Dynamic Pressure Ratio Allocation for Electric Supercharger and Turbocharger Coordination

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Abstract — To improve transient torque response of an aggressively downsized turbocharged spark-ignition engine, a series sequential boosting configuration that utilizes an electrically driven supercharger to assist the main turbocharger compressor is considered. The system is over-actuated with two actuators (i.e. turbocharger wastegate, electric supercharger motor) and one output, the total boost pressure. A dynamic pressure ratio allocation method is designed that apportions the control reference to each actuator. The method is based on the characteristics of each actuator, i.e., the turbocharger is generally more efficient but has a slower dynamic response relative to the electrical supercharger. Therefore, the turbocharger is always asked to deliver the desired boost pressure while the electrical supercharger is utilized only as necessary in transients to make up for any deficit. The control design is validated in a test vehicle and good performance is observed. Transitions between the electric supercharger and turbocharger are smooth and without noticeable disturbances.

1. INTRODUCTION

Downsizing and turbocharging yield considerable improvements in part-load fuel economy for gasoline engines while maintaining or exceeding the power output of larger naturally aspirated engines [1][3][4][5]. For high-end torque and power, a larger turbocharger would be desirable. Such a turbocharger however is not able to produce adequate boost at low engine speed and load conditions due to insufficient turbine energy [2] resulting in slower than desired response called turbo lag.

A variable geometry turbine (VGT) is one of the well applied hardware solutions to improve turbocharger transient dynamics and reduce turbo-lag [6][7] in diesel engines. However, higher exhaust gas temperatures have prevented such technology from being a cost effective solution for gasoline engines. Additionally, VGT technology may not be sufficient to improve the response of the aggressively downsized engine at low engine speed and load because of the lack of the exhaust enthalpy.

Series sequential dual turbocharging technology has been widely studied on gasoline and diesel engines [8]. The controller for such a system is designed to utilize its ability to achieve high boost pressures, while also enabling manageable pressure ratios across the individual turbochargers. The small turbocharger is often scheduled to operate at low airflow conditions for fast response. Meanwhile the larger turbocharger takes over at high airflow conditions.

Electrically assisted compound boosting solutions [9] are emerging synergistic with the increasing range of mild hybrid products available in the market. The concept is derived from conventional compound boosting that combines a mechanically driven supercharger with a turbocharger. The supercharger helps provide transient performance while the compressor and turbine of the turbocharger are designed to efficiently provide peak engine power. Instead of employing a mechanical connection that directly consumes instantaneous engine power, reducing engine torque during the transient, the electric supercharger is connected to an electric motor that draws power from the battery.

The electric supercharger typically has a relatively small compressor hence reduced inertia for fast transient response. Because of the limitations of flow capacity and electrical system, it is not typically used for steady state operation. Therefore, the control is different than for a conventional dual turbocharger system, where the main objective is to utilize the steady state capability of the two turbochargers to deliver the boost pressure efficiently [10][11]. Vilegas et al. [12] used a similar approach to design a controller for a compound boost system with an electric supercharger. Vilegas separates the operation of the boosting devices by using the electric supercharger below engine speeds of 3000rpm and the turbocharger otherwise. Overall system efficiency could be improved by utilizing the more efficient turbocharger at lower engine speeds.

A frequency separation technique was studied to operate and coordinate the two boosting devices simultaneously [13]. The electric supercharger was used to deliver fast transient boost pressure. Meanwhile, the turbocharger was controlled to deliver the lower frequency steady-state boost pressure. There are several challenges in calibrating this controller because of the complexity in estimating the bandwidth of the turbocharger. This approach is limited because the system only uses one pressure sensor. The measurement hence includes boost contributions from both electric supercharger and turbocharger. An improved system, as will be described later, with an added pressure sensor is studied.
Tracking a desired boost pressure, coordinating the turbocharger and the electric supercharger, is an overactuated control problem. There is one output signal to track and two independent control signals, wastegate and motor speed. Approaches for control of overactuated systems can be grouped, roughly, into centralized and modular strategies [14]. In a modular approach, chosen virtual inputs are computed by an outer loop and fed to an inner loop, a control allocator, where the virtual inputs are transformed to the actual control inputs [15]. The dimension of the virtual control input is typically chosen equal to the output dimension and constraints are handled by the allocator. The outer loop design is done for a square system with no input constraints, without much information needed about the inner loop. In a centralized strategy, one controller is designed that directly determines the control inputs.

This work takes a centralized approach where the control structure is chosen based on the characteristics of each of the two actuators. The core of the design, as will be described in detail later, is dividing a desired system pressure ratio into desired pressure ratios for each individual device. This is a dynamic allocation of the control target into two separate targets but it is different to modular strategies in that the structure is not divided into different levels.

The experimental setup is introduced in the Section 2, followed by a section on control system description. Section 4 explains the electric supercharger controller in detail, Section 5 contains vehicle results, and Section 6 concludes the paper.

2. SYSTEM DESCRIPTION

The system under consideration is a compound-boosting system where an electric supercharger is added upstream of the turbocharger compressor, as shown in Figure 1. The electric supercharger is a centrifugal compressor driven by an electric motor. A flow bypass is required for operation of the system beyond the flow capacity of the electric supercharger. The bypass valve managing this flow path is actively controlled to either the open or closed position.

In this paper, $P_0$ denotes the inlet pressure of the electric supercharger compressor; $P_1$ denotes the inlet pressure of the turbocharger compressor; and $P_2$ denotes the throttle inlet pressure. $P_0, P_1, P_2$ are all measured and inputs to the controller.

3. EXPERIMENTAL SETUP

A vehicle equipped with a turbocharged gasoline engine is modified to develop the air-path control of the compound boosting technology as described in Figure 1. The electric supercharger installed in the vehicle has an inner loop controller that delivers a desired speed command and can only be controlled in the positive direction. A conventional throttle body is used as the electric supercharger bypass valve.

For the test vehicle used in this work, a standalone power supply system is used where the batteries are charged by an external power source and are not connected to the vehicle power network. Coordination with the electrical system will be subject of future work.

![Figure 1 Air-path configuration considered for control](image)

4. CONTROL OF COMPOUND BOOSTING SYSTEM

A. Pressure Ratio Based Coordination

The variables in the control system are chosen to be expressed as pressure ratios, which are intrinsic parameters for describing the capability of the boosting device and compound system. The system pressure ratio is thus the product of the contribution of the electric supercharger and the turbocharger.

$$\frac{P_2}{P_0} = \frac{P_1}{P_0} \cdot \frac{P_2}{P_1} \quad (1)$$

The block diagram of the control system is shown in Figure 2. It consists of a wastegate controller $C_2$ and an electric supercharger controller, $C_1$.

B. Wastegate Control

The controller $C_2$ regulates wastegate position to drive the pressure ratio of the turbocharger to the steady-state desired pressure ratio $\left(\frac{P_2}{P_0}\right)$. Both the feedforward and feedback controllers for the wastegate are designed using the pressure ratio based approach to prevent the fast transient boost contribution delivered by the electric supercharger from slowing the turbocharger response. Specifically, the error value ($e_P$) signal used by the wastegate feedback controller is calculated as the difference in pressure ratios,
\[ e_\Pi = \frac{P_2^*}{P_0} - \frac{P_2}{\max\{P_0, P_1\}} \] 

(2)

In this way, the feedback portion of the wastegate controller will not respond to or, in other words compensate for, the boost contribution from the electric supercharger. With a conventional boost error calculation, an overshoot caused by the electric supercharger may lead to wastegate opening too early leading to degraded transient performance.

![Figure 2](Image)

Figure 2 The electric supercharger controller \( C_1 \) in the scheme of the pressure ratio based compound boosting system control

C. Electric Supercharger Control

The electric supercharger controller \( C_1 \), includes a dynamic allocator \( D_1 \) that apportions the desired boost pressure control reference; a state machine \( A_1 \) that manages the operation of electric supercharger system; a closed-loop controller \( L_1 \) that consists of a model based feed forward function \( F_1 \), and a feedback controller \( B_1 \) for calculating the desired electric supercharger speed command. Each element of the electric supercharger controller will be explained in detail in the rest of this section.

a. Dynamic Allocator

The control allocation method is based on the fundamental characteristics of each actuator. A turbocharger that recovers some of the wasted combustion enthalpy is expected to be more efficient compared to the electric supercharger. Therefore, the capability of the turbocharger is fully utilized, while filling in any deficit with the electric supercharger.

The boost pressure ratio deficit \( (\Pi^* \text{)} \) of the turbocharger is defined as follows:

\[ \Pi^* = \frac{P_2^*}{P_0} - \frac{P_2}{\frac{P_2}{P_1^*}} \]

(3)

The desired pressure ratio target of electric supercharger \( \frac{P_1^*}{P_0} \) is \( \Pi^* \) with a constraint on the pressure ratio of turbocharger

\[ \frac{P_1^*}{P_0} = \frac{P_2^*}{P_0} / \max\{1, \frac{P_2}{P_1^*}\} \]

(4)

The constraint is included to prevent the desired pressure ratio target for the electric supercharger from exceeding the total desired system pressure ratio when \( P_1 > P_2 \), which could happen briefly after an opening of the throttle.

The control reference for the electric supercharger, \( \frac{P_1^*}{P_0} \), see Figure 3, is dynamically scheduled based on the measured pressure ratio of the turbocharger. This algorithm allows the control reference to adjust continuously based on the actual turbocharger capability. As the turbocharger spoons up, the desired assist from the electric supercharger is gradually phasing out. Finally, the turbocharger is able to sustain the total desired boost and no assist is required. Figure 3 also shows the non-constrained boost pressure deficit from Equation (3) that briefly exceeds the total desired system pressure ratio. The total desired system pressure ratio is calculated based on driver demand as well as several hardware constraints. Therefore, the control algorithm should not allow any part of the boost pressure request to exceed this target.

![Figure 3](Image)

Figure 3 Dynamic pressure ratio allocation for electric supercharger control targeting during a vehicle acceleration event

b. Model based feed forward function

The feed forward control action \( (n_{\text{ESC,ff}}) \) of the electric supercharger is based on the steady state e-compressor map provided by the supplier. The e-compressor map is converted to look up desired compressor speed based on the inputs of corrected mass flow \( (\dot{m}_{\text{corr}}) \) and desired pressure ratio \( \frac{P_1^*}{P_0} \).
\[ n_{\text{ESC, ff}} = f_1\left(\frac{p_1^*}{p_0}, m_{\text{corr}}\right) \]  
(5)

Mass flow \(\dot{m}\) is estimated as the desired engine airflow, then corrected to the reference pressure \(p_{\text{ref}}\) and temperature \(T_{\text{ref}}\).

\[ \dot{m}_{\text{corr}} = \frac{T_{\text{ref}}}{p_{\text{ref}}} \sqrt{\frac{p_0}{p_1}} \]  
(6)

c. Closed-Loop Feedback Controller

A proportional feedback controller \((B_1)\) is implemented to account for plant model (the electric supercharger compressor map) mismatch and other system uncertainties. Steady-state boost tracking is designed to be delivered by the main turbocharger. Therefore, there is no integral action in the feedback controller. The electric supercharger speed has different bandwidth for controlling the speed in increasing or decreasing directions. Therefore, a lead compensator is added to shape the closed-loop control response.

The feedback controller in Laplace domain can be expressed as:

\[ H_1(s) = K_p + \frac{a s + 1}{\tau s + 1} \]  
(7)

Input \(u\) is the pressure ratio errors of the electric supercharger:

\[ u = \frac{p_1^*}{p_0} - \frac{p_1}{p_0} \]  
(8)

To implement in the discrete system, the following conversion for the lead compensator is derived:

\[ x_k = (1 - f)x_{k-1} + f u_k, f = \frac{T_s}{\tau + T_s} \]  
(9)

\[ y_k = (1 - r)x_k + ru_k, r = \frac{a}{\tau} \geq 1 \]  
(10)

Where \(f\) is the time constant in discrete domain, \(r\) is the lead ratio, and \(T_s\) is the execution rate.

The output of the lead compensator is clipped to specify the range of lead action on \(U\) in increasing and decreasing directions. For example, the electric supercharger installed in the test vehicle does not have the authority to control the motor speed with negative torque. Therefore, the lead control action can only decrease the speed command and will not overdrive the system.

\[ \hat{y}_k = \max(\min(y_k, u_k + \Delta_2), u_k - \Delta_1) \]  
(11)

The clipped lead actions combining with the control actions from the proportional controller is:

\[ n_{\text{ESC, fb}} = K_p u_k + \hat{y}_k \]  
(12)

\[ d. \ State \ Machine \]

In this application, the electric supercharger is a transient device and is not controlled continuously. A state machine \((A_1)\) is developed to coordinate the electric supercharger and electric supercharger bypass valve. The main objective of the state machine is to determine when to activate and deactivate the electric supercharger and electric supercharger bypass valve to meet control requirements, subject to system constraints and hardware protection. At a high level, the electric supercharger is activated and the bypass valve closed when the activation criteria are met and no system faults\(^1\) are observed. Once the deactivation criteria are true or system faults are observed, the electric supercharger is deactivated and the bypass valve opened.

There are several considerations in designing the activation strategy. First, the electric supercharger only needs to be activated when necessary, that is when the vehicle drivability is affected by the slow response of the turbocharger. Second, proactive action is desired so that the strategy does not have to wait until drivability is degraded. Last, the strategy should reduce unwanted activation for driver change-of-mind events.

Recall the boost pressure ratio deficit \(\Pi^*\) in Equation (3). \(\Pi^*\) can be used as an index to determine the driver demand deficit that represents drivability. The proactive action is achieved by looking at the rate of change of \(\Pi^*\). Two first order low pass filters \((H_{l,i=1,2})\) are added, as shown in Equation (13) and (14), to mitigate “false positive” activation during the driver change-of-mind events, e.g. driver aggressively pressing the acceleration pedal but rapidly removing the command.

\[ H_{1,k} = (1 - h_1) H_{1,k-1} + h_1 \Pi^*_k \]  
(13)

\[ H_{2,k} = (1 - h_2) H_{2,k-1} + h_2 (\Pi^*_k - \Pi^*_{k-1}) \]  
(14)

\[ h_i = \frac{T_s}{\tau_i + T_s}, i = 1,2 \]  
(15)

where \(h_i\) is the two time constant of filter \(H_i\) in the discrete domain. The electric supercharger and bypass valve are activated when \(H_2\) exceeds threshold \((f_2)\), which is a function of \(H_1\), i.e. when

\[ H_2 > f_2(H_1) \]  
(16)

\(^1\): System fault is a general term that combines all relevant system faults, for example, electric supercharger system, turbocharger system as well as any engine system faults that will limit the capability for air delivery.
To illustrate the activation strategy, a sample acceleration trajectory is plotted in Figure 4. The trajectory is constructed with X-axis being the filtered boost pressure deficit and Y-axis being the filtered rate of change of the boost pressure deficit. The activation threshold is a function of filtered $\Pi^*$. As calibrated, the electric supercharger will be activated when $\Pi^*$ is increasing rapidly even when the absolute $\Pi^*$ is low. On the other hand, when $\Pi^*$ is gradually developed to reach an unacceptable level, the strategy activates the electric supercharger.

![Figure 4](image_url)

Figure 4 the electric supercharger activation function

The main criteria to deactivate the electric supercharger is when the turbocharger is able to sustain the desired boost, or the boost pressure deficit is reduced and below a threshold.

$$\Pi^* \leq \alpha$$  \hspace{1cm} (17)

Where $\alpha$ is a calibratable constant. In addition, there are several supporting features in the activation and deactivation criteria not discussed here, including hardware protection, anti-toggle hysteresis and control override.

5. VEHICLE RESULTS

In order to illustrate the control design, an acceleration maneuver is performed in the test vehicle. The resulting boost system responses are shown in Figure 5 and Figure 6. Both figures show time traces of boost pressure ratios, $\frac{p_2}{p_0}$, $\frac{p_3}{p_0}$, and $\frac{p_2}{p_1}$ on the top; the middle plot shows the actuator command and response of the electric supercharger system including the bypass valve (eSC bpv), the final speed command ($n_{esc \ cmd}$), the measured actual speed ($n_{esc \ actual}$), and the control actions from feedback controller ($n_{esc fb}$); the bottom plot is the wastegate command of the turbocharger system.

![Figure 5](image_url)

Figure 5 Vehicle results of boost system control action and boost pressure response. Electric supercharger is controlled with open loop controller.
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REFERENCES


Figure 6 Vehicle results of boost system control action and boost pressure response. Electric supercharger is controlled with closed-loop controller.

6. CONCLUSION

A dynamic pressure ratio allocation controller is developed for the series sequential compound boosting system with an electric supercharger and turbocharger. It addresses the difficulties of controlling an overactuated control system. The proposed controller has been validated in a test vehicle with good overall performance of boost pressure delivery.

The control strategy is designed to handle the distinctive characteristics of the two boost devices, a fast transient only electric supercharger and a slower but capable and efficient turbocharger. The dynamic pressure ratio allocator apportions the boost pressure deficit to electric supercharger while the main turbocharger is controlled to deliver the entire desired system pressure ratio target.

Compared to the authors’ prior solution, the frequency separation controller, the dynamic pressure ratio controller is simpler to calibration and more robust to the dramatic changes in turbocharger response that occur over the engine operating range. Compared to known approaches for compound boosting systems, the presented dynamic allocation approach coordinates the boosting devices based on their dynamic characteristics.