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Model-Based Quantitative Feedback Control of EGR Rate and Boost Pressure for Turbocharged Diesel Engines

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Abstract: This paper proposes a closed loop multivariable VGT/EGR control system for a turbocharged diesel engine. The control system is synthesized based on quantitative feedback theory to maintain robust stability and performance in the presence of model variations via sequential MIMO loop-shaping. Simulation results from a turbocharged diesel engine are included to illustrate the effectiveness of the proposed control design.

I. Introduction

Advanced diesel engines are equipped with common rail fuel-injection systems, exhaust gas recirculation (EGR) system, and variable geometry turbo chargers (VGT). Exhaust gas recirculation is used to decrease combustion flaming temperature and reduce NOx emissions. For a typical multi-input multi-output (MIMO) air charging control problem, an EGR valve modulates the amount of EGR, whereas a VGT modulates the compressor power to deliver desired boost pressure and maintain positive engine delta pressure for EGR flow.

This MIMO control problem is very challenging since both control loops are coupled ([7], [8]). Furthermore, this MIMO plant is nonlinear, and designing such a complicated system requires extensive calibration work.

In the past, different approaches ([5], [7], [8], [11], [12], [13], [14]) were applied to this diesel air charging control problem. Those included sliding mode control [13], passivation control design [11], and constructive Lyapunov control design [14]. In [8], the design of two single-loop controllers was presented for an air charging system based on a nonlinear mean value model of a turbo charged diesel engine.

In other approaches, adaptive gain scheduling is often used to control a nonlinear system based on its linearized models. The control gains are tuned at selected operating conditions. The tuning, however, is often based on ad-hoc trial and error methods. Among robust adaptive controls, [5] applied an H-infinity control to a linear parameter varying (LPV) model of an air charging system. The method in [5] seems to be a good robust solution for adaptive gain scheduling. However, an H-infinity control often yields a higher order control structure.

To address those design issues above, this paper presents an approach that designs a robust controller for the advanced air charging system. The paper is organized as follows. Section II includes a statement of the control problem and requirements. In Section III, linear multi-variable models are identified for an air charging system at selected engine operating conditions. The linearized models are analyzed in terms of loop couplings from pole/zero locations. In Section IV, the open-loop system is compensated and decoupled, which makes the system become diagonally dominant over the desired frequency bandwidth. Section V describes how to set up QFD bounds for the control system performances, such as stability, robustness, disturbance rejection, and tracking. The controllers that meet these design requirements are solved by a QFD design. In the final section, design validation is illustrated. Validation results have shown potential advantages for this proposed approach.

II. Statement of EGR/VGT Control Problem

A typical diesel engine plant as illustrated in Fig.1 includes a power cylinder, a variable geometry turbo-charger (VGT), a charge air cooler (CAC), an EGR cooler, and an EGR valve. An exhaust pipe from the engine is connected to an exhaust aftertreatment system that typically consists of an oxidation catalyst (DOC) and a diesel particulate filter (DPF). The level of EGR in the system is modulated by an EGR valve, which adjusts the effective flow area between intake and exhaust manifolds. On the other hand, the variable geometry turbocharger uses the energy from the exhaust flow to raise the intake boost pressure and the engine delta-pressure in order to deliver EGR flow.

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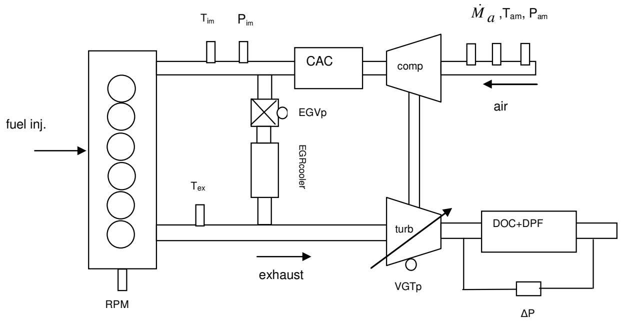

Figure 1: Diesel engine equipped with VGT and EGR systems

Shown in Fig.2 is the schematic diagram of this multivariable air-handling control system. The two actuators that provide inputs to the engine plant include the VGT position and the EGR valve position. Both the EGR rate and the boost pressure are selected as the default outputs.

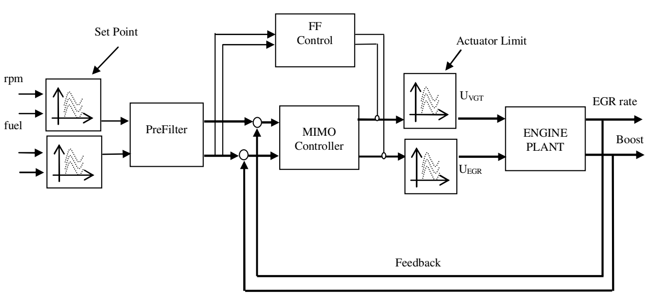

Figure 2: A MIMO control architecture for air handling control system

In this control system, the feed-forward control (FF Control in Fig.2) is essentially an inverse of the plant dynamics, which generates default actuator positions from the control set points (usually scheduled by engine speed and load) as shown in Fig.2. Our goal is to design a multi-variable feedback controller, such that this closed-loop control system meets stability and performance requirements with guaranteed robust margins against plant uncertainties and nonlinearities. The system should also achieve certain level of decoupling between the EGR loop and the boost pressure loop; the system should reject disturbances and exhibit good tracking capability as well. Furthermore, tuning of the controller is desired to be systematic and straightforward for easy calibration.

III. System Identification of EGR/VGT System

As the first step to design gain-scheduling control, a set of linear models are identified from the nonlinear engine plant at selected operating points. For this particular application, superposed sinusoid signals have been used as persistent excitation to the plant for model identification. To determine the frequency range for persistent excitation, the power spectrums of the command signals are analyzed for the boost pressure and the EGR rate commands during a FTP-75 transient emission test. From both spectrums, 96% of energy is found to be concentrated within the 0.5 Hertz frequency band. Since the responses of the air system are required to track the set point commands, the frequency range of the system identification should be at least larger than 0.5 Hertz (π rad/s), and possibly extend beyond the system cut-off frequency:

$$\omega = (0 \quad \omega_H], \quad \omega_H > \max(\pi,\ \omega_{CUT\,OFF}) \tag{1}$$

In this study, we identify a linearized state space system for the air handling system of a diesel engine using the technique developed in [6]. In review of the mean-value physical models of an air handling system derived by applying the conservation of mass and energy, the air path dynamics can be described effectively by a 5th order nonlinear system as follows:

$$\frac{dm_{im}}{dt} = \dot{m}_a + \dot{m}_{egr} - \dot{m}_o \tag{2.1}$$

$$\frac{dP_{im}}{dt} = \frac{R}{c_{v_{im}} V_{im}}\left[\dot{m}_a T_a c_{p_{im}} + \dot{m}_{egr} T_{egr} c_{p_{egr}} - \dot{m}_o T_{im} c_{p_{im}}\right] \tag{2.2}$$

$$\frac{dm_{em}}{dt} = \dot{m}_{ex} - \dot{m}_{egr} - \dot{m}_t \tag{2.3}$$

$$\frac{dP_{em}}{dt} = \frac{R_e}{c_{v_{em}} V_{em}}\left[\dot{m}_{ex} T_{ex} c_{p_{ex}} - (\dot{m}_{egr} + \dot{m}_t) T_{em} c_{p_{em}}\right] \tag{2.4}$$

$$J_t \frac{dN_t}{dt} = \frac{\eta_m c_{pex} \dot{m}_t (T_{em} - T_{tot}) - \dot{m}_{air} c_{pair} (T_{\cot} - T_{amb})}{N_t} \tag{2.5}$$

where, the five states are the air mass m_{im} of the intake volume, the exhaust mass m_{em} in the exhaust manifold, the boost (intake manifold) pressure P_{im}, the exhaust manifold pressure P_{em}, and the turbo speed N_t. The other parameters in Eqn.(2) represent mass flows through different volumes, temperatures, and specific heat coefficients. By neglecting the mass balance equations for the intake and exhaust volumes, the model can be further reduced to a third order differential equation with the boost pressure, the exhaust pressure, and the turbo speed as the state variables. This observation indicates that one can select either a 5th order or a 3rd order model structure for the identified linear system.

As an example, a linear model is identified from a 4.9L diesel engine for the following operating point:

Engine speed=2000rpm
Engine load=374ft-lb
EGR valve position =7% open (first control input)
VGT position=70% close (second control input)

Superposed sinusoid signals are applied to perturb the EGR valve and VGT vane position. The magnitudes of the excitation signals are chosen as ±10% around the set point values for the EGR and VGT valve positions. The system model is identified as 5th order multivariable linear system in the following discrete-time form:

$$x_{k+1} = Ax_k + Bu_k, \qquad u = [u_1 \ \ u_2]' \qquad (3)$$
$$y_k = Cx_k, \qquad y = [y_1 \ \ y_2]'$$

where the EGR and VGT positions are the inputs and the EGR rate and the boost pressure are the outputs:

$$u_1 = u_{EGR},\ u_2 = u_{VGT},\ y_1 = EGR\% \ ,\ y_2 = P_{im} \qquad (4)$$

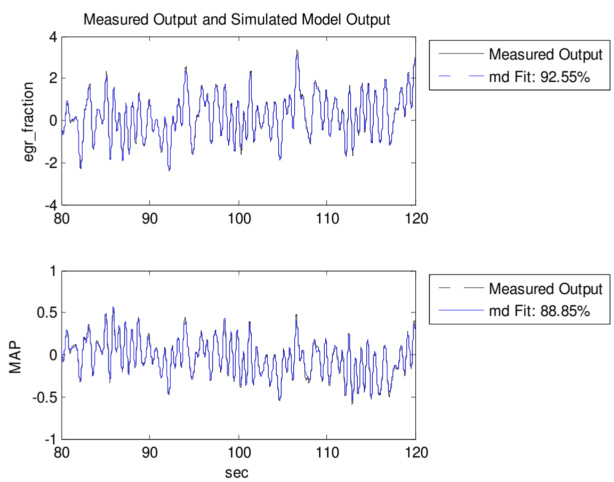

Figure 3: Model validation for the identified air handling system

This identified model is then validated and its time responses against the excitation signals are shown in Fig.3. In Fig.3, the solid black curves represent the perturbed outputs of the boost pressure and EGR rate whereas the blue or dashed curves represent the corresponding model outputs of Eqn. (3). The model is around 90% accurate.

The pole-zero locations of this linear model are also shown in Fig.4 in the continuous frequency domain. Define the transfer functions G_{11}, G_{12}, G_{21}, and G_{22} as individual elements for this system:

$$y_1 = G_{11}u_1, \quad y_1 = G_{12}u_2, \quad y_2 = G_{21}u_1, \quad y_2 = G_{22}u_2 \qquad (5)$$

Since the transfer functions share the same poles, their transmission zeros determine the differences among these subsystems. In each sub-plot of Fig.4, three zeros are almost canceled by three poles at those locations. Therefore, the remaining fourth zero indicates how strong a control input will have influence on the outputs. For example, the fourth zero of G_{12} is located at –23 on the real axis, which is the furthest away from the imaginary axis relative to the other zeros in the other subsystems G_{11}, G_{21}, and G_{22}. This means that the VGT input, not only affects the boost pressure, but also affects the EGR rate substantially at this operating point. By comparing the distances of the fourth zeros in G_{21} and G_{22} to the imaginary axis, we see that the EGR valve input has less coupling effect on the boost pressure.

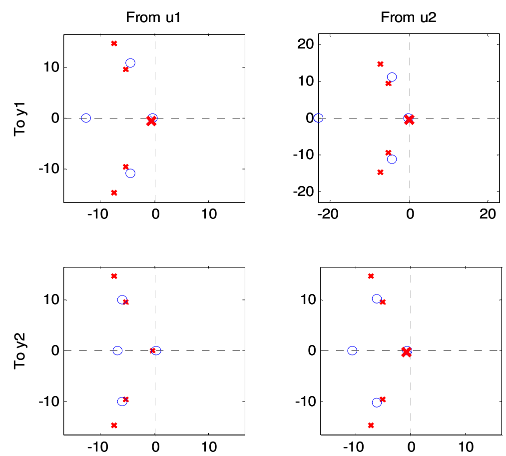

Figure 4: Pole-zero locations for the identified model

This means that VGT is the main leverage to modulate both boost pressure and EGR rate. Further, the system (3) is a non-minimum phase system, where there is a zero on the right half s-plane for G_{12} as shown in Fig.4.

IV. Decoupling Between EGR Rate and Boost Pressure

In this section, an approach is proposed to decouple the EGR rate from the boost pressure control for this multivariable system. Define a compensated plant as

$$P(s) = G(s)*W(s) \qquad (6)$$

where G(s) is the original open-loop system or plant, W(s) is designed such that P(s) is diagonally dominant. After nearly the 'cancellation' of the three poles and zeros, the system G(s) of Eqn(3) can be approximated by a second order system as follows:

$$G(s) \approx \frac{1}{d(s)}\begin{bmatrix}(s+a_{11}) & (s+a_{12})\\(s+a_{21}) & (s+a_{22})\end{bmatrix}, \qquad (7)$$

where *d(s)* is a second order polynomial. Define a decoupling weighting matrix W(s) to be

$$W(s) = \frac{1}{(s+a_0)^2}\begin{bmatrix}(s+a_{22}) & -(s+a_{12})\\-(s+a_{21}) & (s+a_{11})\end{bmatrix}, \qquad (8)$$

where a_0 determines a roll-off frequency for W(s), such that W(s) is stable and strictly proper. Multiplying G(s) by W(s) yields a diagonally dominant plant:

$$P(s) \approx G(s)W(s) = \frac{1}{d(s)(s+a_0)^2}\begin{bmatrix} N(s) & 0 \\ 0 & N(s) \end{bmatrix}, \qquad (9)$$
$$N(s) = (s+a_{11})(s+a_{22}) - (s+a_{12})(s+a_{21}).$$

For low frequency decoupling, the weighting matrix *W(s)* can be simplified as a constant matrix, which is the inverse of the DC gain of G(s), i.e.

$$W = G^{-1}(s)|_{s=0} * diag(\alpha_1, \alpha_2) \qquad (10)$$

where α_1 and α_2 are additional scaling parameters. After compensated by *W*, the plant *P(s)* becomes *P(s)=G(s)*W*. The frequency responses of *P(s)* are illustrated in Fig.5.

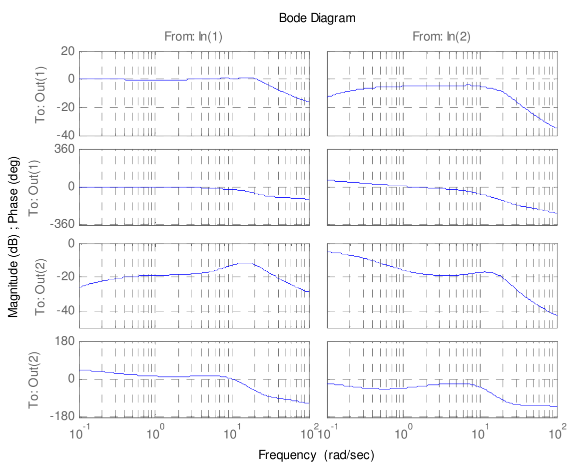

Figure 5: Bode diagram of the decoupled system at low frequencies, where In(1)=u_1, In(2)=u_2, Out(1)=y_1, Out(2)=y_2

It can be seen from Fig.5 that the VGT input now mainly affects the boost pressure with much less effect on the EGR rate than before. Similarly, the EGR valve input has dominant effect on the EGR rate, but much less effect on the boost pressure. The compensated system becomes diagonally dominant at the low frequency ranges of interest.

V. Quantitative Feedback Design for EGR/VGT Control

Based on the compensated system, *P(s) = G(s)*W*, a diagonal controller is then structured in the form of Fig.6, which renders the dynamics of the closed-loop control system as follows:

$$G_{CL}(s) = (I + P(s)C(s))^{-1}P(s)C(s) \qquad (11)$$

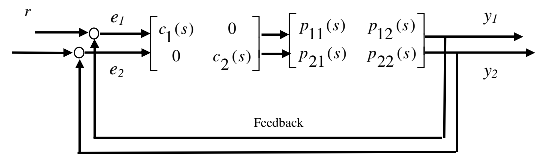

Figure 6: Compensated MIMO system stabilized by diagonal controllers

The design of feedback control is then performed by a MIMO loop shaping in terms of the QFT design framework [1]. Specifically, denote the loop transmissions of this MIMO system for the first and second control loops or channels to be $L_1(s)$, $L_2(s)$:

$$L_1(s) = p_{11}c_1 - p_{12}c_2(1+p_{22}c_2)^{-1}p_{21}c_1$$
$$L_2(s) = p_{22}c_2 - p_{12}c_2(1+p_{11}c_1)^{-1}p_{21}c_1 \qquad (12)$$

The MIMO loop shaping is performed through the following steps [1]:

First step design: Assuming that $c_2(s)$ is a high gain controller, such that

$$L_1(s) \approx (p_{11} - p_{12}p_{22}^{-1}p_{21})c_1 = \tilde{p}_{11}c_1 \qquad (13)$$

$c_1(s)$ is designed such that it stabilizes the simplified sensitivity function of the first control channel denoted by s_{11}, while satisfying the following loop-shaping criterion:

$$|s_{11}| = \left|\frac{1}{1+\tilde{p}_{11}c_1}\right| \le \tilde{m}_1(\omega) \le m_1(\omega), \quad \omega \in [\omega_L \ \omega_H]. \qquad (14)$$

for the given sensitivity bounds $\tilde{m}_1(\omega)$ and $m_1(\omega)$. The frequency-based bound $\tilde{m}_1(\omega) \le m_1(\omega)$ is used for overdesign as a part of a sequential design since $L_1(s)$ is approximated at this step.

Second step design: For a given $c_1(s)$ from the first step design, the sensitivity function of the second control channel becomes

$$s_{22} = \frac{1}{1+L_2(s)}$$
$$= \frac{1}{1+(p_{22} - p_{12}(1+p_{11}c_1)^{-1}p_{21}c_1)c_2} = \frac{1}{1+\tilde{p}_{22}c_2}, \qquad (15)$$

$c_2(s)$ is designed for $\tilde{p}_{22}$, which stabilizes the sensitivity function s_{22}, while the following inequalities are also satisfied:

$$|s_{22}| = |\frac{1}{1+L_2(s)}| \le m_2(\omega), \quad (16)$$

$$|s_{11}| = |\frac{1}{1+L_1(s)}|$$

$$= |\frac{1}{1+(p_{11}-p_{12}c_2(1+p_{22}c_2)^{-1}p_{21})c_1}| \le m_1(\omega),$$
(17)

where the bounds for the sensitivity functions are selected to achieve desired tracking and disturbance attenuation for the specified plant. As is well known, gain and phase margins are the effective measures of system robustness. Phase margin is also related to overshoot and system response time. Let *PM* and *GM* denote the desired phase and gain margins, respectively. It was shown in [10] that by setting

$$m(\omega) = 1/\sin(PM) \quad (18)$$

the sensitivity function is bounded by

$$|\frac{1}{1+kp_0(j\omega)c(\omega)}| \le \frac{1}{\sin(PM)}, \ for\ all\ k \in [k_1,\ k_2]\ and\ \omega \ge 0 \quad (19)$$

so that the closed-loop system has guaranteed lower and upper gain margins of

$$k_2[1-\sin(PM)]^{-1} \text{ and } [1+\sin(PM)]/k_1. \quad (20)$$

Similar conclusion holds true for a complementary sensitivity function, such that

$$|\frac{p_0(j\omega)c(\omega)}{1+kp_0(j\omega)c(\omega)}| \le \frac{1}{\sin(PM)}, \ for\ all\ k \in [k_1,\ k_2]\ and\ \omega \ge 0 \quad (21)$$

For a MIMO control system, PM_1 and PM_2 are defined as the phase margins for the loop transmissions $L_1(s)$ and $L_2(s)$ in the first and the second control channels, respectively. Similarly, the gain margins are defined as GM_1 and GM_2. Based on the relations of Eqns. (18)~(21), the QFD bounds are determined in terms of the desired loop transmission gain and phase margins to shape the sensitivity transfer functions s_{11} and s_{22}.

For simplicity, we will implement the controllers of $c_1(s)$ and $c_2(s)$ in the form of a PI type control with an additional low pass filtering as follows:

$$c_i(s) = (K_{pi} + \frac{K_{Ii}}{s})(\frac{w_{ci}}{s+w_{ci}})|_{i=1,2} \quad (22)$$

Literature [1] developed a method to derive QFD bounds for the proportional and integral gains K_p, K_i, and for the cut-off frequency w_c from the QFD bounds of the sensitivity functions s_{11} and s_{22}. A family of controller $c_1(s)$ and $c_2(s)$ were then computed, such that the inequalities of Eqns. (16)-(17) are satisfied. Among these controllers, one can choose an optimal solution to further balance the trade-off between boost pressure and EGR rate responses.

VI. Controller Design and Simulations

The design of the EGR/VGT control system is validated on a diesel engine model for a variety of operating points defined in Table 1 below:

Operating points	Speed (rpm)	Load (Nm)	VGT (% close)	EGR valve (% open)
1	2000	374	70%	7%
2	2000	200	75%	9%
3	1500	374	75%	3%
4	1500	200	70%	7%
5	1000	200	75%	9%

Table 1: Validation points

The design of MIMO controllers for each operating point is based on an identified linear model represented by Eqn.(3). For the first operating point in Table 1, the design requirements are specified as 8db gain margin, 45^0 phase margin for the EGR rate control loop; and 18db gain margin, 60^0 phase margin for the boost pressure control loop.

Two different designs are compared in this example. In the first design, one combines the decoupling weighting matrix in Eqn.(10) together with PI controllers, whereas the second only designs a single PI control for each control loop. The designed controllers are then connected to the nonlinear system of Eqn.(2) to form closed-loop systems, respectively. The step responses of the closed-loop systems are compared in Fig.7. Starting at zero second, the EGR rate has a step response. At the 10^{th} second, the boost pressure has a step response. The red curves in Fig.7 represent the responses of the system that is not compensated by the decoupling matrix, while the blue curves represent the responses of the system compensated by the decoupling matrix. It can be seen for both cases that a change of EGR rate has minimum impact on the boost pressure. The boost pressure responses to step changes are almost similar. However, for the non-decoupled system, the change of the boost pressure causes a significant overshoot in the EGR rate with a maximum magnitude of 2.5, which will generate more soot during engine acceleration. In comparison, the change of the boost pressure in the decoupled system has minimum impact on the EGR rate, where the caused overshoot has merely a magnitude of 1.3.

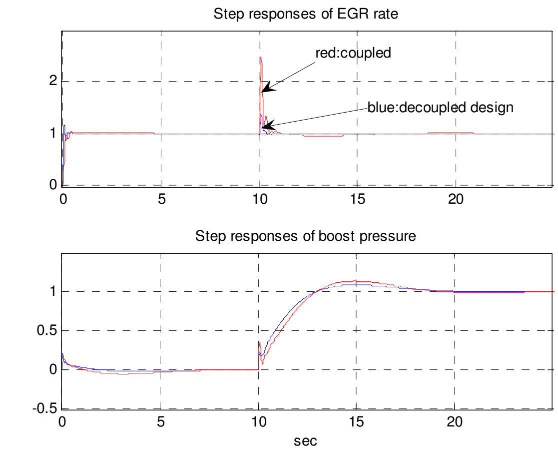

Figure 7: Comparison between decoupled and non-decoupled system responses

Now let us design controllers for the second operating point at 2000rpm and 200Nm. The system at this operating point is highly nonlinear. To control such a system, one needs to minimize the loop transmission gains at high frequencies, and increase the gains at low frequencies. The controllers designed by the QFD method are connected to the system of (2), and simulated in Fig.8.

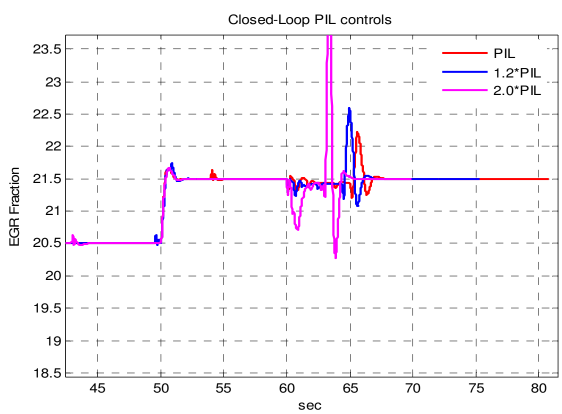

Figure 8: Control of highly nonlinear operating point for a diesel engine

At this operating point, the EGR rate responds fast to a step change at the 50^{th} second, and this change has almost no impact on the boost pressure. A step change is commanded for the boost pressure at the 60^{th} second. However, the response of the boost pressure has been delayed. After we increase the controller gain in the boost pressure loop by 20% or 100%, the boost pressure responses faster to the step change, but that increases the overshoot of the EGR rate. A derivative term in the boost control loop or model-based feed-forward control could help improve this nonlinear control performance.

In the third example, we will demonstrate the fine-tuning of a controller gain based on an initial QFD design. Consider the operating condition 3, 10db gain margin and 45^0 phase margin are specified for the EGR loop; 18db gain margin and 60^0 phase margin are specified for the boost pressure loop.

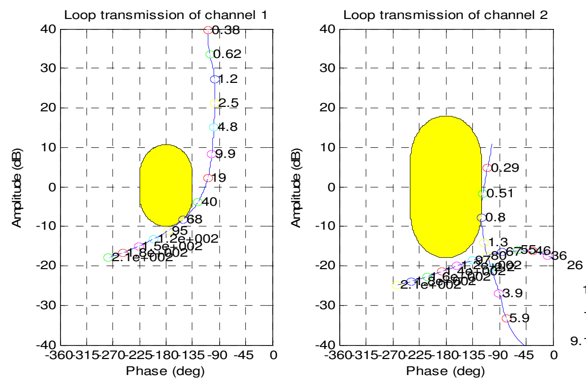

Figure 9: The Nichols charts of the loop transmissions $L_1(s)$ and $L_2(s)$

The Nichols charts of the loop transmissions $L_1(s)$ and $L_2(s)$ with an initial design for the specified QFD bounds are plotted in Fig.9. It is seen from this figure that the designed system meets the gain and phase margin requirements. A boost pressure response to a step change is plotted in Fig.10, see the blue curve, where its response is a little slow. To increase the bandwidth, one can fine-tune the existing controller without repeating the full design. For example, by doubling the integration gain of the boost control channel, the time response of the boost pressure becomes faster as shown in Fig.10.

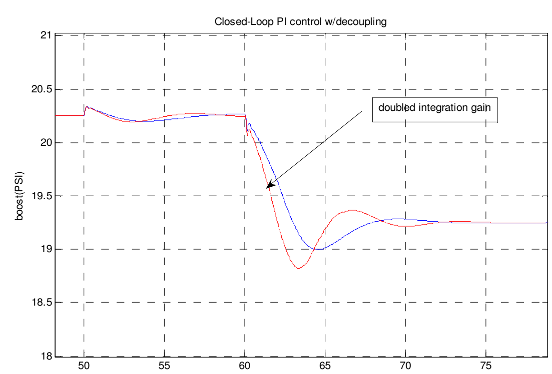

Figure 10: Fine tuning of the boost pressure control response

The corresponding Nichols chart after this gain change is plotted in Fig.11. The boost control channel meets the same gain margin, but the phase margin is reduced to 45^0, which still preserves good robustness margin. For the EGR control channel, the phase margin remains unchanged; the gain margin is reduced only by 1dB.

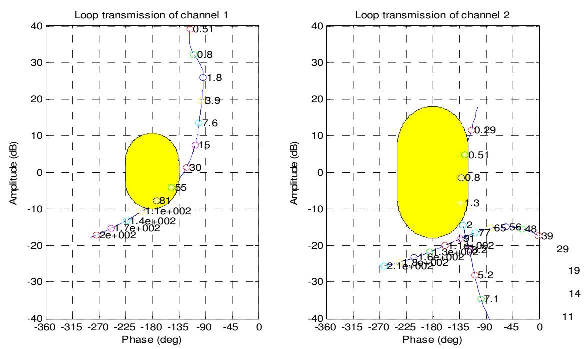

Figure 11: Nichols chart of the modified design

The design of the controllers for the 4th and 5th operating points are similar, Fig.12 illustrate the design results meeting the specifications for the 4th operating conditions. The EGR rate has a step response at 90 second; the fluctuation of the EGR rate at 100th second is caused by a step change in the boost pressure.

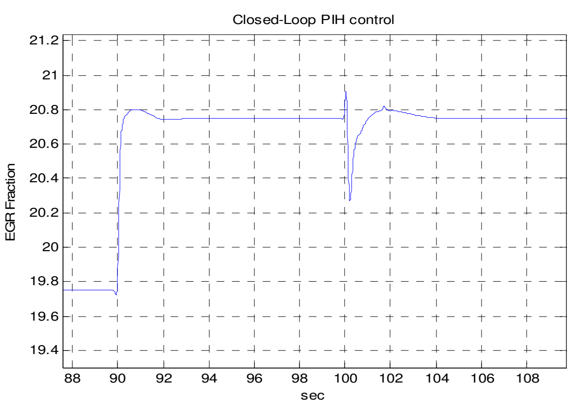

Figure 12: EGR response at operating condition 4

VII. Conclusions

In this paper, a model-based approach using the QFD methodology is proposed for the design of a multivariable control system for the air charging system of a diesel engine. Utilizing the information derived from a plant model, the approach is able to design a robust controller to achieve better trade-off for various control performances. Design examples are presented and demonstrated the merits of this approach. On-going investigation has shown promising robustness as plant parameter changes and against system time delays. Transient performances will be further investigated for this robust control design.

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