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Decoupled, Disturbance Rejection Control for A Turbocharged Diesel Engine with Dual-loop EGR System

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Abstract: Dual-loop exhaust gas recirculation (EGR) with variable geometry turbocharger (VGT) provides a flexible architecture for achieving desired intake manifold conditions, including temperature, pressure and oxygen concentration, which are critical for advanced combustion mode control in modern engines. However, the highly nonlinear system dynamics and strong coupling effects between loops make the widely used, control oriented model failed to trace the system states well, especially in transient situations. This paper presents a control method for this nonlinear, multi-input multi-output (MIMO) system with the compensation for modeling error and disturbance. By taking both the advantages of the knowledge of system dynamics and the idea of disturbance rejection, the modified active disturbance rejection control (ADRC) with extended states observer (ESO) was utilized to achieve such a control objective. Here, the term "disturbance" lumps all the internal and external errors, including the modeling errors and coupling effects. By actively estimating the disturbance and compensating its effects, the intake manifold conditions can be controlled in a decoupled fashion. The proposed method was validated through high-fidelity GT-Power model simulations.

Keywords: Disturbance rejection control, Extended state observer, System decoupling, Diesel engine, intake conditions.

1. INTRODUCTION

To reduce the more and more strictly regulated engine out emissions and meanwhile increasing the thermal efficiency, advanced combustion modes, such as homogeneous charge compression ignition (HCCI) (Lü et al. 2005), premixed charge compression ignition (PCCI) (Borgqvist 2013) as well as low temperature diffusion combustion (LTDC) (Wang 2008), offer an promising solution. However, the advanced combustion modes usually requires narrower thermodynamic boundary conditions, such as the in-cylinder temperature, pressure and oxygen concentration (Park et al. 2010). Thus the control of in-cylinder conditions is critical to achieve stable and smooth multiple advanced combustion mode operations.

Dual-loop exhaust gas recirculation (EGR) system (Park et al. 2010)(Hosseini et al. 2014), with variable geometry turbocharger (VGT) (as shown in Fig.1), provides a flexible and also complicated control architecture which has been widely utilized in modern engines to achieve desired intake manifold conditions and further the in-cylinder conditions. With three actuators in such an engine setup, high pressure loop (HPL) EGR valve, low pressure loop (LPL) EGR valve, and the rack position of VGT, as indicated in Fig.1, intake manifold conditions, including the temperature, pressure and oxygen concentration can be well controlled.

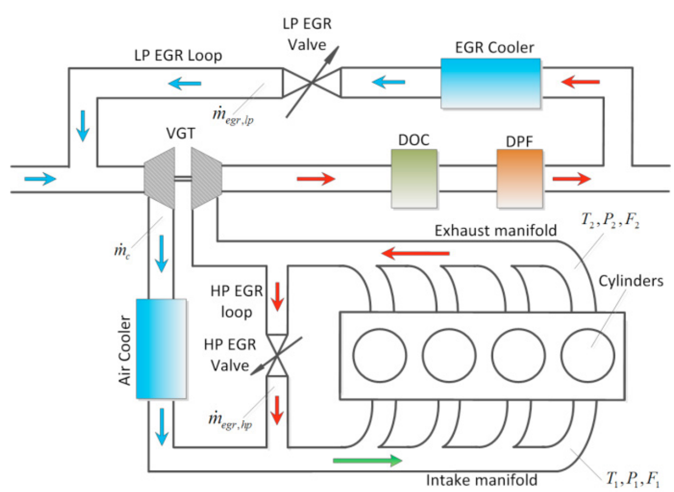

Fig.1. Engine architecture with a variable geometry turbocharger and dual-loop EGR system

Several strategies for controlling dual-loop EGR system have been demonstrated. A sliding-mode controller was utilized in (Wang 2008) for switching between conventional combustion and low temperature combustion mode. A generic model based control structure for Diesel engine with dual-loop EGR and VGT was described in (Grondin et al. 2009). Air-path and fuel-path coordinated control based on reference governor is designed in (Yan & Wang 2012). Control of dual-loop EGR engine air-path system with adjustable priorities by combining

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the reference governor technique and feedforward control is demonstrated in (Zeng & Wang 2014). For simultaneously control the intake manifold temperature, pressure and burned gas fraction with lower overshoot and shorter settling time, LQR with feedforward control is introduced in (Chen & Yan 2015).

For all the above mentioned strategies, same or similar system dynamic models are applied. The deriving of such control-oriented model can be summarized as: 1) the intake temperature dynamics is derived under the assumption that the heat transfer between gas and pipe walls are ignorable (Karlsson 2001)and ; 2) the pressure dynamics is obtained by taking the derivative of the ideal gas law and then combining with the temperature dynamics; 3) the deriving of the oxygen concentration (or burned gas fraction) dynamics is on the basis of applying the mass conservation law and ideal gas law. As can be implied that the failure of taking the heat transfer into consideration will bring nontrivial error not only to the temperature and pressure dynamics, but also to the oxygen concentration dynamics, especially in transient situations when the mass flow rate can hardly be calculated accurately. The modeling error combines with the high coupling effects between the three loops (temperature, pressure and oxygen concentration)account for the necessity that lots of control techniques require feedforward loops (Zeng & Wang 2014) (Chen & Yan 2015) or looking up tables (Grondin et al. 2009).

Since the heat transfer and the coupling effects between loops are prohibitively complicated for the control purpose, an active distance rejection control (ADRC) (Zheng et al. 2009)to handle the according modeling error and coupling effects in this paper. The dual-loop EGR system is decoupled into 3 separate loops. For each loop the coupling effects and modeling error are lumped into one term, called disturbance term. The disturbance term is treated as an additional state and estimated by an extended state observer(ESO). Its impact on the final output is compensated and then eliminated by the control law in real time. For such a nonlinear, multi-input-multi-output, strong coupled system, the ADRC is modified to take all the existing knowledge into the estimating and controlling process to increase the responses and accuracy.

The arrangement of the rest of this paper is as follows: In section II, Modeling for intake manifold conditions are summarized. The ADRC controller is designed in section III. Simulation study is introduced in section IV. Conclusive remarks are given in section V.

2. SYSTEM MODELING

The control target system is a modern four-cylinder engine as shown in Fig.1. A VGT, high pressure loop (HPL) EGR and low pressure loop (LPL) EGR, LPL cooler, air cooler and the after-treatment system are equipped in this engine. The dynamic models of each part of the engine air-path loop are presented in the following subsections.

2.1 Intake Manifold Conditions

Neglecting the heat transfer between the mixture and pipe walls, and applying the energy and mass conservation law (Park et al. 2010)(Chen & Yan 2015), the intake manifold conditions dynamics is derived as:

$$\begin{cases} \dot{T}_1 = \dfrac{RT_1^2}{P_1V_1}\dot{m}_{cyl}(1-\gamma) + \dfrac{RT_1}{P_1V_1}(\gamma T_{ac} - T_1)\dot{m}_c + \\ \quad \dfrac{RT_1}{P_1V_1}(\gamma T_2 - T_1)\dot{m}_{egr,hp} \\ \dot{P}_1 = -\dfrac{R\gamma}{V_1}\dot{m}_{cyl}T_1 + \dfrac{R\gamma}{V_1}T_{ac}\dot{m}_c + \dfrac{R\gamma}{V_1}T_2\dot{m}_{egr,hp} \\ \dot{F}_1 = \dfrac{RT_1}{P_1V_1}(F_{air} - F_1)\dot{m}_{maf} + \dfrac{RT_1}{P_1V_1}(F_2 - F_1)\dot{m}_{egr,hp} + \\ \quad \dfrac{RT_1}{P_1V_1}[\breve{F}_2 - F_1]\dot{m}_{egr,lp} \end{cases} \tag{1}$$

where T_{ac} is the temperature in the area after the compressor (downstream of the compressor):

$$T_{ac} = \frac{m_{maf}T_{air} + m_{egr,lp}[\eta_{ic}T_{coolant} + (1-\eta_{ic})T_2]}{m_{maf} + m_{egr,lp}}, \tag{2}$$

and T, P and F represent temperature, pressure and oxygen concentration fraction, respectively; $\dot{m}_c$, $\dot{m}_{egr,hp}$, and $\dot{m}_{egr,lp}$ are the mass flow rate of the gas flow through compressor, the valve in the high-pressure loop and the valve in the low-pressure loop respectively ; m_{maf} is the mass of air flow; T_{cyl} and is the temperature in the cylinder; R and γ are the ideal gas constant and specific heat ratio; subscript 1, 2 stand for the intake and exhaust manifolds respectively; $\breve{F}_2$ is the delayed oxygen concentration due to long distance, and can be simplified as a constant delay to F_2:

$$\breve{F}_2(t) = F_2(t-\tau). \tag{3}$$

In (1), $\dot{m}_{cyl}$ is the engine intake gas flow rate, which can be calculated by speed-density models as:

$$\dot{m}_{cyl} = \frac{\lambda_v P_1 V_d N}{120RT_1}, \tag{4}$$

where λ_v is the volumetric efficiency which need to be calibrated, N is the engine speed and V_d is the swept volume.

The mass flow rate through valves, including $\dot{m}_{egr,hp}$ and $\dot{m}_{egr,lp}$, can be calculated by the orifice equations (Moulin & Chauvin 2008) as follows:

a) When the flow is not choked ($\frac{p_d}{p_u} > \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)}$),

$$\dot{m} = \frac{C_D A_R p_u}{\sqrt{RT_u}}\left(\frac{p_d}{p_u}\right)^{1/\gamma}\left\{\frac{2\gamma}{\gamma-1}\left[1-\left(\frac{p_d}{p_u}\right)^{(\gamma-1)/\gamma}\right]\right\}^{1/2}, \tag{5}$$

b)When the flow is choked ($\frac{p_d}{p_u} \le \left(\frac{2}{\gamma+1}\right)^{\gamma/(\gamma-1)}$),

$$\dot{m} = \frac{C_D A_R p_u}{\sqrt{RT_u}}\sqrt{\gamma}\left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/2(\gamma-1)}, \tag{6}$$