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Nonlinear Model Predictive Control of Dual Loop - Exhaust Gas Recirculation in a Turbocharged Spark Ignited engine

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Abstract—**This paper proposes a Nonlinear Model Predictive Control (NMPC) strategy for Turbocharged SI engines with a Dual Loop Exhaust Gas Recirculation system (DL-EGR). The control problem has been formulated as a reference tracking problem wherein the intake manifold pressure and recirculated cooled exhaust gas fraction are arbitrary references. The desired control actuator positions are derived by solving the NMPC. Simulation results demonstrate that the proposed NMPC optimally regulates the dual EGR circuits depending upon the requested reference trajectories and engine operating conditions resulting in improved tracking of references compared to the NMPC controlling only single loop EGR architectures.**

I. Introduction

The external Exhaust Gas Recirculation (EGR) technology is widely used on Compression Ignition (CI) engines for its ability to reduce NOx emission [1]. This technology was introduced to Spark Ignition (SI) engines recently due to its ability to mitigate knock and reduce pumping loss [2]. However, SI engines require much more precise EGR concentration control than the CI engines. The employment of High Pressure (HP) and Low Pressure (LP) dual loop (DL) EGR provides more degrees-of-freedom for EGR concentration control. The LP-EGR loop has a characteristic transport delay much longer than the HP-EGR loop, which introduces additional difficulty in air fraction control in the intake manifold. Conversely, exhaust enthalpy exchange through the turbine is reduced in the case of HP-EGR thereby reducing turbine power and hence adversely affecting compressor mass flow rate. It is desirable to coordinate the dual loop EGR systems to achieve given EGR concentration target and maximize turbocharger efficiency. In this work, a Nonlinear MPC (NMPC) is proposed to control the intake manifold pressure (MAP) and EGR concentration of a turbocharged SI engine with DL-EGR, by manipulating the throttle, EGR valves of HP and LP loops. For a given preview horizon of tacking references, the NMPC minimizes tracking error and control effort subject to the actuator limit and states constraints.

The control of turbocharged gasoline engine air handling system with EGR is a Multi-Input Multi-Output (MIMO) problem with nonlinear and coupled dynamics. NMPC has been utilized in [8], to track torque for a turbocharged SI engine where the throttle and wastegate are considered as the control inputs. External EGR control has not been included in the aforementioned research. While references for the DL-EGR control of SI engine are limited, the control of the diesel engine air system with dual loop EGR has been extensively discussed in literature [1] and [5], where the authors successfully implemented a switched linear MPC with feed-forward and feedback controllers for the multi-input multi-output nonlinear DL EGR and VGT system. In these papers, multiple linear models were developed over engine speed and fueling operating regions and a switched linear MPC approach with quadratic cost function was employed to calculate control actions. In [1], the MPC controller was designed to control charge pressure and total EGR mass flow whereas in [5], MPC mainly uses the VGT and Exhaust Throttle (ET) actuators to track the charge and LP EGR flows, respectively.

In [6] and [7], the authors have implemented an MPC for diesel engine air path control using a mean value model. Piecewise linear models are used to approximate the nonlinear air path dynamics. At every time step a linear MPC problem is solved to find the optimal EGR valve and Variable Geometry Turbocharger positions that tracks the mass air flow and the absolute manifold pressure references.

The EGR control strategy developed for CI engines may not be suitable for SI engines. The LP EGR transport dynamics considered in previous CI engine research are highly simplified and usually similar to a first order delay. This approximation may be tolerable for CI engine operation as the ignition process is more robust against EGR dilution. Such an assumption for a SI engine can potentially lead to undesirable operation like misfire and partial combustion due to over dilution of EGR. Hence, it is necessary to improve the accuracy of transport delay dynamics in the control oriented model of the airpath as shown in [3], wherein a NMPC has been utilized with a segmented boost manifold based transport delay model integrated into the airpath model. This research employs a similar approach in which a high order transport delay model is utilized to improve the prediction of LP-EGR fraction from the location of its delivery to the engine cylinders.

The paper is organized as follows. A control oriented model is derived for the turbocharged engine airpath with DL-EGR, followed by formulation of the NMPC control problem, simulation results, performance quantification and concluding remarks.

II. Control oriented air path model

The individual components of the air path have been modeled separately and ultimately integrated to form a control oriented air path as depicted in Figure 1.

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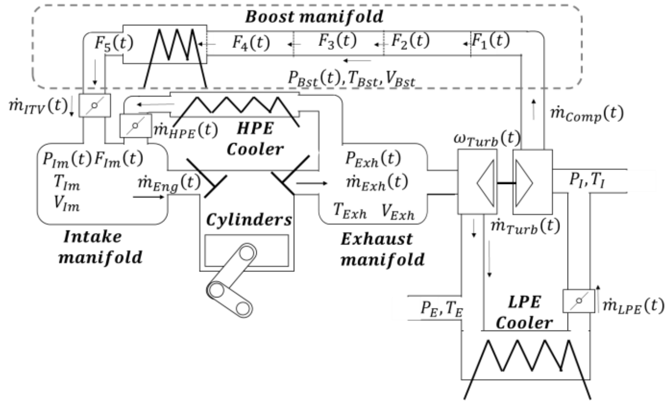

Figure 1. Schematic of Engine and Air path model.

Here, $P, T, F, \dot{m}, \omega$, and V are pressure, temperature, mass fraction, mass flow rate, rotational speed and volume respectively. Subscripts $Exh, Bst, Im, HPE, LPE, ITV, Comp, Turb, I$ and E are the exhaust manifold, boost manifold (between compressor and throttle), intake manifold, high-pressure EGR, low-pressure EGR, intake throttle, compressor, turbine, ambient compressor inlet and post-turbine exhaust respectively. The following section is dedicated to description of the component level models used in the air path model.

A. Compressor mass flow rate model

The compressor model chosen in this work is proposed by Hadef et al. [9] which is an evolution of the Jensen and Kristensen empirical mean value model [10]. The latter has been widely used in publications relating to modeling, control and estimation of air path on turbocharged engines.

Figure 2. shows the correlation between the modeled iso-speed curves and the manufacturer's data.

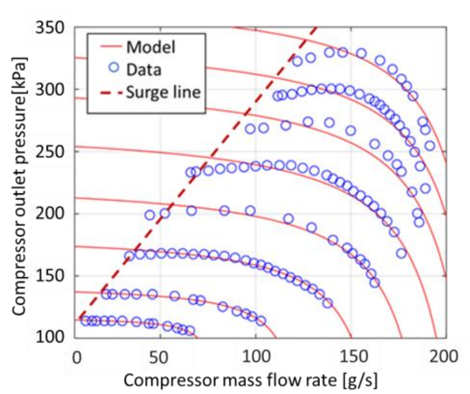

Figure 2. Compressor map data and modeled iso-speed curves of pressure ratio and mass flow rate.

B. Turbine inlet pressure model

Control oriented turbine mass flow rate models described in [11] is a good candidate model for this system. This approach typically requires the P_{Exh} to be modeled as a state. However, the volume of the exhaust manifold along with the high gas temperature causes the dynamics of P_{Exh} to be significantly faster than the control system sample time. Hence, a simple polynomial function was utilized to predict turbine P_{Exh} as shown in Figure 3. It is noteworthy that the polynomial coefficients are constants for a fixed turbine inlet temperature and outlet pressure. The variation in outlet pressure of the turbine is negligible compared to the variation in the inlet pressure of the turbine and it is reasonable to assume ambient pressure at the outlet of the turbine. The inlet temperature is assumed constant at 900K which is a representative nominal value.

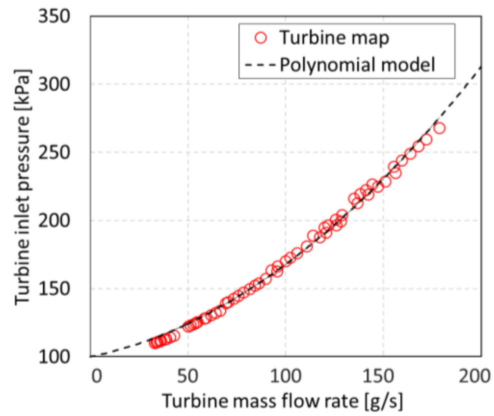

Figure 3. Turbine map data with second order polynomial approximation at reference inlet temperature and outlet pressure.

C. Gas transport dynamics in boost manifold

The boost manifold is typically the largest control volume in a turbocharged engine air path and hence consideration of the gas transport dynamics in the boost manifold is critical for transient control of EGR dilution at the cylinders. This manifold has been traditionally considered as a lumped volume in the turbocharged diesel engine air path [10], [12]. However, in typical turbocharged automotive engines the boost manifold is characterized more by thin long pipes rather than plenums. Therefore, a lumped volume based assumption for the boost manifold is done at the expense of accuracy. Experimental testing has confirmed the above mentioned phenomenon as shown in Figure 4 where a step input in EGR valve position results in a dead time delayed and mixing influenced response from the EGR measurement sensor installed at the throttle valve. The mixing influence cannot be attributed purely to mixing in the manifold but also to the filling and emptying dynamics of the sensor itself.

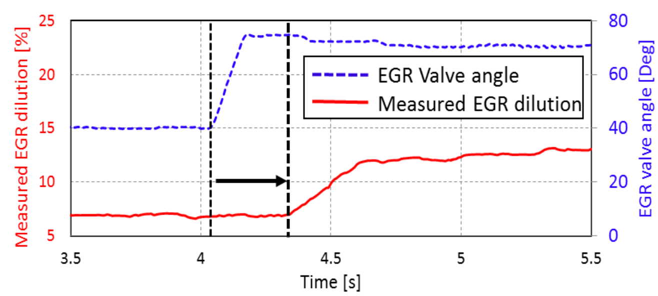

Figure 4. Step response of oxygen sensor based EGR measurement installed at compressor outlet location

The consequence of excessive EGR dilution in diesel engines typically results in increased smoke and soot emission. However, with excessive EGR dilution, gasoline spark ignited engines can suffer from very high cyclic variability in torque production. Furthermore, the three way catalyst can encounter premature damage due to misfires caused by excessive dilution. The gas transport model chosen here [13] is based on conservation of mass and the ideal gas law in a control volume.

$$\dot{F}_i = \frac{RT_i}{P_i V_i}\left(F_{In}\dot{m}_{In} - F_i \dot{m}_{Out}\right) \tag{1}$$

where, F_i is the EGR fraction in the i^{th} control volume V_i. The number of control volumes utilized to discretize the boost manifold is the same as the number of fraction states F_i used in the transport model. Hence, the order of the transport model is equal to the number of sections used to discretize the control volume.

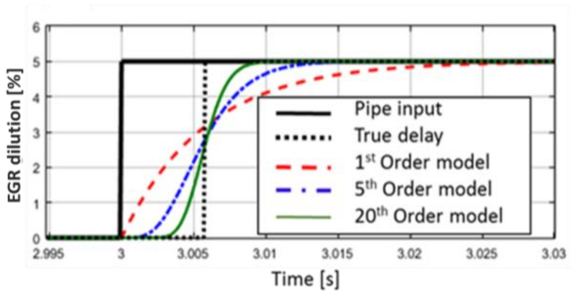

Figure 5. Comparison of step response of transport delay models with true dead time delay response

Higher order models show increasing accuracy and similarity to the 'true' dead time delayed response (if pure pipe plug flow is considered) as shown in . A 5th order transport delay model was chosen for this research as it has significantly higher accuracy than the 1st order model which is a single lumped volume for the entire boost manifold. Moreover, higher order models would have diminishing improvement in accuracy over the computational burden of additional state dynamics.

III. Control Problem Formulation

A. Non-linear state space model of airpath

Using the component level models described in the previous section, iso-thermal pressure dynamics and turbocharger rotor dynamics described in [14] the discrete time state-space model is derived as given below. Since the goal of this research was to evaluate control of mass flow rates of the EGR valves, temperature dynamics were ignored to keep system dimensionality low and allow for more transport delay states.

$$x(k+1) = f(x(k), u(k)) \tag{2}$$

$$y = Cx = [P_{Im}\ F_{Im}]^T \tag{3}$$

$$x = [P_{Im}\ P_{Bst}\ \omega_{Turbo}\ F_1\ F_2\ F_3\ F_4 F_5\ F_{Im}\ \dot{m}_{LPE}]^T \tag{4}$$

$$u = [\dot{m}_{ITV}\ \dot{m}_{HPE}\ \dot{m}_{LPE}]^T \tag{5}$$

The output vector y consists of the intake manifold pressure P_{Im} and intake manifold EGR% F_{Im}. For gasoline SI engines the intake manifold pressure and EGR% strongly associated with the engine load and these references are assumed to be available from an external source (eg. Torque management system, supervisory control). The remaining states P_{Bst}, ω_{Turbo}, F_i and $\dot{m}_{LPE}$ are the compressor outlet pressure, turbocharger rotational speed, the transport delay states and the LP-EGR mass flow rate respectively. The input vector u consists of the intake throttle and high pressure mass flow rate $\dot{m}_{ITV}$ and $\dot{m}_{HPE}$. The LP-EGR mass flow rate is controlled using input $\dot{m}_{LPE}$, as this allows tuning of the rate of change of the LP-EGR mass flow rate. The model is also normalized to mitigate potential numerical issues during execution.

B. Model Predictive Control problem formulation

The objective of the control algorithm is to minimize the squared error between a reference value $y_{Ref}(k)$ and the system output $y(k)$ for the preview horizon. Hence, the optimal control problem has been designed to be an output tracking controller with the cost function defined as follows.

$$J\big(x(k), U(k)\big) = \bar{y}(k+N)^T H \bar{y}(k+N) + \sum_{i=k}^{k+N-1} \bar{y}(i)^T Q \bar{y}(i) + u(i)^T R u(i) \tag{6}$$

where,
$\bar{y}(k) = y(k) - y_{Ref}(k)$, $Q \in \mathbb{R}^{2\times2}_{>0}$, $H \in \mathbb{R}^{2\times2}_{>0}$, $R \in \mathbb{R}^{3\times3}_{>0}$
$U(k) = [u(k), u(k+1), \ldots u(k+N-1)]^T$

Q and H are the tuning matrices for the penalty on output tracking error within the prediction horizon and the terminal output error. The tuning of these matrices was done to place more emphasis on the tracking of F_{Im} as the consequence of poor tracking of this parameter can result in misfires. R is the tuning matrix for penalty on magnitude of input u. The non-linear optimization problem is hence formulated as follows.

$$\min_{U(k)} J\big(x(k), U(k)\big)$$

Subject to: (7)
$\dot{m}_{ITV}(i) \geq 0$,
$\dot{m}_{HPE}(i) \geq 0$,
$P_{Bst}(i) - P_{Im}(i) \geq 0$,
$\omega_{Turbo}(i) \geq 0$.

where, $i = k, k+1.., k+N-1$

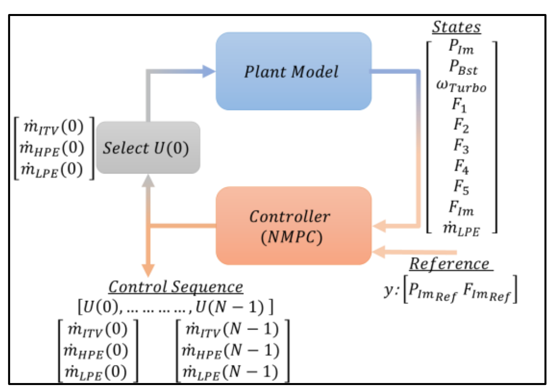

Figure 6. Schematic of closed loop NMPC control structure

All of the constraints are imposed to maintain physical feasibility in the system. The turbocharger rotational speed has the longest time constant in the system (1.1 seconds) Hence, for a sample time of 0.1 seconds the horizon length N is 11. The numerical values chosen in the tuning matrices are chosen heuristically with more emphasis on the EGR% control. Fig. 6 shows the overall schematic representation of Simulation results and discussion the NMPC control structure. The NMPC problem was simulated using ACADO toolkit [15].

The simulation was performed to study the ability of NMPC to handle the following aspects of the system:

- Reference tracking of P_{Im} and F_{Im}
- LP-EGR transport delay in the boost manifold
- Interaction between EGR loops and turbocharger
- Control of split ratio between HP-EGR and LP-EGR

The intake manifold pressure reference trajectory was deliberately chosen to have boosted and un-boosted values so that the interaction between the EGR circuits and the turbocharger performance could be studied. Arbitrary time varying references were chosen for EGR%. The simulation was performed for three engine speeds, 1000, 2000 and 3000 RPM with the same reference trajectories. The speeds were chosen because external cooled EGR has maximum benefit in knock mitigation at low speed high load conditions. In the following section simulation trajectories are shown for 2000 RPM.

C. High-pressure EGR control only

It is evident from Figure 7a. and 7b. that in this case the system is able to track the reference of the intake manifold mass better than the low pressure only and dual loop cases. However, the intake manifold pressure reference tracking under a reference value above atmospheric pressure is the poorest in this case as seen between 5 and 7 seconds. This is because of the reduction of mass flow rate through the turbine caused by the high-pressure EGR system. The HP-EGR circuit effectively behaves like a waste-gate and hence results in reduced turbine power. As per the perception of the driver, this would seem as poor engine torque response. The controller overshoots the intake throttle and HP-EGR mass flow rates whenever there is a sudden change in reference. Another thing to note is that the boost pressure always stays above or equal to the intake manifold pressure implying that $\dot{m}_{ITV}$ is physically feasible.

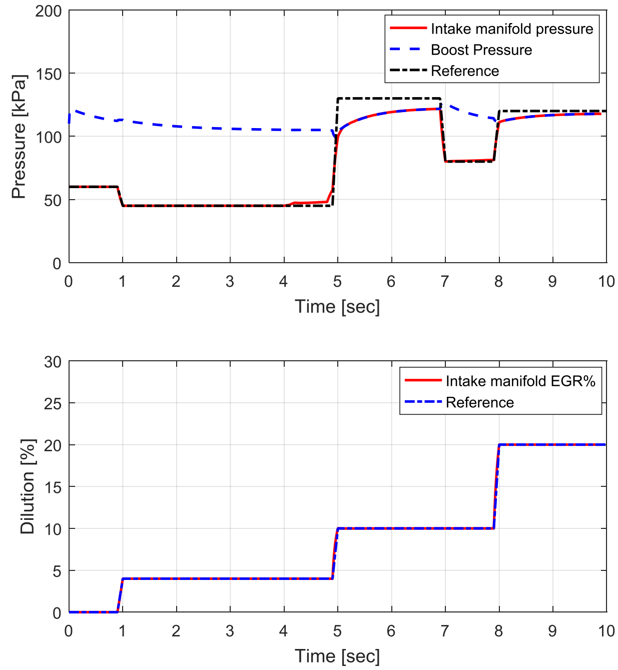

Figure 7a. Pressure states and intake manifold EGR dilution trajectories for HP-EGR control only.

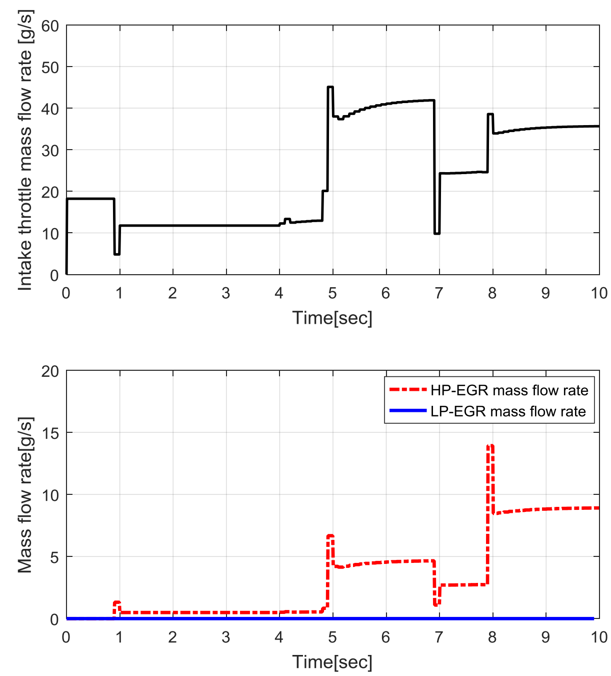

Figure 7b. Input trajectories for HP-EGR control only.

D. Low-Pressure EGR control only

The most significant difference between this case and the HP-EGR case is that the intake manifold pressure reference tracking is significantly better as seen in Fig. 8a. and 8b. as all of the exhaust mass flow passes through the turbine resulting in higher turbine power to drive the compressor to higher boost levels. However, the reference tracking for EGR mass in the intake manifold is worse with oscillation at the EGR% reference changes at 5 and 8 seconds. Due to the transport delay model being incorporated into the NMPC, there is also some lead action visible at 0.5 seconds on the LP-EGR mass flow rate to meet the EGR% reference change at 1 second. The LP-EGR mass flow rate has an immediate effect on only the EGR% in the first section of the boost manifold which is far upstream of the intake manifold. The dilution dynamics in every consequent air-path section diminishes the effect of this input on the intake manifold EGR%. Hence, the tuning weight in matrix R associated with the rate of change of LP-EGR mass flow rate had to be modified to reduce oscillations in the LP-EGR mass flow rate.

E. Dual-loop EGR control

In this case the intake manifold pressure and EGR mass reference tracking appear to be reasonably better compared to the LP-EGR only case as shown in Figure 9a. and 9b. Since the engine is operated throttled up to 5 seconds the NMPC

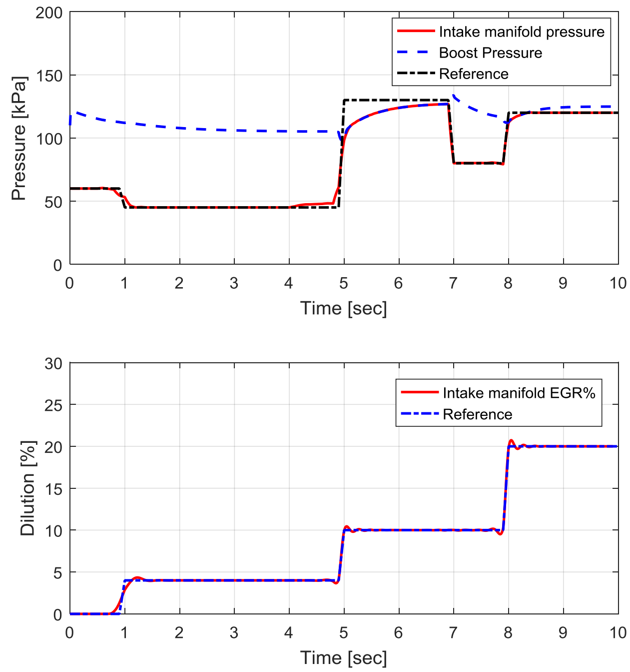

Figure 8a. Pressure states and intake manifold EGR dilution trajectories for LP-EGR control only.

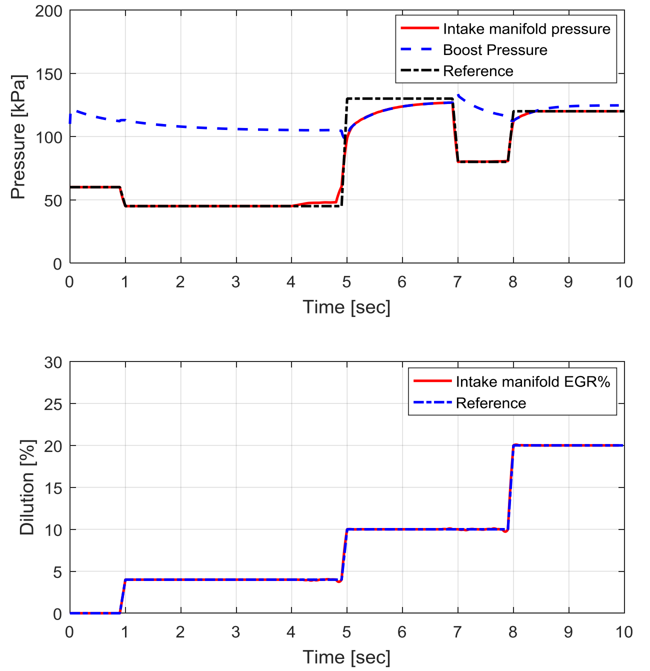

Figure 9a. Pressure states and intake manifold EGR dilution trajectories for Dual-loop EGR control.

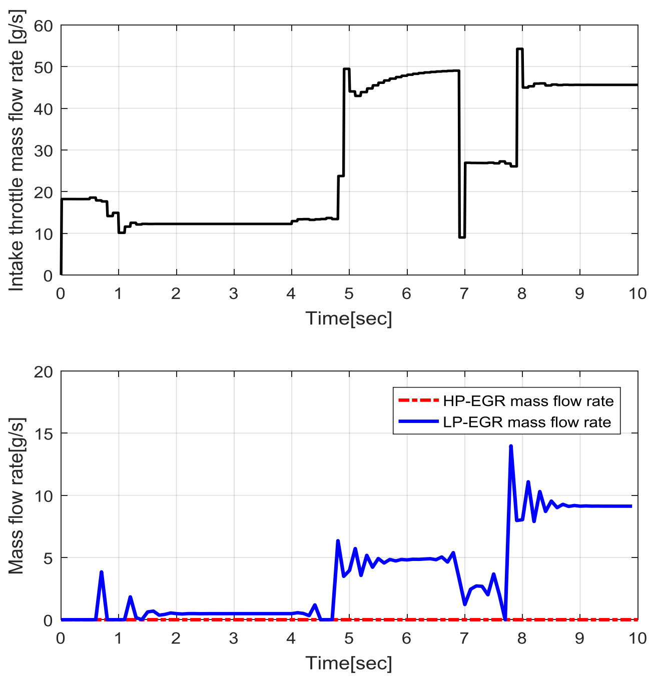

Figure 8b. Input trajectories for LP-EGR control only.

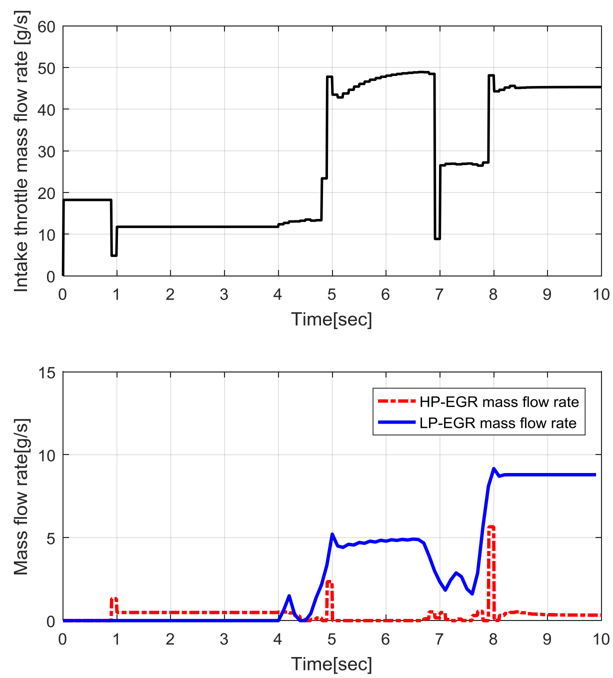

Figure 9b. Input trajectories for Dual-loop EGR control.

commands only HP-EGR up to 4 seconds. At the transition from throttled to boosted operation LP-EGR flow is initiated to maximize turbine power. As the intake manifold pressure level increases due to turbo spool-up, a sharp HP-EGR mass flow rate spike is commanded at 5 and 8 seconds to increase intake manifold pressure further while maintaining the desired EGR% reference. The penalty on rate of change of LP-EGR mass flow rate was also relaxed compared to the LP-EGR only case. It is noteworthy that the commanded LP-EGR mass flow rate shows lesser oscillations despite the relaxed penalty indicating that NMPC effectively uses HP-EGR to 'supplement' the LP-EGR during boosted operation.

F. Additional comparison between different cases

The intake manifold EGR% for different cases at 2000 RPM is shown in Fig. 10. Even though the HP-EGR case settles to the reference at 5 seconds, there is an error associated with the first order filling dynamics of the intake manifold visible between 4.9 and 5 seconds. This error is diminished for the LP-EGR and DL-EGR case as the primary EGR actuator in these cases is the LP-EGR mass flow rate which is farther downstream from the intake manifold.

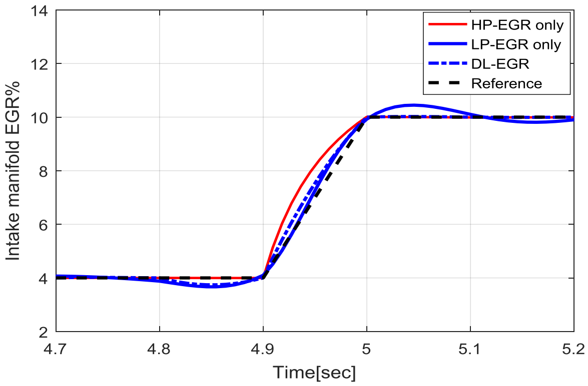

Figure 10. Transient intake manifold EGR dilution for multiple EGR architectures

TABLE 1. PERFORMANCE SUMMARY OF EGR % CONTROL USING NMPC

Engine speed [RPM]	EGR architecture	RMSE [EGR%]	Max Error [EGR%]
1000	HP-EGR	0.105	1.06
	LP-EGR	0.13	0.66
	DL-EGR	0.0489	0.6
2000	HP-EGR	0.13	1.53
	LP-EGR	0.17	1.3
	DL-EGR	0.07	0.061
3000	HP-EGR	0.16	2.07
	LP-EGR	0.1	0.7
	DL-EGR	0.2	2.34

The performance of the NMPC for dual and single loop EGR architectures has been summarized in TABLE 1. At 1000 and 2000 RPM the DL-EGR shows reduced RMSE and Max errors compared to the single loop architectures. However, at 3000 RPM the DL-EGR showed poorer performance than the single loop systems. This can be attributed to non-ideal tuning for the output tracking and input penalties

IV. CONCLUSION AND FUTURE WORK

The proposed dual-loop EGR NMPC strategy is capable of coordinating the HP and LP EGR mass flow rate to track the intake manifold pressure and intake manifold EGR% references. Inclusion of turbocharger dynamics in the model facilitates optimal balancing between the EGR loops whilst minimizing turbocharger lag. The NMPC guarantees constraints and hence physically feasible values for the control inputs. It also demonstrates the ability to consider the transport delay dynamics in the LP-EGR loop due to the utilization of a multi-segment transport delay model. For the lower engine speeds, DL-EGR control shows superior EGR% tracking performance. Inclusion of compressor and turbine bypass valve, engine cylinder models in addition to further refinement of the tuning parameters in the NMPC will be considered as future work.

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