirty derivatives as an unknown input observer is employed to estimate recirculated mass flowrate based on inlet manifold pressure [15]. Also the exhaust pressure is estimated based on turbine performance [16], [17]. The performance of integrated estimator is verified in different engine operation and the effects of lower level estimators on burned gas mass fraction estimation is studied.

The process of estimator design and different phases of research is depicted in Fig. 1.

II. Modelling of Burned Gas Fraction

In order to estimate the BGF in inlet manifold, a dynamic system with inlet manifold BGF (or fresh air fraction) as a state and measurable outputs should be identified. By obtaining such a system, it would be possible to estimate the unknown state (BGF) by measuring the system output and using the obtained model. By considering inlet and exhaust manifolds as the related dynamic system and employing a broad band lambda sensor in exhaust system, the desired system is obtained as depicted in Fig. 2. In order to design and implement the estimator, a 1.5 litre common rail turbocharged diesel engine is proposed. The specification of under studied engine is as shown in Table I.

The mentioned dynamic systems is composed of inlet and exhaust manifolds, exhaust gas recirculation valve, turbocharger,

TABLE I
The Engine Specifications

Displaced volume	1497 CC
Number of Cylinders	4
Stroke	82.5 mm
Bore	76 mm
Connecting Rod	134.25 mm
Compression ratio	16.5:1
Number of Valves per	4
Injection System	Common Rail
Air Boost System	VGT+Intercooler

injection system and combustion system. The main dynamics of system belongs to manifolds filling process. Since the engine management system is provided with an inlet mass flow rate sensor, $\dot{m}_c$ is known. Also, the exhaust manifold out-flow is measured by a pressure difference-based flow sensor attached on catalytic convertor. As inlet and exhaust flow rates are measured, there is no need to contain turbocharger dynamics into observer model. Since the oxygen sensor in exhaust manifold measures the fresh air ratio, the observer is designed to observe fresh air instead of burned gas. Since only two species of fresh air and burned gas are assumed to exist in manifolds, equation (1) is valid:

$$X = 1 - F \tag{1}$$

In which X is BGF and F is air fraction. In order to estimate fresh air fraction, the manifold species governing state space model is employed. The state vector of $\mathcal{F}$ is composed of two states of a) air fraction in inlet manifold F_i and b) air fraction in exhaust manifold (F_e) as follows:

$$\boldsymbol{\mathcal{F}} = \begin{bmatrix} F_i & F_e \end{bmatrix}^T \tag{2}$$

Based on continuity equation and also species continuity equation for air and burned gas:

$$\begin{cases} \dot{F}_i = \dfrac{RT_i}{P_i V_i}\left[F_e \dot{m}_{egr} + \dot{m}_c - F_i \dot{m}_{as}\right] \\ \dot{F}_e = \dfrac{RT_e}{P_e V_e}\left[F_x \dot{m}_x - F_e \dot{m}_{egr} - F_e \dot{m}_{ex}\right] \end{cases} \tag{3}$$

Where $\dot{m}_{as}$ is aspirated mass flow rate and $\dot{m}_{egr}$ is exhaust recirculated mass flow rate. Also $\dot{m}_x$ is the sum of entering mass flow rate into exhaust manifold which includes inducted mass and injected fuel mass in cylinder and is calculated by equation (4) as follows:

$$\dot{m}_x = \dot{m}_{as} + \dot{m}_f \tag{4}$$

Furthermore F_x is the fraction of fresh air in cylinder out flow. F_x is calculated based on injected fuel flow as follows:

$$F_x = \frac{F_i \dot{m}_{as} - AFR_{st}.\dot{m}_f}{\dot{m}_{as} + \dot{m}_f} \tag{5}$$

Since a UEGO sensor is employed to measure air fraction in exhaust manifold, the second state F_e is considered as the only output of system. The equation (3) is reformed as follows:

$$\begin{bmatrix} \dot{F}_i \\ \dot{F}_e \end{bmatrix} = \begin{bmatrix} -\alpha \dot{m}_{as} & \alpha \dot{m}_{egr} \\ \beta \dot{m}_{as} & -\beta\left(\dot{m}_{egr} + \dot{m}_x\right) \end{bmatrix} \begin{bmatrix} F_i \\ F_e \end{bmatrix} + \begin{bmatrix} \alpha \dot{m}_c \\ -\beta \dot{m}_f.AFR_{st} \end{bmatrix} \tag{6}$$

In which:

$$\begin{cases} \alpha = \dfrac{RT_i}{P_i V_i} \\ \beta = \dfrac{RT_e}{P_e V_e} \end{cases} \tag{7}$$

The equation (6) may be simplified as follows:

$$\begin{bmatrix} \dot{F}_i \\ \dot{F}_e \end{bmatrix} = \begin{bmatrix} \rho_1 & \rho_2 \\ \rho_3 & \rho_4 \end{bmatrix} \begin{bmatrix} F_i \\ F_e \end{bmatrix} + \begin{bmatrix} \rho_5 \\ \rho_6 \end{bmatrix} \tag{8}$$

In which ρ_i variables are defined as follows:

$$\begin{aligned} \rho_1 &= -\alpha \dot{m}_{as} \\ \rho_2 &= \alpha \dot{m}_{egr} \\ \rho_3 &= \beta \dot{m}_{as} \\ \rho_4 &= -\beta\left(\dot{m}_{egr} + \dot{m}_x\right) \\ \rho_5 &= \alpha \dot{m}_c \\ \rho_6 &= -\beta \dot{m}_f.AFR_{st} \end{aligned} \tag{9}$$

Briefly we have:

$$\begin{cases} \dot{\boldsymbol{\mathcal{F}}} = A(\boldsymbol{\rho})\boldsymbol{\mathcal{F}} + w(\boldsymbol{\rho}) \\ y = F_e = C\boldsymbol{\mathcal{F}} = \begin{bmatrix} 0 & 1 \end{bmatrix} \boldsymbol{\mathcal{F}} \end{cases} \tag{10}$$

Finally the governing equations for air fraction system is turned to a linear parameter varying (LPV) system, in which $A(\rho)$ and $w(\rho)$ varies based on engine operating points. Since the mentioned matrixes are functions of ρ_i, and they are functions of engine operating parameters (basically engine internal mass flow rates), the matrixes will be bounded as follows:

$$l_i \le \rho_i \le u_i \tag{11}$$

In which l_i and u_i are lower and upper bound for ρ_i respectively. On the other hand, since ρ_i parameters are physical depended parameters, they have relatively slow rates and cannot change abruptly.

III. Fresh Air Observer Development

Different methods are developed for LPV systems [3], [4]. In this article a stable observer is designed based on Lyapunov theorems. Based on equation (10), the observer equation is obtained as follows:

$$\begin{cases} \dot{\widehat{\boldsymbol{\mathcal{F}}}} = A(\boldsymbol{\rho})\widehat{\boldsymbol{\mathcal{F}}} + w(\boldsymbol{\rho}) \\ \widehat{y} = C\widehat{\mathrm{F}} = \widehat{F}_e \end{cases} \tag{12}$$

Based on equation (10) and (12), the error dynamics is defined as follows:

$$\begin{aligned} \dot{\tilde{\boldsymbol{\mathcal{F}}}} = \dot{\boldsymbol{\mathcal{F}}} - \dot{\widehat{\boldsymbol{\mathcal{F}}}} &= A(\boldsymbol{\rho})\left(\boldsymbol{\mathcal{F}} - \widehat{\boldsymbol{\mathcal{F}}}\right) - \boldsymbol{\mathcal{L}}(y - \hat{y}) \\ &= A(\boldsymbol{\rho})\tilde{\boldsymbol{\mathcal{F}}} - \boldsymbol{\mathcal{L}}C\tilde{\boldsymbol{\mathcal{F}}} \end{aligned} \tag{13}$$

In which $\tilde{\mathcal{F}}$ is the estimation error vector and $\mathcal{L}$ is the observer coefficients vector: $\boldsymbol{\mathcal{L}} = [\,L_1\ L_2\,]$. The design aim is to obtain $\mathcal{L}$ in order to stabilize observer. A Lyapunov candidate function is considered as follows: $V = \frac{1}{2}\tilde{F}_i^2 + \frac{1}{2}\tilde{F}_e^2$. In which V is a positive definite function (PDF). If the time derivative of $V(\mathcal{F})$ in every system depended trajectory is negative, the estimation error is asymptotically stable to origin, which means the estimated values converge to real quantities. So, the derivation of system along system trajectories are calculated as follows:

$$\begin{aligned} \dot{V} = \dot{\tilde{F}}_i\tilde{F}_i + \dot{\tilde{F}}_e\tilde{F}_e &= \tilde{F}_i\left(\rho_1\tilde{F}_i + \rho_2\tilde{F}_e - L_1\tilde{F}_e\right) \\ &\quad + \tilde{F}_e\left(\rho_3\tilde{F}_i + \rho_4\tilde{F}_e - L_2\tilde{F}_e\right) \\ &= \rho_1\tilde{F}_i^2 + \rho_2\tilde{F}_i\tilde{F}_e - L_1\tilde{F}_i\tilde{F}_e \\ &\quad + \rho_3\tilde{F}_i\tilde{F}_e + \rho_4\tilde{F}_i\tilde{F}_e + \rho_4\tilde{F}_e^2 - L_2\tilde{F}_e^2 \\ &= \rho_1\tilde{F}_i^2 + (\rho_4 - L_2)\tilde{F}_e^2 + (\rho_2 + \rho_3 - L_1)\tilde{F}_i\tilde{F}_e \end{aligned} \tag{14}$$

Different methods are developed to force the $\dot{V}$ to be negative [3], [4]. In this paper, a simple yet conservative approach is employed to guarantee that derivative of Lyapunov function is negative along trajectories. Equation (14) shows that $\dot{V}$ is composed of 3 distinct terms. We aim at design $\mathcal{L}$ so that every three terms are always negative. It is obvious that it is a rigorous constraint on $\mathcal{L}$. We have:

a) Based on equation (9), ρ_1 is always negative: the first term $\rho_1\tilde{F}_i^2$ is always negative.
b) According to equation (9), ρ_4 is always negative: L_2 should be positive so that the second term always remains negative.
c) In the third term $(\rho_2 + \rho_3 - L_1)\tilde{F}_i\tilde{F}_e$, due to positive value of $\tilde{F}_i\tilde{F}_e$, the term $\rho_2 + \rho_3 - L_1$ should always be negative. Regarding equation (9) it is obvious that $\rho_2 + \rho_3 = \alpha\dot{m}_{egr} + \beta\dot{m}_{as}$, so L_1 should be obtained so that it is always greater than $\alpha\dot{m}_{egr} + \beta\dot{m}_{as}$.

So two necessary (but not sufficient) conditions on $\mathcal{L}$ vector is briefly:

$$\begin{aligned} &L_2 > 0 \\ &L_1 > \alpha\dot{m}_{egr} + \beta\dot{m}_{as} \end{aligned} \tag{15}$$

The first condition is simply applicable; anyway, in order to apply the second condition, $\dot{m}_{egr}$, $\dot{m}_{as}$, β and α, should be obtained in whole engine operation, so that $L_{\max} = \max(\alpha\dot{m}_{egr} + \beta\dot{m}_{as})$ is calculated in whole operating point. However the dynamic variation of $\alpha\dot{m}_{egr} + \beta\dot{m}_{as}$ factor should be considered for calculation of $L_{\max}$, in this research the steady state values of parameters are employed and a correction factor of 20% is employed for dynamic compensation. The result of $\rho_2 + \rho_3 = \alpha\dot{m}_{egr} + \beta\dot{m}_{as}$ in whole engine operation is shown in Fig. 3.

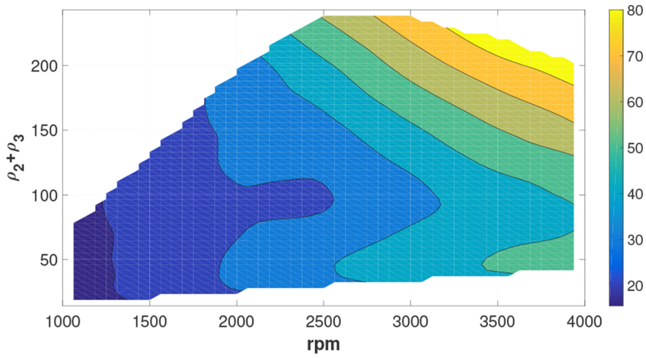

Fig. 3. Variation of $\rho_2 + \rho_3$ in whole engine operation.

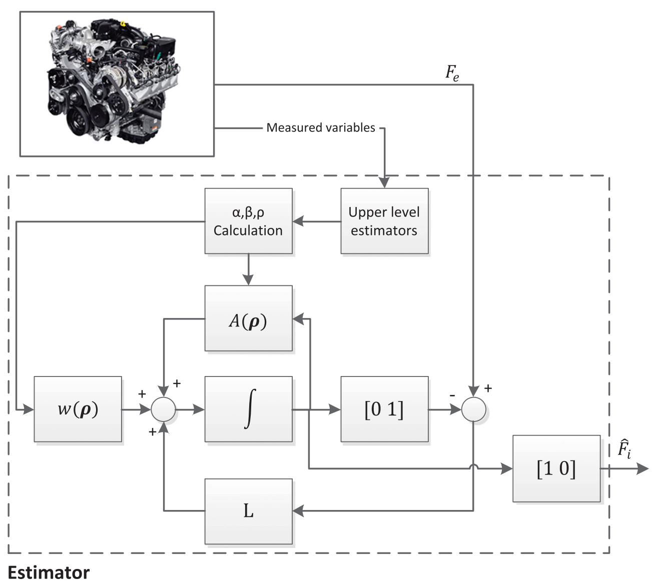

Fig. 4. Implementation of burned gas estimator in inlet manifold.

Based on Fig. 3, $L_{\max} = 88$, where by applying 1.2 safety factor, L_1 is choose 105 and $L_2 = 10$. It is seen that the maximum values for $\alpha \dot{m}_{egr} + \beta \dot{m}_{as}$ is obtained in high speed-high load condition of engine operation.

The implementation of burned gas estimators is depicted in Fig. 4. As shown in Fig. 4, in order to implement the designed stable observer, it is needed to calculate all ρ_i parameters as well as and α and β. It was shown earlier in equation (9) that calculating these parameters needs inlet and exhaust pressure and temperature as well as engine internal flows such as: cylinder aspirated flow rate ($\dot{m}_{as}$), recultaed gas mass flow rate ($\dot{m}_{egr}$), entering air to inlet manifold ($\dot{m}_c$), outgoing gas flow rate from exhaust manifold ($\dot{m}_{ex}$) and injected fuel rate ($\dot{m}_f$). $\dot{m}_c$ and $\dot{m}_{ex}$ are measured using appropriate sensors. Also $\dot{m}_f$ might be calculated using injector signal width and rail pressure. But other mass flow rates such as recirculated gas mass flow rate and cylinder aspirated mass flow rate are not directly measured and should be estimated. Also exhaust manifold temperature and pressure are hard to measure and need expensive sensors and it is needed to develop appropriate estimators for estimation of these parameters.

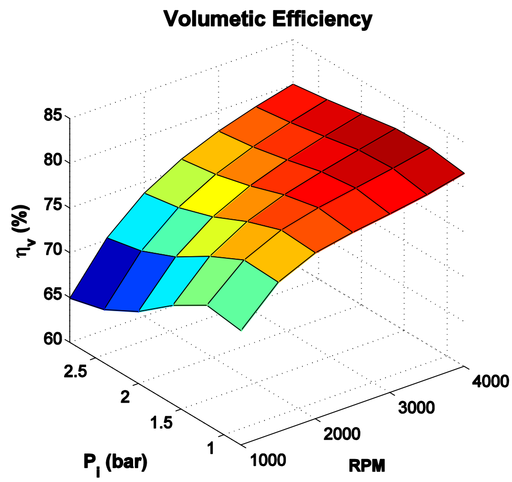

Fig. 5. Volumetric efficiency as a function of engine speed and inlet manifold pressure.

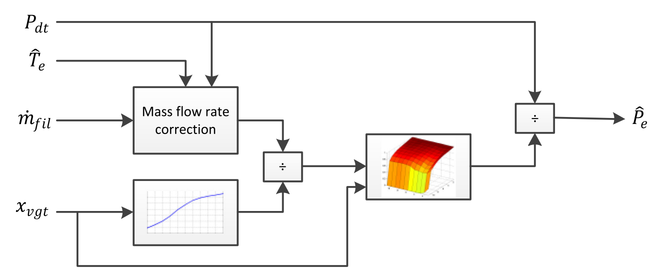

Fig. 6. Block diagram of exhaust manifold pressure open loop estimator implementation.

A. Cylinder Aspirated Mass Flow Estimator

In order to estimate aspirated air flow, density-volumetric efficiency model is employed as follows [18]:

$$\dot{m}_{as} = \frac{RPM}{60} \frac{P_i}{T_i R} \eta_v V_d \tag{16}$$

In which P_i and T_i are inlet manifold pressure and temperature respectively, η_v is volumetric efficiency and V_d is engine displacement volume. Volumetric efficiency is assumed to be a strict function of engine speed and inlet manifold pressure [19]. The function is obtained using experimental results and depicted in Fig. 5.

B. Recirculated Gases Mass Estimator

An important variable which should be calculated in estimation of fresh air in inlet manifold is recirculated gases mass flow rate. Due to the low mass flow rate of recirculated gases and cost of proper sensors, estimators are employed instead.

In order to develop the estimator, the equation of pressure variation is considered. The problem (estimation of mass flow rate) is defined as finding an unknown input to an integrator-like system, in which the output of integrator is known in every instant. Stosky and kolmansovski have categorized different methods of estimating such parameters [20]. The general form of integrator-like system is as follows:

$$\dot{z} = x + y \tag{17}$$

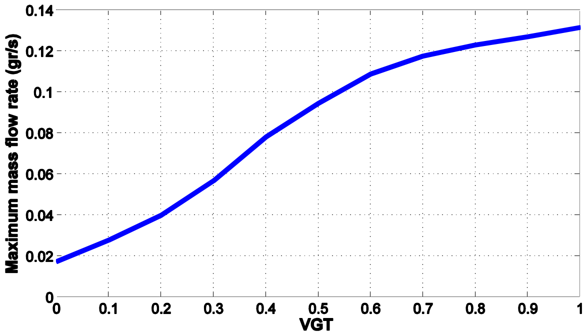

Fig. 7. Maximum turbine flow based on VGT position.

In which z and y are measurable parameters, while x is not measurable and should be estimated. The air manifold process is a such an integrator-like system. The observers which are employed to observe x parameters in processes like equation (17) are called unknown input estimators. Different method such as high gain observers, dirty derivative observers, Luenberger observers and sliding mode observers are employed to observe unknown parameters. In this paper high gain observer approach is employed to observe recirculated mass flow rate. Assume an integrator-like system just described in equation (17) as follows:

$$x = \dot{z} - y \tag{18}$$

By applying a low pass filter on equation (18):

$$\begin{aligned}\frac{1}{\tau s+1}x &= \frac{1}{\tau s+1}\ (\dot{z}-y) = \frac{1}{\tau s+1}\ (sz-y)\\ &= \frac{z}{\tau} - \frac{1}{\tau s+1}\left(\frac{z}{\tau}+y\right)\end{aligned} \tag{19}$$

In which τ is filter time constant. Assuming $\varepsilon = \frac{1}{\tau s+1}$ $(\frac{z}{\tau}+y)$ we will have:

$$\frac{1}{\tau s+1}x = \frac{1}{\tau}z - \varepsilon \tag{20}$$

The left part of equation (20) is the filtered value of parameters x and might be assumed to be an estimation of x ($\hat{x} = \frac{1}{\tau s+1}\ x$). In this manner we have:

$$\hat{x} = \frac{1}{\tau}\ z - \varepsilon \tag{21}$$

It should be noted that decreasing τ will increase the observer speed. In order to estimate the unknown parameter in equation (17), following observer may be used:

$$\begin{cases}\varepsilon = \dfrac{1}{\tau s+1}\ \left(\frac{z}{\tau}+y\right)\\ \hat{x} = \dfrac{1}{\tau}\ z - \varepsilon\end{cases} \tag{22}$$

In order to implement this observer on the manifold process, we have:

$$\begin{cases}\varepsilon\ = \dfrac{1}{\tau_g s+1}\ \left[\dfrac{P_i}{\tau_g} + \dfrac{RT_i}{V_i}\left(\dot{m}_c - \dot{m}_{as}\right)\right]\\ \widehat{\dot{m}}_{egr} = \dfrac{V_i}{RT_i}\ \left(\dfrac{1}{\tau_g}P_i - \varepsilon\right)\end{cases} \tag{23}$$

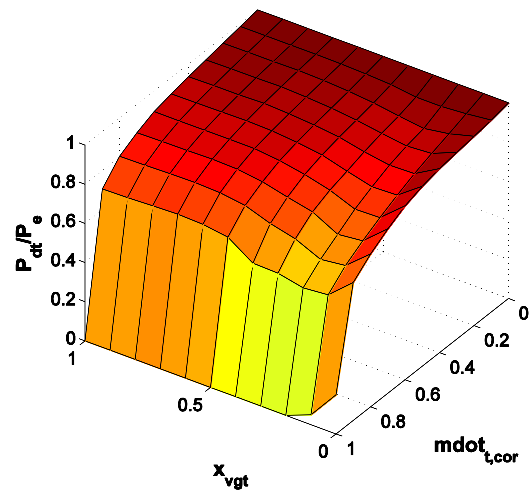

Fig. 8. Turbine pressure ratio as a function of normalized flow and VGT position.

In which $\widehat{\dot{m}}_{egr}$ is estimation of recirculated mass flowrate. In design of estimator τ is obtained to be 0.05 s.

C. Exhaust Pressure Estimation

The exhaust manifold pressure of engine varies between 2.5 to 4 bar in different operating conditions. Also exhaust temperature is in the range of 800 °C to 900 °C [1], [21]. The exhaust manifold condition is not appropriate for pressure sensors. That is why we employ an estimator for exhaust pressure. Different methods are suggested by researchers to estimate exhaust pressure; most of them are based on turbine performance and exhaust mass flow rate [22]. Outlet mass flow rate from exhaust manifold $\dot{m}_{fil}$ is usually measured using a pressure differential sensor which is attached after turbine and beside exhaust filter. Also the turbocharger governing equations and turbine performance map are available. Due to available information, an open loop estimator is employed to estimate exhaust pressure. In this paper, the estimation is done based on variable geometry turbine (VGT) map in [23]. As discussed in [23], mass flow rate through turbine is a function of pressure ratio and turbine blade angle. So the pressure ratio through turbine might be estimated based on turbine blade angle and mass flowrate through turbine, based on which and by measuring after turbine pressure P_{dt}, open loop estimator is employed to estimate exhaust manifold pressure. Finally, the exhaust pressure estimator is implemented as depicted in Fig. 6.

Turbine maps takes into account the corrected mass flowrate, so the exhaust measured mass flowrate is corrected based on exhaust pressure and temperature. On the other hand, the turbine flow is normalized based on maximum flow rate in each blade angle. The maximum turbine flowrate as a function of blade angle is depicted in Fig. 7. The normalized turbine mass flowrate and VGT angle is used to estimate turbine pressure ratio. The look-up table surface is depicted in Fig. 8.

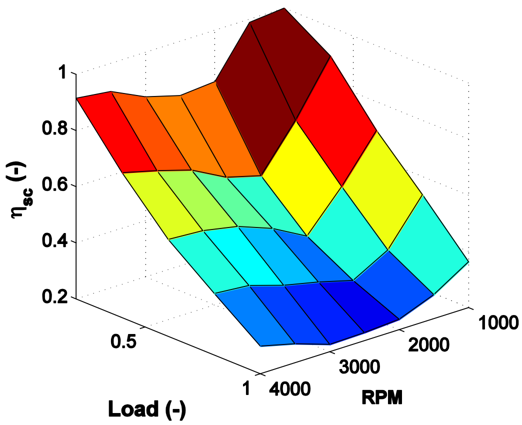

Fig. 9. Efficiency compensation factor as a function of engine speed and load.

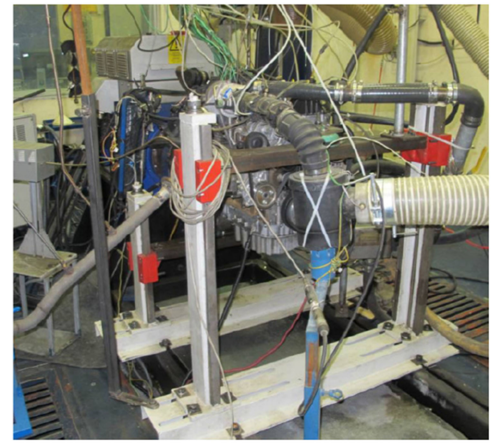

Fig. 10. Engine on test bench.

By measuring the pressure after turbine and estimating the pressure ratio through turbine, it is possible to estimate exhaust manifold pressure.

D. Exhaust Temperature Estimation

Some diesel engines are provided with a temperature sensor for measuring exhaust manifold temperature. However, in the under studied engine, no temperature sensor is attached on exhaust manifold. Instead an open loop estimator is developed for it. Exhaust manifold temperature is a dynamic variable which is affected by exhaust gas mass flow rate, in cylinder temperature in exhaust stroke, exhaust manifold condition and turbocharger performance. In this research, it is assumed that contained gas temperature in exhaust manifold equals the outlet gas from cylinder in exhaust stroke. However, the in-cylinder gas temperature is drastically variable in combustion process, it is possible to assume it constant while exhaust process. Based on Seliger cycle governing equations, the exhaust temperature is obtained from equation (24) [24]:

$$T_e = \eta_{sc}\left(\frac{P_e}{P_i}\right)^{1-\frac{1}{\gamma}}\left(\frac{1}{r}\right)^{\gamma-1}\left(1+\frac{q_{in}}{c_v T_1 r^{\gamma-1}}x_{cv}\right)^{\frac{1}{\gamma-1}} \times \left(q_{in}\left(\frac{1-x_{cv}}{c_p}+\frac{x_{cv}}{c_v}\right)+T_i r^{\gamma-1}\right) \quad (24)$$

In which η_{sc} is compensating factor of non-ideal cycle, x_{cv} is the fraction of burned fuel in constant volume process and q_{in} is released heat. Equation (24) shows that in order to estimate exhaust temperature, inlet gas temperature and pressure, exhaust pressure and released heat should be known. The inlet gas temperature and pressure is measured by a TMAP sensor attached in inlet manifold, while the exhaust pressure is estimated by the earlie developed estimator. Also released heat can be calculated based on fuel injection duration and rail pressure. Two parameters of η_{sc} and x_{cv} should be obtained based on static engine test data. Based on executed tests and result of [24], x_{cv} is assumed to be constant and 0.6 and η_{sc} is obtained as two variable function of engine load and speed as depicted in Fig. 9.

IV. Result and Discussion

Hardware in the loop (HiL) simulation is employed to implement the higher and lower level estimators. Matlab Real-time package is used for execution of estimation algorithm and a National Instrument NI 6009 USB data acquisition system is employed to transfer data to Matlab. Bosch boost pressure sensor is employed to measure pressure and temperature is inlet manifold.

To evaluate the observer performance, three different test procedures are conducted as depicted in Fig. 10. In the first test, the higher and lower observers are evaluated using proper affecting input signals. In the next test, the effects of atmospheric condition on observer performance is studied, while in the last test, the effects of error in lower observers are studied on performance of the high level observer.

In order to evaluate the higher level fresh air observer and also lower lever estimators, two test scenarios are executed on engine. In design of test scenarios, the affecting variables on inlet manifold BGF have varied and the estimated BGF is compared with real data. In the first scenario the EGR valve have stimulated with a square waveform pattern and in the second scenario the turbine blade angle has been harmonically varied. In the both cases the estimated BGF in inlet manifold, aspirated gas to cylinder, exhaust pressure and temperature are compared with test data. In order to evaluate the hierarchical estimator in whole engine operation, the test scenarios are done for four points with various load and speed. The operating points are chosen so that high speed-low load, high speed-high load, low speed-low load and low speed-high load operation are covered. In order to eliminate the controller operation on tests, the operation of low-level controllers is deactivated.

The result of first scenario test is depicted in Fig. 11 to Fig. 14. The result of estimation with EGR valve stimuli in low speed and low load operation is illustrated in Fig. 11. The results show that the BGF is estimated with high accuracy. The results also

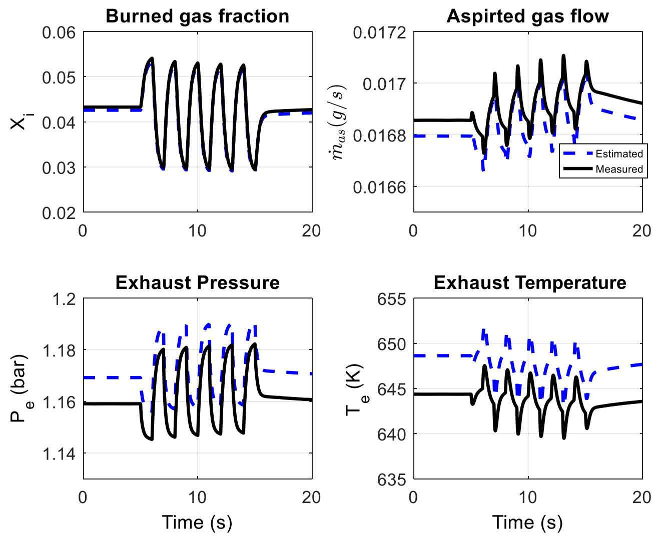

Fig. 11. Comparison of BGF, aspirated flow, exhaust pressure and temperature estimators with measured data in 1500 rpm and 20% load operating point in response to EGR valve square waveform opening.

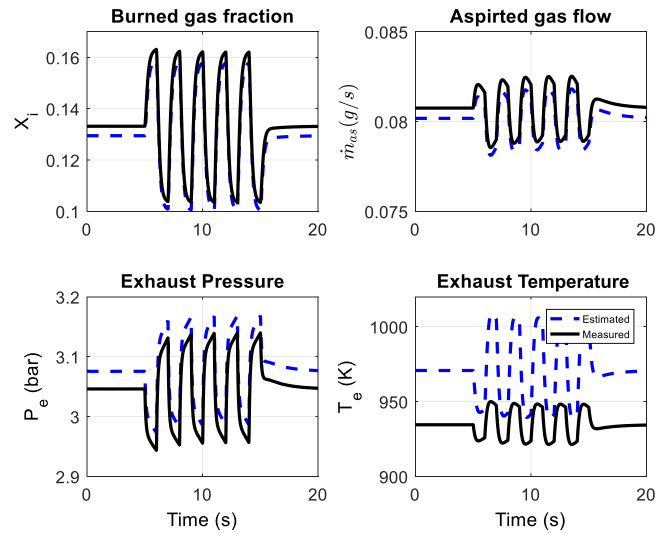

Fig. 13. Comparison of BGF, aspirated flow, exhaust pressure and temperature estimators with measured data in 3000 rpm and 80% load operating point in response to EGR valve square waveform opening.

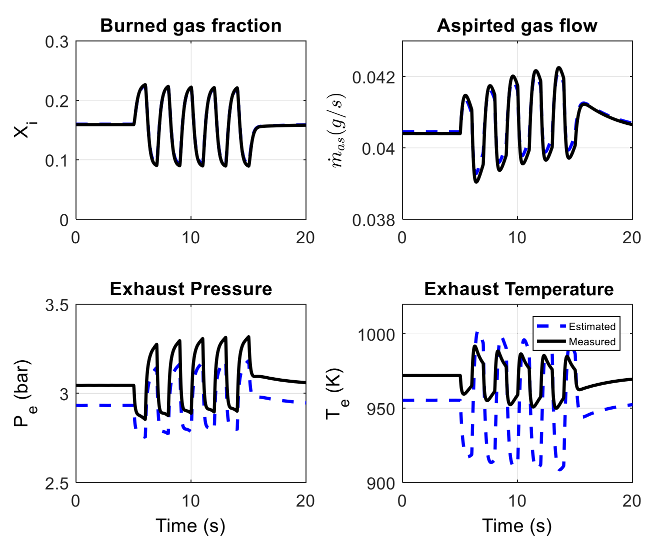

Fig. 12. Comparison of BGF, aspirated flow, exhaust pressure and temperature estimators with measured data in 2000 rpm and 80% load operating point in response to EGR valve square waveform opening.

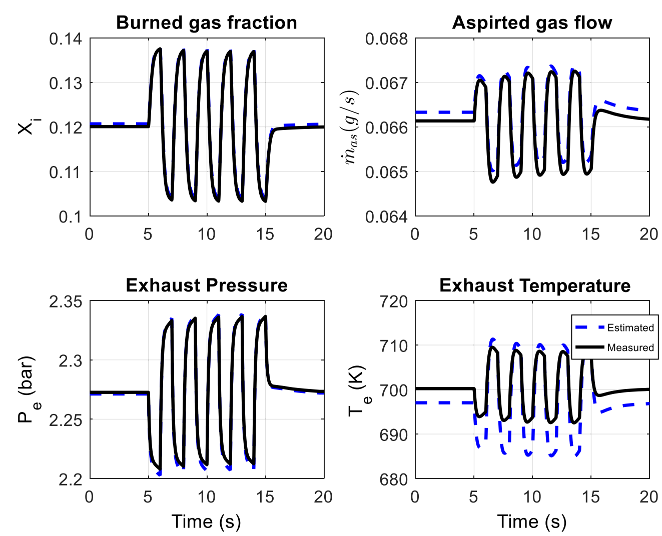

Fig. 14. Comparison of BGF, aspirated flow, exhaust pressure and temperature estimators with measured data in 3500 rpm and 20% load operating point in response to EGR valve square waveform opening.

show 1.8% error for steady state estimation which is tolerated for control applications. Also cylinder inducted gas estimator shows acceptable performance. The steady state estimation error is 1.6%. Also comparison between exhaust manifold pressure and measured values show 0.85% error for steady state condition. Exhaust manifold temperature estimation also shows 0.6% error for steady state condition.

The result of first scenario test in low speed-high load is depicted in Fig. 12. The results show that BGF and aspirated gas flow estimators are able to estimate variables with high accuracy. The exhaust manifold pressure shows 1.4% error in both steady and dynamic condition. The exhaust manifold temperature estimator shows a 7.1% drift in low temperature region in steady while the error drops for dynamic measuring. The results also show that the performance of upper level estimators does not decrease with estimation error in exhaust pressure and temperature.

The first scenario is done for high speed-high load and the results are depicted in Fig. 13. The steady state BGF observer error is 2.7% in this operating point. Also aspirated gas flow and exhaust pressure estimators show 1% and 1.2% error for steady state operation respectively. The exhaust temperature estimator shows 3.6% error for steady state operation while in dynamic operation it will increase to 5%. The main reason of estimation error in dynamic operation is neglecting manifold

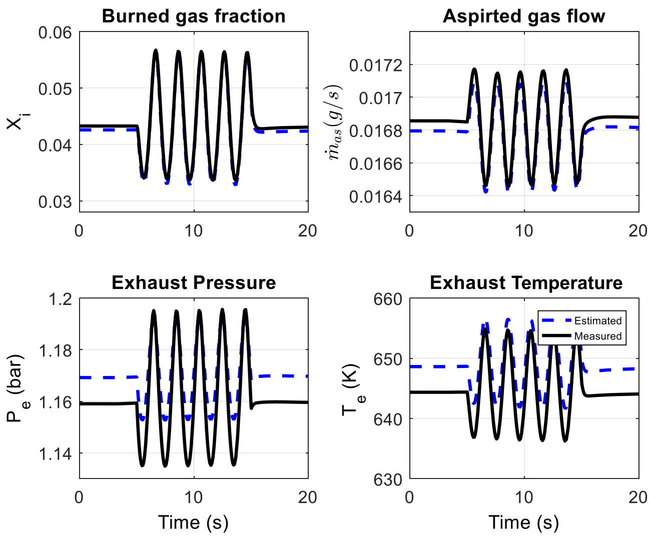

Fig. 15. Comparison of BGF, aspirated flow, exhaust pressure and temperature estimators with measured data in 1500 rpm and 20% load operating point in response to a sinusoidal VGT variation.

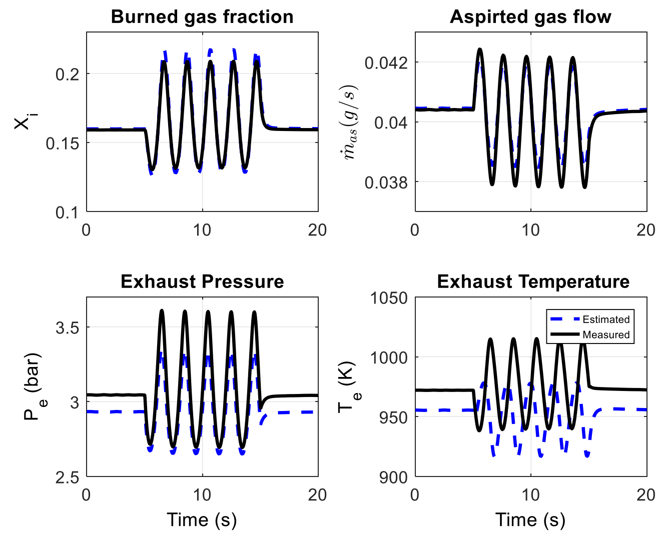

Fig. 16. Comparison of BGF, aspirated flow, exhaust pressure and temperature estimators with measured data in 2000 rpm and 80% load operating point in response to a sinusoidal VGT variation.

process dynamics in estimator construction, where the cylinder exhaust gas temperature is estimated instead of in-manifold gas.

Finally the first scenario is done for high speed and low load, where the test results are illustrated in Fig. 14. The results show that BGF estimation is done with high accuracy. Also the steady state error for aspirated air mass estimation and exhaust pressure estimation in both transient and steady conditions is acceptable. Exhaust temperature estimation error in steady state is about 1% and increases in transient mode.

In the second scenario, the performance of estimator is evaluated for a harmonic VGT blade angle variation of 0.5 Hz. Just like the first scenario, the evaluation is done for four operation condition and results are reported.

The result of estimators to VGT blade angle variation on low speed and low load is depicted in Fig. 15. It is shown that BGF in negligible. Also aspirated air flow estimator shows a steady state error of about 0.3% and the exhaust gas pressure and temperature estimator has less than 1% error in estimation.

The estimation error of estimators in response to VGT blade angle variation in low speed and high load is illustrated in Fig. 16. In this working operation, the main estimator error is for steady state is negligible; however, it reaches to 4% in dynamic processes. Also the aspirated gas flow estimation is done with high accuracy. The steady state error for exhaust pressure and temperature is 3% and 1% respectively.

Also the test has been done for high load and high speed and the result is depicted in Fig. 17. The main estimator error in steady state is 3% and the other estimator performances are in accepted range, however estimation error of the exhaust temperature is 3%.

The estimator results and real data are compared in high speed and high load are illustrated in Fig. 18. In this condition, the results of main estimator show acceptable accuracy. However, the aspirated gas flow shows a slight error for steady state condition.

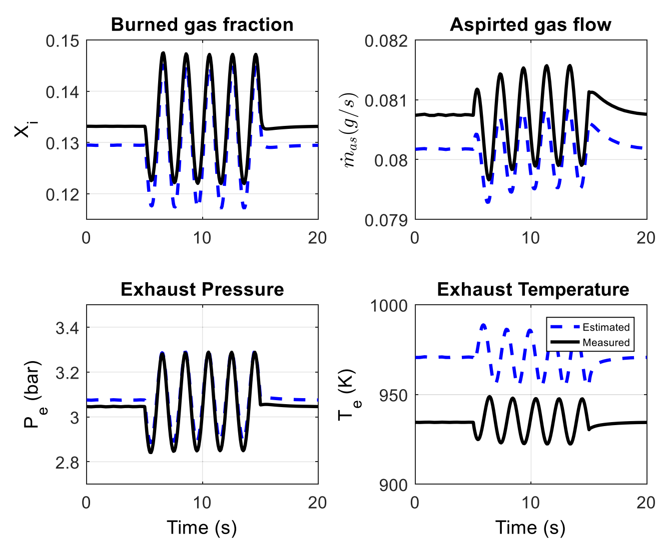

Fig. 17. Comparison of BGF, aspirated flow, exhaust pressure and temperature estimators with measured data in 3000 rpm and 80% load operating point in response to a sinusoidal VGT variation.

In this condition, exhaust pressure is done with high accuracy. Also open loop exhaust temperature estimator shows acceptable performance and about 0.3% error in steady state estimation.

In order to study the effects of atmospheric conditions on observers, two tests are done in nonstandard atmospheric conditions. In the first test, the condition of test Fig. 11 is done in atmospheric condition of 90 kPa pressure and temperature of 270 K. The results of test is depicted in Fig. 19. The results show that the higher level observe can estimate the level of burned gas with sufficient accuracy. Also the exhaust manifold pressure is estimated better than normal condition.

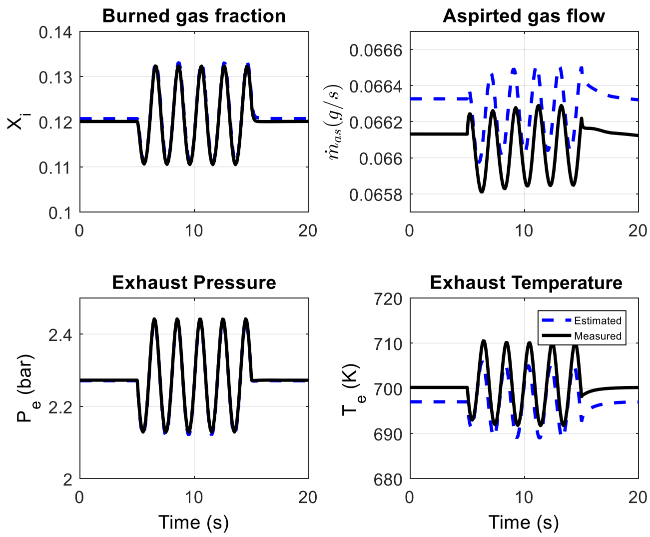

Fig. 18. Comparison of BGF, aspirated flow, exhaust pressure and temperature estimators with measured data in 3500 rpm and 20% load operating point in response to a sinusoidal VGT variation.

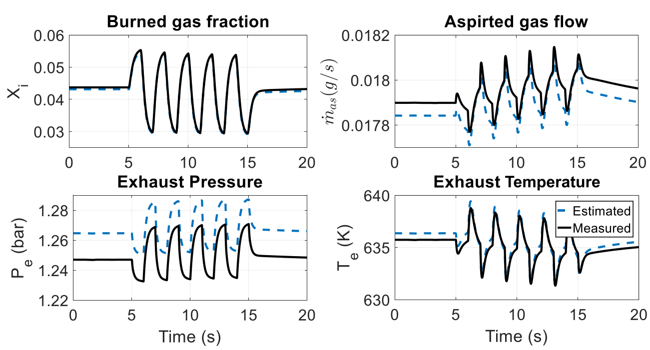

Fig. 19. Comparison of BGF, aspirated flow, exhaust pressure and temperature estimators with measured data in 1500 rpm and 20% load operating point in response to EGR valve square waveform opening in $P_i = 90$ kPa and $T_i = 270$ K.

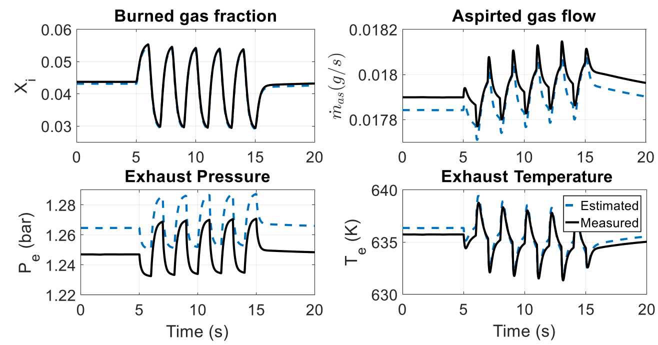

Fig. 20. Comparison of BGF, aspirated flow, exhaust pressure and temperature estimators with measured data in 1500 rpm and 20% load operating point in response to EGR valve square waveform opening in $P_i = 110$ kPa and $T_i = 320$ K.

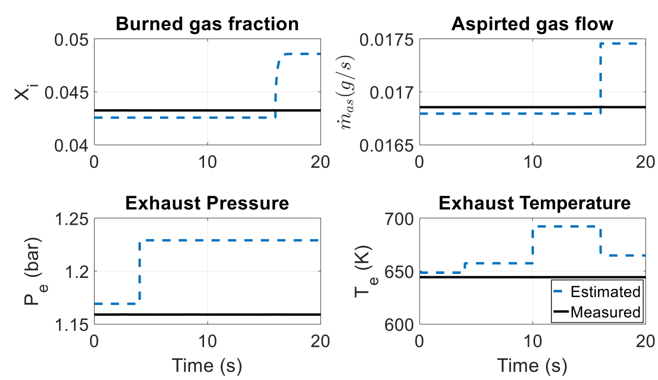

Fig. 21. Effects of lower level estimator error on higher level BGF observer.

Also the atmospheric test is done in 110 kPa pressure and 320 K conditions. Again the test is done in the same condition of Fig. 11. The results of test is depicted in Fig. 20. The results show that the higher level observer can estimate the BGF with desired accuracy. Also, the exhaust manifold pressure error is increased while the exhaust manifold temperature estimator performance is promoted in this atmospheric conditions.

The last test is done to study the effect of error in lower estimators on higher level observer. The values of lower level estimators are changed and the results are considered on both lower level and high level estimators as depicted in Fig. 21. 3 lower level estimator values are increased 20% from estimated value and the estimated values are employed to observe the BGF in inlet manifold. The results show that the aspirated gas flow estimation is affective on BGF observed value while the exhaust pressure and temperature estimation effects on higher level observer is negligible.

V. Conclusion

EGR control is a main task in diesel engine management systems. In order to properly control the EGR system, BGF in inlet manifold should be controlled precisely. Unfortunately, the available sensing systems for measurement of inlet manifold BGF are too expensive and complex to be used in convectional engine management systems. Using advantage of universal exhaust gas oxygen, estimators are employed to estimate BGF instead. Since different unknown parameters should be available in estimation of BGF in inlet manifold, in this paper a hierarchical estimation scheme was developed to estimate the BGF in inlet manifold of a diesel engine. A dynamic model was developed for air fraction in inlet and exhaust manifold, based on which a linear parameter varying is derived. The derived LPV system contains some unknown and hard to measure parameters which are function recirculated gas flowrate, inducted mass flowrate and exhaust manifold pressure and temperature, therefore appropriate estimators are developed besides the main estimator. Lyapunov stability theorem is applied to stabilize the higher level estimator, while dirty derivative, unknown input observer and open loop estimation methods are employed to design lower level estimators. Since the precise operation of higher level estimator is affected by lower level estimators, the influence of each lower estimator is studied on inlet manifold BGF estimation. The evaluation of overall performance of the proposed hierarchical estimation scheme estimator shows that it is able to estimate BGF with high accuracy in both transient and steady state operation. However, the performances of other estimators are investigated simply to evaluate the main estimator and the effects of lower level estimators on the main estimator performance, their operation falls in the acceptable range for whole engine operation. Among lower level estimators, the exhaust temperature estimator shows an error of more than 5% for estimation in some specific operation points, which is mainly due to omitting exhaust manifold dynamic processes. Furthermore, it is shown that the accuracy of exhaust pressure and temperature estimation is not influencing the BGF estimator while the aspirated gas flow has significant effects of main estimator performance.

References

[1] K. Nikzadfar and A. H. Shamekhi, "Investigating the relative contribution of operational parameters on performance and emissions of a common-rail diesel engine using neural network," *Fuel*, vol. 125, pp. 116–128, 2014.

[2] Y. Park, K. Min, J. Chung, and M. Sunwoo, "Control of the air system of a diesel engine using the intake oxygen concentration and the manifold absolute pressure with nitrogen oxide feedback," *Proc. Institution Mech. Eng. Part D J. Automobile Eng.*, vol. 230, no. 2, pp. 240–257, 2016.

[3] K. Nikzadfar and A. H. Shamekhi, "More than one decade with development of common-rail diesel engine management systems: A literature review on modelling, control, estimation and calibration," *Proc. Institution Mech. Eng. Part D J. Automobile Eng.*, vol. 229, no. 8, pp. 1110–1142, 2015.

[4] K. Min, J. Shin, D. Jung, M. Han, and M. Sunwoo, "Estimation of intake oxygen concentration using a dynamic correction state with extended Kalman filter for light-duty diesel engines," *J. Dyn. Syst. Meas. Control*, vol. 140, no. 1, 2018, Art. no. 011013.

[5] J. Wang, "Air fraction estimation for multiple combustion mode diesel engines with dual-loop EGR systems," *Control Eng. Pract.*, vol. 16, no. 12, pp. 1479–1486, 2008.

[6] F. Castillo, E. Witrant, V. Talon, and L. Dugard, "Simultaneous air fraction and low-pressure EGR mass flow rate estimation for diesel engines," *IFAC Proc. Volumes*, vol. 46, no. 2, pp. 731–736, 2013.

[7] S. D'Ambrosio, D. Iemmolo, A. Mancarella, N. Salamone, R. Vitolo, and G. Hardy, "Zero dimensional models for EGR mass-rate and EGR unbalance estimation in diesel engines," SAE Technical paper 2017-24-0070, 2017.

[8] S. Diop, P. E. Moraal, I. V. Kolmanovsky, and M. Van Nieuwstadt, "Intake oxygen concentration estimation for DI diesel engines," in *Proc. IEEE Int. Conf. Control Appl.*, 1999, vol. 1, pp. 852–857.

[9] P. Chen and J. Wang, "Observer-based estimation of air-fractions for a diesel engine coupled with aftertreatment systems," *IEEE Trans. Control Syst. Technol.*, vol. 21, no. 6, pp. 2239–2250, Nov. 2013.

[10] X. Zeng and J. Wang, "A physics-based time-varying transport delay oxygen concentration model for dual-loop exhaust gas recirculation (EGR) engine air-paths," *Appl. Energy*, vol. 125, pp. 300–307, 2014.

[11] J. Zhao and J. Wang, "Observer based oxygen fraction estimation for a dual-loop EGR diesel engine fueled with biodiesel blends," in *Proc. ASME Dyn. Syst. Control Conf.*, 2013, pp. V001T11A001.

[12] F. Liu and J. Pfeiffer, "Estimation algorithms for low pressure cooled egr in spark-ignition engines," *SAE Int. J. Engines*, vol. 8, no. 4, pp. 1652–1659, 2015.

[13] H. Wang, Q. Qu, and Y. Tian, "Nonlinear observer based sliding mode control and oxygen fraction estimation for diesel engine," *Trans. Inst. Meas. Control*, vol. 40, no. 7, pp. 2227–2239, 2018.

[14] H. Lee, Y. Park, and M. Sunwoo, "Observer design for exhaust gas recirculation rate estimation in a variable-geometry turbocharger diesel engine using a model reference identification scheme," *Proc. Inst. Mech. Eng. Part D J. Automobile Eng.*, vol. 228, no. 14, pp. 1688–1699, 2014.

[15] A. Stotsky and I. Kolmanovsky, "Simple unknown input estimation techniques for automotive applications," in *Proc. IEEE Amer. Control Conf.*, 2001, vol. 5, pp. 3312–3317.

[16] Y.-Y. Wang and I. Haskara, "Exhaust pressure estimation and its application to detection and isolation of turbocharger system faults for internal combustion engines," *J. Dyn. Syst. Meas. Control*, vol. 134, no. 2, 2012, Art. no. 021002.

[17] F. Castillo, E. Witrant, L. Dugard, and V. Talon, "Exhaust manifold pressure estimation diesel equipped with a VGT turbocharger," SAE Technical paper 2013-01-1752, 2013.

[18] J. B. Heywood, *Internal Combustion Engine Fundamentals*. New York, NY, USA: McGraw-Hill, 1988.

[19] T. Polóni, B. Rohal'-Ilkiv, and T. A. Johansen, "Mass flow estimation with model bias correction for a turbocharged diesel engine," *Control Eng. Pract.*, vol. 23, pp. 22–31, 2014.

[20] A. Stotsky and I. Kolmanovsky, "Application of input estimation techniques to charge estimation and control in automotive engines," *Control Eng. Pract.*, vol. 10, no. 12, pp. 1371–1383, 2002.

[21] H. Bauer, *Diesel-Engine Management*. Hoboken, NJ, USA: Wiley, 2004.

[22] H. Jin, S. Choi, and S. Kim, "Design of a compressor-power-based exhaust manifold pressure estimator for diesel engine air management," *Int. J. Automot. Technol.*, vol. 15, no. 2, pp. 191–201, 2014.

[23] K. Nikzadfar and A. H. Shamekhi, "An extended mean value model (EMVM) for control-oriented modeling of diesel engines transient performance and emissions," *Fuel*, vol. 154, pp. 275–292, 2015.

[24] P. Skogtjärn *et al.*, "Modelling of the exhaust gas temperature for diesel engines," M.S. thesis, Dept. Elect. Eng., Linköping Univ., Linköping, Sweden, 2002.

Kamyar Nikzadfar received the B.Sc. degree in mechanical engineering from the Ferdowsi University of Mashhad, Mashhad, Iran, in 2006, the M.Sc. degree in automotive engineering from the Iran University of Science and Technology, Tehran, Iran, in 2009, and the Ph.D. degree in mechanical engineering from the K.N. Toosi University of Technology, Tehran, Iran, in 2015. After more than ten years of experience in automotive industry, he joined the Babol Noshirvani University of Technology in 2016 as an Assistant Professor of mechanical engineering. His research interests include control engineering, mechatronics, thermotronics and control of powertrain systems. He is a member IFAC-Automotive technical committee.

Amir H. Shamekhi received the B.Sc. degree in mechanical engineering from the University of Tehran, Tehran, Iran, in 1994, and the M.Sc. and Ph.D. degrees in mechanical engineering from the K.N. Toosi University of Technology (KNTU), Tehran, Iran in 1997 and 2004, respectively.

He is an Associate Professor of Mechanical Engineering Faculty with the KNTU since 2006. His fields of research are automotive control systems, mechatronics systems, and internal combustion engines.