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A COORDINATED BOOST CONTROL IN A TWINCHARGED SPARK IGNITION ENGINE WITH HIGH EXTERNAL DILUTION

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ABSTRACT

This paper proposes a novel master-slave control strategy for coordination of throttle, wastegate and supercharger actuators in an electrically twincharged engine in order to guarantee efficient boost control during transients, while at steady state a throttle-wastegate coordination provides minimum engine backpressure hence engine efficiency elevation. The benefits and challenges associated with Low Pressure Exhaust Gas Recirculation (LP-EGR) in a baseline turbocharged engine, including improved engine efficiency, mainly due to better combustion phasing, and sluggish engine response to a torque demand due to slowed down air path dynamics were studied and quantified in [1]. Hence in this paper an electrical Eaton TVS roots type supercharger at high pressure side of the turbocharger compressor (TC compressor) is added to the baseline turbocharged engine and the performance of the proposed controller in the presence of LP-EGR, which is a more demanding condition, is evaluated and compared to the turbocharged engine. One dimensional (1D) crankangle resolved engine simulations show that the proposed master-slave control strategy can effectively improve the transient response of the twincharged engine, making it comparable to naturally aspirated engines, while the consumed electrical energy during transients can be recovered from the decreased fuel consumption due to LP-EGR conditions at steady state in approximately 1 second. Finally, a simple controller is developed to bypass the TC compressor and maximize the engine feeding charge during the transients in order to avoid TC compressor choking and achieve faster response.

INTRODUCTION

Nowadays turbocharging is the most prevalent mean of boosting the engine feeding charge. Unfortunately turbocharged engines can suffer from drivability issues such as insufficient boost pressure at low engine speeds and slower torque response known as ”turbo-lag”. Supercharging is the next popular method of boosting the engines, which does not possess the drivabil-

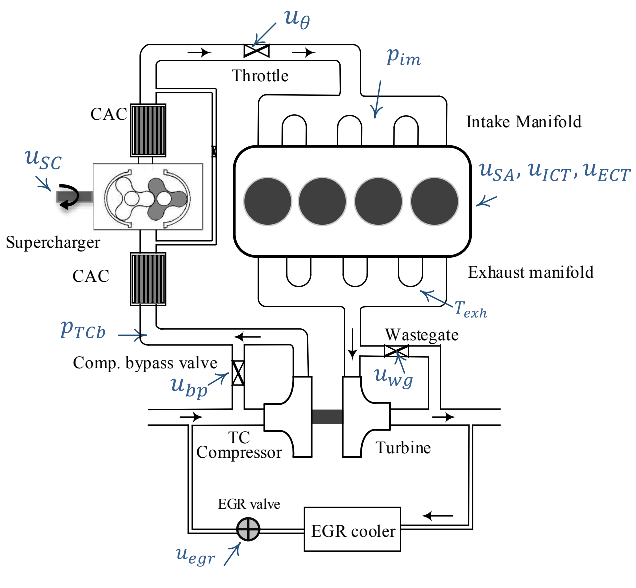

FIGURE 1. Schematic of twincharged SIDI engine

ity challenges associated with turbocharging but at the expense of lower engine fuel economy. Including a supercharger in a turbocharged powertrain or twincharging can combine the drivability pros of supercharging with fuel economy benefits of turbocharging [2–4].

Variable speed supercharging provides the possibilities of more flexible boost control, using a smaller supercharger and reduced supercharger bypass losses, resulting in both engine fuel economy and peak torque improvement [5]. Variable speed supercharging can be achieved easily through electrical supercharging. Although several attempts have been reported to use electrical roots type superchargers to retrieve otherwise wasted throttling losses [6, 7], recovering the fuel penalty associate with supercharger consumed electricity is not easy. Considering this fact, electrical supercharging seems to be a good option for transient performance improvement and for enabling high fuel economy concepts such as highly diluted combustion.

During recent years cooled external Exhaust Gas Recirculation (eEGR) has gained a lot of attraction as one potential practice to decrease vehicles fuel consumption. Boosted gasoline engines fuel economy improves significantly by including cooled eEGR, [8–11]. Presented at [1], despite considerable improvement in engine fuel consumption by introducing LP-EGR, mainly due to closer to Maximum Brake Torque (MBT) spark timing and less heat transfer and pumping losses, LP-EGR slows down the air path dynamics of the turbocharged systems significantly because of increased air path volumes and decreased exhaust gas specific enthalpy. As a direct result of this effect the engine response time to a torque demand deteriorates, which means higher drivability challenges for turbocharged engines equipped with eEGR.

Considering this fact and the ability of variable speed supercharging in turbo-lag reduction, this paper proposes incorporating an electrical supercharger in a turbocharged engine with low pressure loop eEGR. The application of a supercharger in addition to throttle and wastegate actuators provides an additional degree of freedom in air path control of the engine, hence the coordination of these three actuators would be very important for engine fuel consumption reduction in addition to faster charge control. The focus of current study would be the design of a controller for coordination of throttle, wastegate and supercharger speed, which minimizes the engine backpressure at steady state for fuel economy maximization and uses the supercharger only during the transient through a novel master-slave control strategy, where the supercharger speed is slaved to both throttle and wastegate.

This paper is organized as follows. First the twincharged engine structure and its model are described. The controller structure and different parameters actuation method is explained next. The subsequent section compares the transient results including the response time and the engine fuel economy of the twincharged engine to the baseline turbocharged engine with two different control strategies. One without eEGR and fast torque response throttle-wastegate actuation and the second case with eEGR and throttle-wastegate coordination for high fuel economy. Different actuators movement and the supercharger performance for the studied transients are illustrated next. In order to avoid choking in the turbocharger compressor during sever tip-ins a simple controller is designed to bypass the compressor using a bypass valve and help speeding up the engine torque response even more. The engine transient performance with and without this controller is compared at the final section of this paper.

SYSTEM AND MODEL DESCRIPTION

The studied baseline engine is a 1.6 liter 4 cylinder four-stroke turbocharged spark ignited gasoline direct injection engine. In the proposed twincharged configuration, an electrical Eaton TVS roots type supercharger at high pressure side of the turbocharger compressor (TC compressor) along with a charge air cooler (CAC) and a bypass line are included. Figure 1 shows the schematic of the twincharged engine and its air path. Spark timing (u_{SA}), intake and exhaust cam timing (u_{ICT} and u_{ECT}), throttle (u_θ), wastegate (u_{wg}), EGR valve (u_{egr}), TC compressor bypass valve (u_{bp}) and supercharger speed (u_{SC}) are various actuators and their input signals, represented on the figure.

The employed GT-power model, [12], captures 1-D gas dynamics in the air path, turbocharger performance, heat transfer, valve lift and port flow behavior, fuel injection and vaporization, combustion and other details necessary to predict engine performance. The model has been calibrated and validated against the baseline turbocharged engine data.

CONTROL SYSTEM

Figure 2 shows the schematic of the proposed controller for twincharged engine. The aim of this controller is to use the electrical supercharger only during the transient part of the response and minimize the engine backpressure at steady state while having a desirable response time and overshoot. Desired intake manifold pressure (p^*_{im}) is determined for the desired BMEP ($BMEP^*$), desired eEGR level ($eEGR^*$) and engine speed (N) and the minimum necessary turbocharger boost pressure, p^*_{TCb}, that ensures least engine backpressure is determined as following:

$$p^*_{TCb} = \begin{cases} p^*_{im} & if\ p^*_{im} \geq p_{ambient}, \\ p_{ambient} & if\ \ p^*_{im} < p_{ambient} \end{cases} \qquad (1)$$

where $p_{ambient}$ is the ambient pressure. In the designed controller configuration, the throttle, wastegate and supercharger speed are coordinated such that the supercharger speed is controlled based on the error in tracking the desired intake manifold and desired boost pressure hence slaved to both throttle and wastegate. This will provide fast and effective boost response during the transients and idling at steady state. Assuming a perfect SC motor control, the supercharger speed is commanded through two proportional controllers and a feedforward part. One of the controllers output is proportional to the error in intake manifold pressure, p_{im}, and the second one is proportional to the error in turbocharger boost pressure, p_{TCb}:

$$u_{SC} = min(\tilde{u}_{SC}, u_{SC}^{max}) \tag{2}$$

$$\tilde{u}_{SC} = k_1 e_{im} + k_2 e_{TCb} + u_{SC}^{idle} \tag{3}$$

$$e_{im} = p_{im}^{*} - p_{im} \tag{4}$$

$$e_{TCb} = p_{TCb}^{*} - p_{TCb} \tag{5}$$

where u_{SC} is the supercharger speed, k_1 and k_2 are proportional controller gains. The variable u_{SC}^{idle} is the idle speed of supercharger in which the pressure ratio across the supercharger is equal to unity and is in form of a look up table calibrated for different engine speed and loads.

When a step is applied in BMEP during a tip-in transient, desired intake manifold pressure, p_{im}^{*} and desired turbocharger boost, p_{TCb}^{*} change accordingly. The supercharger speed rises and its pressure ratio and pumping flow increase consequently, responding fast to the errors e_{im} and e_{TCb} that gets generated till the throttle and wastegate controllers function to reduce these errors. Then the intake manifold pressure reaches its desired value ($e_{im} \rightarrow 0$) and the supercharger speed decreases proportionally but it takes more time for the turbocharger to produce the required boost. Eventually when the turbocharger boost reaches its target steady state value ($e_{TCb} \rightarrow 0$) the supercharger speed approaches its new idle value ($u_{SC} \rightarrow u_{SC}^{idle}$), at which it does not produce any boost. This controller is gain scheduled for different operating points.

The throttle valve (u_θ) controls intake manifold pressure and consists of a model based feedforward part and a PI feedback part,

$$u_\theta = k_{p,\theta}(p_{im}^{*} - p_{im}) + k_{i,\theta}\int_{t_0}^{t}(p_{im}^{*} - p_{im})dt + u_{\theta}^{ff} \tag{6}$$

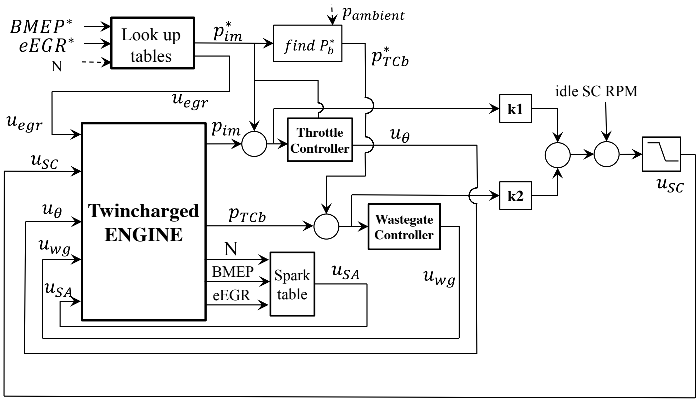

FIGURE 2. Control system schematic

where in above equation u_{θ}^{ff} is the feedforward portion and is calculated based on desired intake manifold pressure and parameters such as engine speed, engine size, throttle size and intake manifold volume. $k_{p,\theta}$ and $k_{i,\theta}$ are the proportional and integral feedback gains respectively. The wastegate (u_{wg}) controls the turbocharger boost pressure and is a PI controller as following:

$$u_{wg} = k_{p,wg}(p_{TCb}^{*} - p_{TCb}) + k_{i,wg}\int_{t_0}^{t}(p_{TCb}^{*} - p_{TCb})dt \tag{7}$$

where $k_{p,wg}$ is the proportional gain and $k_{i,wg}$ is the integral gain and both are gain scheduled for different operating points. The master-slave structure of the controller is more clear considering that the throttle and wastegate controllers both possess integral action, hence they will meet the steady state set points of intake manifold and TC boost pressure, while the supercharger controller includes only proportional parts for the errors in these two parameters. Besides, targeting minimum required TC boost pressure, wastegate controller guarantees minimizing the engine back pressure and helps improving the fuel economy.

Cam timings are calibrated for different engine speeds and loads. EGR valve position is specified based on engine speed and desired values of load and eEGR level. The spark timing is determined using a look up table which accounts for engine load, speed and instantaneous residuals in intake manifold. The residual fraction in intake manifold is measured using a fast O2 sensor. The knock model used to generate this look up table is presented in [13].

In modern turbocharged engines smaller turbochargers are used in order to decrease the engine turbo-lag, however this practice imposes some limitations such as higher choking probability. In the twincharged engine during tip-ins the supercharger aims to fill the intake manifold as fast as possible. While, the TC compressor is located at upstream of the supercharger and in extreme transient conditions it chokes and acts as a restriction, limiting the pumping ability of the supercharger. To avoid this problem, a simple controller is designed to bypass TC compressor when vacuum is sensed in its down stream using a bypass line and a valve. It can be stated that for a portion of time the bypass valve and the turbocharger work in parallel to maximize the air flow into the engine. The bypass valve control law is as following:

$$u_{bp} = \begin{cases} 1 \; if \; pr_{TCC} < 0.98 \\ 0 \quad otherwise \end{cases} \tag{8}$$

where pr_{TCC} is the cycle averaged pressure ratio across the TC compressor, 1 means fully open and 0 means fully closed valve. The bypass valve dynamic behavior is here approximated by a first order system with 40 ms time constant.

TRANSIENT RESULTS

In this paper three different engine configurations were simulated and compared against each other:

- Twincharged engine with LP-EGR, denoted by "TwC 15% eEGR" on result plots. The control system configuration for this case was explained in the prior paragraphs and through Eqn. (1) to (8).
- Baseline turbocharged engine with LP-EGR system, marked with "TC 15% eEGR" on result plots. In this case the desired boost pressure is determined using Eqn. (1) and throttle and wastegate controllers actuate based on Eqn. (6) and Eqn. (7) respectively, so in this case minimum engine backpressure is achieved too. Spark timing and cam timings are controlled similar to twincharged case.
- Baseline turbocharged engine without LP-EGR, represented by "TC 0% eEGR" on plots. In this system throttle actuator controls intake manifold pressure, with a controller similar to prior two configurations but wastegate is kept closed during the simulations ($u_{wg} = 0$) to provide higher boost pressure hence minimizing the engine response time to a torque demand. In this case spark timing and cam timings are also controlled similar to previous configurations.

Engine Parameters

Figure 3 compares the twincharged engine parameters (black lines) for 10% to 90% of full load tip-in at 2000 rpm engine speed to the baseline turbocharged engine parameters in two cases: high fuel economy case, with 15% eEGR and minimum backpressure wastegate control strategy (dash-dot red lines) and fast torque response case with closed wastegate and without eEGR (dashed blue lines). The load responses (the first plot on left) show that the new proposed powertrain is significantly faster in terms of torque response. The response time (10-90%) for the twincharged system is 0.28 sec, which is as fast as a naturally aspirated engine. Comparing this number to that of turbocharged engine without eEGR 1.28 sec (dashed blue line), and with eEGR 2.30 sec (dash-dot red line), it is evident that the new powertrain configuration can completely eliminate the turbo-lag in the boosted engine.

The corresponding plot in Fig.4 shows similar results for 10% to 60% of full load tip-in at 2000 rpm constant engine speed. The response time (10-90%) is 0.2 sec for twincharged engine, 0.50 sec for turbocharged engine and 0.70 sec for turbocharged engine with 15% eEGR, confirming the torque response improvement of twincharging even for medium loads.

The first plots on right of Fig.3-4 represent the intake manifold pressure versus time. As expected the intake manifold pressure increases much faster for the twincharged cases and its steady state value is higher for the cases with eEGR. The second plots on the left demonstrate the variation in Brake Specific Fuel Consumption (ΔBSFC) of three cases. The numbers are changes relative to BSFC at 10% load for twincharged engine. The turbocharged engine with 15% eEGR and the twincharged engine have lower fuel consumption at steady state.

The supercharger energy consumption is also included in the presented plots and this is why the specific fuel consumption of twincharged cases are slightly higher than the turbocharged cases during the transient part of the response. Although some of the electric energy that powers the SC can come from regenerative braking, it is assumed here that all the electricity required to power the supercharger comes from the engine crank train. The calculated supercharger equivalent fuel consumption and the twincharged engine corrected BSFC depend on the supercharger consumed power (P_{SC}) and an assumed gross efficiency (η_e) of 70% for electrical energy production, storage and conversion,

$$P_{SC,eq} = P_{SC}/\eta_e \tag{9}$$

where $P_{SC,eq}$ is the supercharger equivalent crank train power and P_{SC} includes the required power to compress the flow and the friction losses. Equivalent fuel flow rate of the supercharger ($\dot{m}_{f,SC}$) can be computed assuming the same brake efficiency for power generation as the twincharged engine brake efficiency, η_b without considering an alternate source such as energy stored in the battery during regenerative braking,

$$\dot{m}_{f,SC} = P_{SC,eq}/\eta_b \tag{10}$$

$$\eta_b = \frac{P_b}{\dot{m}_f Q_{LHV}} \tag{11}$$

where P_b is the engine brake power, $\dot{m}_f$ is the fuel flow and Q_{LHV} is the lower heating value of the fuel. And finally the twincharged engine corrected specific fuel consumption is computed as the ratio of total fuel flow rate to the engine brake power, P_b

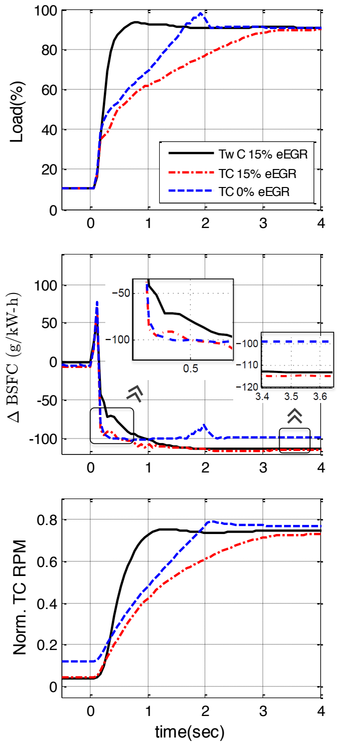

FIGURE 3. Engine parameters for 10% to 90% of full load tip-in

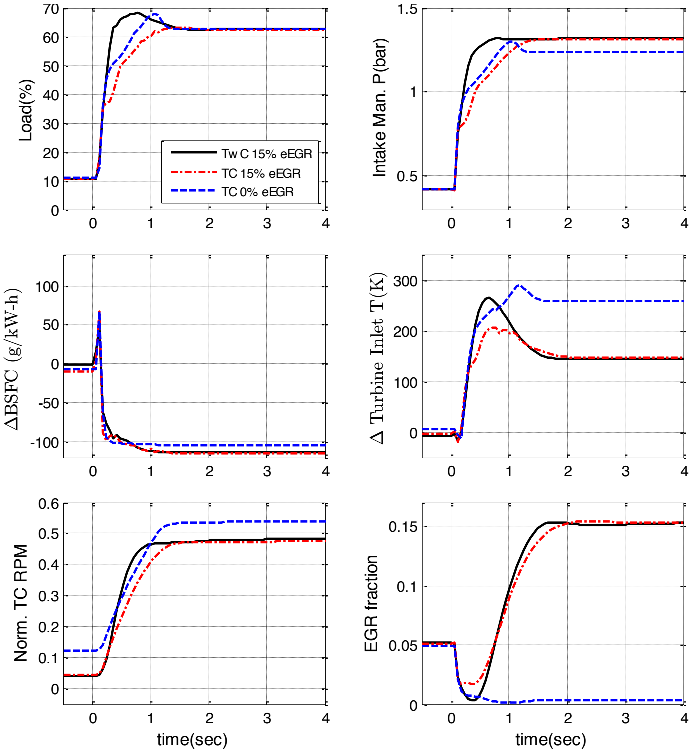

FIGURE 4. Engine parameters for 10% to 60% of full load tip-in

$$BSFC_{cor} = (\dot{m}_f + \dot{m}_{f,SC})/P_b \tag{12}$$

The main assumption in these calculations is the assumed brake efficiency for supercharger equivalent crank train power generation. This assumption is somewhat conservative since the engine brake efficiency is very low during the transient response, while in a drive cycle there is a flexibility in electrical energy production and it can be generated when the engine brake efficiency is around its optimum value.

Although the twincharged engine consumes more energy during the transient response compared to the baseline engine without eEGR, it has a better fuel economy at steady state mainly due to included low pressure EGR [1]. So it would be interesting to know how much operation time at steady state is required to compensate for the electrical energy consumed by the supercharger during the transient. The computations show that for 10% to 90% of full load tip-in case only 1.1 sec operation at high load is sufficient to make up for the fuel consumed by the supercharger during the studied transient. This number for 10% to 60% of full load case is 0.8 sec, meaning that if the twincharged engine operates only 0.8 sec at 60% of full load, it recovers the extra consumed energy for speeding up the response. In addition, in case of free available electrical energy e.g. from brake recovery in mild hybrid vehicles, no extra fuel is burnt for the supercharger consumed electricity and in this condition depending on the batteries state of charge (SOC) it could be efficient to use the supercharger at steady state too.

The second plot on the right compares the turbine inlet temperature variation during the tip-in for different models. The numbers are changes relative to turbine inlet temperature at 10% load in twincharged engine. For 90% of full load, the exhaust temperature decreases by 140 °C and for 60% of full load case it drops by 110 °C through adding 15% eEGR and minimum back-pressure wastegate control. The third plots on the left illustrate the normalized turbocharger speed (as the ratio of turbocharger speed to it maximum allowable speed) versus time for three cases. For the baseline turbocharged engine with 0% eEGR, the turbocharger speed is higher before and after the transient since the wastegate is closed in this case and more flow passes through the turbine. The turbocharger speeds up faster in twincharged case compared to turbocharged case with 15% eEGR. The reason is that the target load is achieved faster in twincharged engine and the exhaust gas enthalpy follows the engine power, so the available energy of turbocharger increases much faster for the twincharged engine.

Finally the third plots on the right represent the residuals in the intake manifold. For 10% of load (before the step) although EGR valve is fully closed at this point (see Fig.5 and Fig.6), the

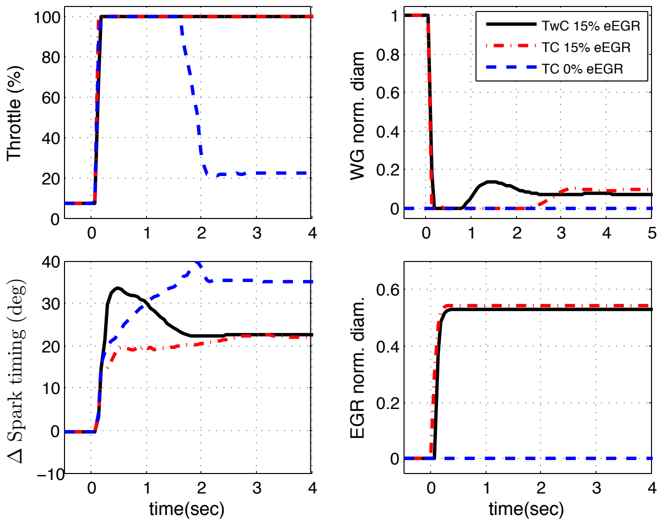

FIGURE 5. Actuators movement for 10% to 90% of full load tip-in

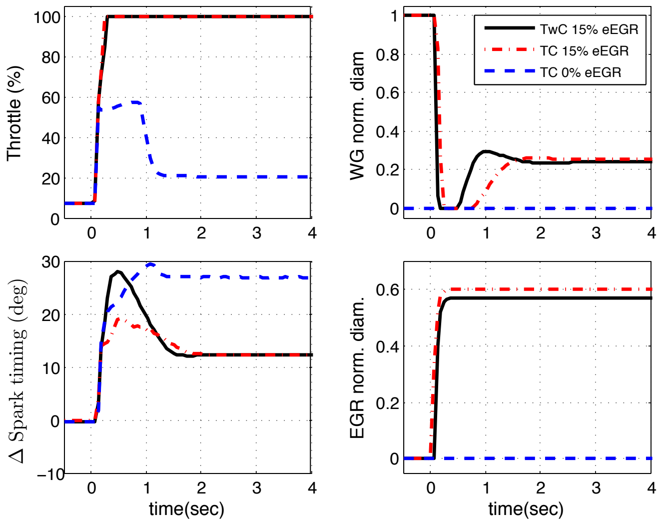

FIGURE 6. Actuators movement for 10% to 60% of full load tip-in

residual fraction is around 0.05 for all cases. This part of residuals enters the intake manifold during the valves overlap from the cylinders and exhaust manifold. After the tip-in cam timings change; this part of residuals is eliminated and for the cases with eEGR it takes some time for the burnt residuals to pass the air path and fill the intake manifold. For the studied tip-ins the transients start without LP-EGR because the studied engine maintains high internal residuals at low load and it is not possible to have high LP-EGR at this condition due to both combustion stability problems and LP-EGR limitations at low loads.

In twincharged engine case this intake manifold residuals undershoot causes an undesirable overshoot in load response. Although the intake manifold pressure is regulated with less than 0.5% overshoot, the load response of the two studied cases show 3.1% and 10.6% overshoot respectively. This overshoot could be mitigated by decreasing the supercharger controller gains but this will slow down the response as well and could make the benefits of the electrical supercharging disappear. Nonlinear control design will be investigated in the future to attenuate this effect.

Actuator Movement

Figure 5 illustrates the actuators movement for 10% to 90% of full load transient and Fig.6 represents the same parameters for 10% to 60% of full load case. In cases with 15% eEGR, where minimum backpressure wastegate control strategy is employed, the throttle is fully open after the step is applied. This control approach minimizes the boost pressure and pressure drop across the throttle consequently, and the minimum of this parameter occurs at wide open position for both 90% and 60% of full load.

In the cases with minimum backpressure wastegate control strategy, the wastegate is open at 10% load, since no boost is required at this load; it closes after the step is applied to help speeding up the turbocharger and opens again to its maximum possible value later to avoid producing unnecessary boost pressure. This event happens sooner for the twincharged case because as presented before, the turbocharger speeds up more rapidly in twincharged engine. Normalized wastegate diameter is calculated as the ratio of effective wastegate diameter to its maximum value.

The second plot on left shows the variation in spark timing compared to low load value. The spark timing of cases with eEGR is advanced at high load compared to the baseline engine without eEGR. During the transient part the spark timing is more retarded for the twincharged case compared to the turbocharged case with eEGR. The reason is that as explained before the spark timing is controlled based on the engine speed, BMEP and eEGR level in intake manifold. In twincharged engine the BMEP reaches its target value faster than the other case, while the residual fraction is still low in intake manifold. So the spark is retarded to avoid knocking as the result of this specific spark control strategy. The last plots show the EGR valve normalized diameter during the studied transients governed by its actuator dynamics and a look up table value. Specifically the valve is closed at low load and opens to its desired value after the tip-in is applied for the cases with eEGR.

Supercharger Performance

Figure 7 shows the supercharger performance parameters for the two presented transient results (10% to 90% of full load and 10% to 60% of full load transients). The first plot shows the supercharger speed. The supercharger rpm increases during the transient period and approaches its idle value during the steady state. The second set of plots represents the operating line of the

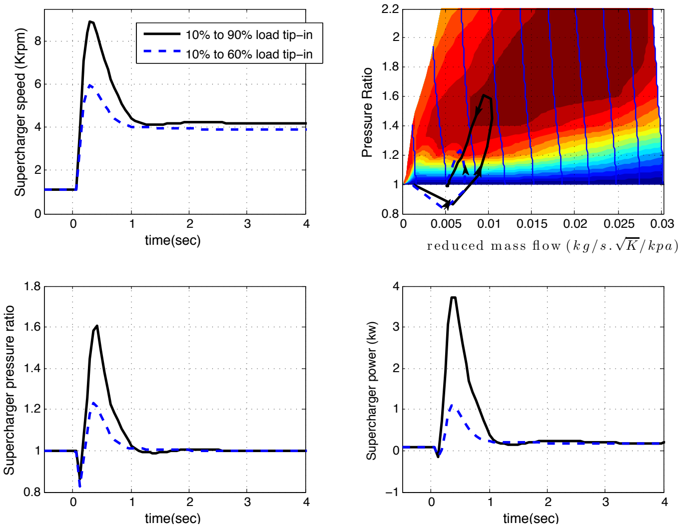

FIGURE 7. Supercharger parameters

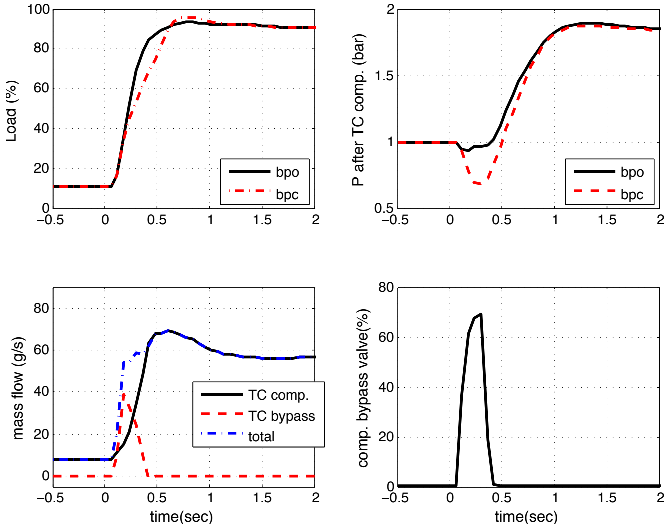

FIGURE 8. Comparing cases with and without using TC compressor bypass valve

supercharger on its characteristic map and the third set of plots displays the pressure ratio across the supercharger. The supercharger is used just in transient portion of the results and its pressure ratio is unity before and after the transient part. The pressure ratio across the supercharger drops to less than unity at the beginning of the transient. The reason for this effect is that after applying the step in desired load the throttle valve which is a fast actuator fully opens but due to its inertia, supercharger response to the command is slower and it cannot speed up and provide the required feed charge immediately, as the result its downstream volume pressure drops to sub-atmospheric. This effect is similar to pressure ratio drop across the TC compressor at the start of transient explained before.

The last plot represent the supercharger power consumption. For 10% to 90% of full load tip-in, the supercharger consumes up to 3.7 kW and for 10% to 60% of full load tip-in its maximum power is 1.1 kW. The supercharger consumed power before and after the transient part is very small (100 to 200 W) and includes friction losses.

TC Compressor Bypassing

It was mentioned before that a simple controller is employed in order to bypass the TC compressor when it is acting as a restriction for the supercharger in severe transients. Figure 8 compares the load response of the twincharged engine with and without this mechanism. The results clearly show that the case without using the bypass valve (dash-dot red line -bpc) has a slower load response. The response time (10-90%) of this case is 0.14 sec larger than the case which uses the bypass valve, equal to 0.42 sec. The top right plot represents the pressure in the pipe between the TC compressor and the supercharger. In the case that the bypass valve is not used and stays always closed, this pressure drops to 0.69 bar while in the other case it declines only to 0.93 bar. The bottom left plot shows the charge flow through the turbocharger compressor (black line), the flow through the bypass valve (dashed red line) and the total feeding flow (dash-dot blue line). The last plot represents the bypass valve opening profile and shows that the valve opens only for the first half second of the transient response.

CONCLUSION

This paper proposed a novel control configuration for coordination of throttle, wastegate and supercharger speed in an electrically twincharged engine in order to achieve high fuel economy as well as efficient boost control by using the supercharger only during the transient part of the response and enabling high eEGR and minimum backpressure wastegate control at steady state. The performance of the twincharged engine with the introduced controller at 15% low pressure eEGR was compared to the baseline engine for two different transient cases. For the large and the medium tip-in cases the twincharged engine response time decreases to 0.28 sec and 0.20 sec, compared to 2.30 sec and 0.70 sec for turbocharged engine with same eEGR level and wastegate controller respectively. Although the twincharged engine has lower fuel economy during the transient, 1-D simulations show that for 90% of full load case 1.1 sec and for 60% of full load case only 0.8 sec operation at steady state is adequate to compensate for the consumed electricity because of decreased fuel consumption mainly due to included eEGR and more advanced spark timing shown in [1]. In addition, the simulations show choking in the TC compressor during the large tip-in can increase the response time of the twincharged engine up to 140 ms, which was avoided in current study through bypassing TC compressor when the pressure ratio across it dropped to less than a marginal number.

REFERENCES

[1] Nazari, S., Stefanopoulou, A., and Martz, J. "Minimum backpressure wastegate control for a boosted gasoline engine with low pressure external egr". In ASME 2016 Dynamic Systems and Control Conference.

[2] George, S., Morris, G., Dixon, J., Pearce, D., and Heslop, G., 2004. Optimal boost control for an electrical supercharging application. Tech. rep., SAE Technical Paper.

[3] Stan, C., and Taeubert, S., 2009. Analysis of engine performances improvement by down sizing in relationship with super-and turbocharging, adapted scavenging and direct injection. Tech. rep., SAE Technical Paper.

[4] Amann, M., and Ouwenga, D., 2014. Engine parameter optimization for improved engine and drive cycle efficiency for boosted, gdi engines with different boosting system architecture. Tech. rep., SAE Technical Paper.

[5] McBroom, S., Smithson, R. A., Urista, R., and Chadwell, C., 2012. Effects of variable speed supercharging using a continuously variable planetary on fuel economy and low speed torque. Tech. rep., SAE Technical Paper.

[6] Hu, B., Copeland, C., Lu, P., Akehurst, S., Brace, C., Turner, J., Romagnoli, A., and Martinez-Botas, R., 2015. "A new de-throttling concept in a twin-charged gasoline engine system". *SAE International Journal of Engines,* ***8***(2015-01-1258).

[7] Hu, B., Copeland, C., Brace, C., Akehurst, S., Romagnoli, A., Martinez-Botas, R., and Turner, J., 2014. A new turboexpansion concept in a twin-charged engine system. Tech. rep., SAE Technical Paper.

[8] Teodosio, L., De Bellis, V., and Bozza, F., 2015. "Fuel economy improvement and knock tendency reduction of a downsized turbocharged engine at full load operations through a low-pressure egr system". *SAE International Journal of Engines,* ***8***(2015-01-1244).

[9] Francqueville, L., and Michel, J.-B., 2014. "On the effects of egr on spark-ignited gasoline combustion at high load". *SAE International Journal of Engines,* ***7***(2014-01-2628), pp. 1808–1823.

[10] Alger, T., Gingrich, J., Roberts, C., and Mangold, B., 2011. "Cooled exhaust-gas recirculation for fuel economy and emissions improvement in gasoline engines". *International Journal of Engine Research,* ***12***(3), pp. 252–264.

[11] Siokos, K., Koli, R., Prucka, R., Schwanke, J., and Miersch, J., 2015. "Assessment of cooled low pressure egr in a turbocharged direct injection gasoline engine". *SAE International Journal of Engines,* ***8***(2015-01-1253).

[12] GT-SuiteV7.4, 2014. Gamma Technologies, Inc., Westmont, IL. `http:// www.gtisoft.com`.

[13] Middleton, R. J., Gupta, O. G. H., Chang, H.-Y., Lavoie, G., and Martz, J., 2016. Fuel efficiency estimates for future light duty vehicles, part a: Engine technology and efficiency. Tech. rep., SAE, No. 2016-01-0906.