Linear-Quadratic Tracking Control of a Commercial Vehicle Air Brake System

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ABSTRACT This article proposes to utilize linear-quadratic tracking (LQT) control to reduce the air brake system response time and vehicle stopping distance, and hence, to significantly improve the system performance of a commercial vehicle air brake system equipped with electro-pneumatic proportional valve actuators. The nonlinear dynamic model of the air brake system, consisting of a control actuator (a proportional valve) and braking actuator (the brake chamber), is developed and linearized using the \textit{q}-Markov COVariance Equivalent Realization (\textit{q}-Markov Cover) method in our early work. Based on the linearized dynamic model, an infinite horizon LQT controller is designed along with Kalman state estimation at each linearized operational condition. To apply the LQT control law over a wide operational range to track the target pressure, the designed controller was interpolated between the neighboring controllers to have a control law cover the entire operational range. To validate this control law, the control scheme is implemented into a dSPACE unit and validated through bench tests under different supply and reference pressures. The LQT control performance is also compared with the PID (proportional-integral-derivative) one. The bench test results confirm the effectiveness of the proposed control scheme.

INDEX TERMS Air brake system, proportional valve, control-oriented model, linearization through system identification, linear quadratic tracking.

I. INTRODUCTION

As more and more vehicles get on the road, their safety is crucial. One of the key vehicle safety subsystems is its brake system since it plays an essential role in collision avoidance. Vehicle brake systems have been evolved from rear- or front-wheel brake system only to all-wheel one to improve brake efficiency [1]. Further brake system improvement is made for improving driving safety, such as anti-lock braking (ABS) [2] and adaptive cruise control (ACC) [3]. There are many research literature focusing on the dynamic characteristics of the ABS solenoid valve [2] and the associated ABS control strategies such as fuzzy logic control [4], sliding-mode control [5], linear quadratic control [6], iterative learning [7], nonlinear adaptive control [8], and other control methods [9]. ABS has been used in most passenger cars and commercial vehicles for improving vehicle driving safety and its steering ability [10]. On the other hand, the ABS also requires the driver to utilize it to fulfill its performance [11] properly.

Note that most of the commercial vehicles use air brake systems because of its high brake force capacity. Due to the complex pipe layout of the air brake system in a commercial vehicle, brake response delay (or lag) is one of the vital air brake system performances challenges since the air is compressible. Reducing response time delay between the time when the brake pedal is pressed and that when the brake system generates brake force is necessary. To utilize model-based control, the precise physical system model for analyzing system characteristics is crucial. Zamzamzadeh \textit{et al.} [12] analyzed the effect of driver pedal force to brake distance through multi-body dynamic
The main contributions of this article are three-fold: a) the LQT controllers are designed based on the Linear Parameter-Varying (LPV) model of the air brake system with the proportional valve to improve the performance of the system at each operating point, where the detailed modeling work is shown in our early work [28]; b) the designed controllers are interpolated to cover the entire operating range; and c) the interpreted LQT controller is validated on a test bench and the simulation and experimental results are compared with that of the PID controller.

II. SYSTEM OVERVIEW AND CONTROL OBJECTIVE

A. AIR BRAKE SYSTEM WITH ELECTRO-PNEUMATIC PROPORTIONAL VALVES

There are many power-assisted vehicle brake systems such as hydraulic, pneumatic, and electrical ones. The air brake system is widely used in heavy-duty commercial vehicles. The overall air brake system for a commercial vehicle consists of pneumatic and mechanical subsystems. The pneumatic brake system includes an air compressor, storage reservoir, brake line, quick release valve, treadle valve, relay valve, brake chamber, etc. and the mechanical system includes mainly push rod, slack adjuster, and so on (see [29] for details). To reduce brake distance and the response time from the time when the brake pedal is pressed to the time...
when brake force is generated, the treadle valve in the air brake system is replaced by two proportional valves in this study. The simplified pneumatic brake system is shown in Figure 1 below. When the driver presses the brake pedal, compressed air is supplied to the brake chamber from the reservoir through the supply solenoid valve; and when the brake pedal is released by the driver, the air is exhausted from the brake chamber through the exhaust solenoid valve, returning the push rod in the brake chamber.

**B. EXPERIMENT SETUP**

An air brake system test bench was constructed, consisting of an air reservoir, proportional valve, precision regulator, brake chamber, amplifier and a dSPACE microcontroller; see Figure 2 for details. The gas source is generated by an air compressor to provide compressed air with a pressure between 1 and 9 bar, and a precision regulator is used to provide an accurate target supply pressure. The supply air flows through the tank reservoir to stabilize the airflow. A proportional valve is used to adjust the brake chamber pressure to its reference level. Note that the brake chamber pressure is the control target.

For this study, experiments were conducted under two different supply pressure levels, 4.5 and 5.8 bar, while the reference pressure was kept at 2, 3, and 4 bar; see Figure 3 that shows experimental results of the brake chamber pressure under different reference and supply pressure levels with a well-tuned PID controller.

**C. CONTROL PROBLEM FORMULATION**

For the air brake system, the main control objective is to make the brake chamber pressure $y$ follow the reference (desired) one $r$. By carefully following the brake chamber pressure reference signal, the brake response time can be significantly reduced, leading to reduced stopping distance. Note that the control output $u$ is used to control the solenoid valve voltage in the proportional valve, and the feedback is the measured brake chamber pressure; see Figure 4.

**III. SYSTEM MODEL**

**A. SYSTEM DYNAMICS**

The proportional valve used in this article is part of the ITV series from SMC corporation [30]. In [30], the related pneumatics and hydraulics components are introduced in detail so that the structure, function and application of each component can be found. The proportional valve consists of a pair of solenoid valves, a pilot chamber, a spool, and a pilot valve; see Figure 5 for details.
The following assumptions are made for this model: 1) leakage in the chamber is negligible; 2) the orifice flow process is isentropic; 3) air is treated as ideal gas; 4) Coulomb friction effect is ignored; and 5) the inertia dynamics of the brake chamber diaphragm is neglected.

With the voltage input signal, the supply and exhaust solenoid valves can be turned on or off, respectively. The mass flow rate through the supply solenoid valve can be defined by a quasi-steady-state isentropic orifice flow equation as a function of the needle displacement. Similarly, the mass flow rate through the exhaust solenoid valve also can be defined. Then the mass flow rate through the pilot chamber is the flow difference \( \delta q_{m2} \) between the supply solenoid valve and the exhaust solenoid valve. As a result, gas flows through the pilot chamber, and the pressure change in the pilot chamber can be calculated.

As the pilot chamber pressure changes, the push rod of spool moves up and down. Spool valve displacement is a function of pilot chamber pressure, outlet pressure, spring and damping forces applied to the spool. Also, the mass flow rate properties of the exhaust valve can be found similarly. There are three springs in this valve, which are spring 1 under the diaphragm \( \delta \) of Figure 5, spring 2 in the exhaust valve \( \delta \) of Figure 5, and spring 3 in the supply valve \( \delta \) of Figure 5. So the analysis of the spool is divided into three parts, which are the pushrod (part 1), exhaust valve (part 2), and supply valve (part 3). The equation of motion can be derived by Newton’s second law (1), as shown at the bottom of the page.

In equation (1), subscripts 1, 2, 3 represent the first, second, and third parts, respectively. \( m_1, m_2, \) and \( m_3 \) are the mass of spring 1 and piston, spring 2, and spring 3, respectively; \( B_1, B_2, \) and \( B_3 \) are the damping coefficient of springs 1, 2, and 3; \( k_1, k_2, k_3 \) is coefficient of elasticity of springs 1, 2, and 3; \( a_1 \) is the diaphragm area of pilot chamber; \( a_2 \) is the contact area of push rod; \( f_{10}, f_{20}, \) and \( f_{30} \) are preload of springs 1, 2, and 3, respectively; \( y \) is the spool valve displacement; \( P_{\text{pil}} \) is the pressure in pilot chamber; and \( P_{\text{out}} \) is the pressure in the brake chamber.

When gas flows through the brake chamber, the brake chamber pressure increases and the push rod moves. Considering the structure of the brake chamber with a push rod, using Newton’s second law, and neglecting the inertia effect of the brake chamber diaphragm, the relationship between brake chamber pressure and mass flow through brake chamber can be found in (2) below.

\[
P_{\text{out}} = \begin{cases} 
\delta q_{m2} \gamma RT_0 P_0^{\gamma-1} / V_{01} P_0^{\gamma-1} & x_b = x_b \max \\
\delta q_{m2} / (V_{02} P_0^{\gamma-1} / V_{02} P_0^{\gamma-1} + A_2 P_{\text{out}}^{\gamma-1} / R \gamma \delta k_b) & 0 < x_b < x_b \max \\
\delta q_{m2} \gamma RT_0 P_0^{\gamma-1} / V_{02} P_0^{\gamma-1} & x_b = x_b \max 
\end{cases} \\
\text{where } \delta q_{m2} \text{ is the mass flow rate through the brake chamber; } V_{01} \text{ is brake chamber initial volume; } V_{02} \text{ is its maximum volume; } \gamma \text{ is the ratio of specific heats; } R \text{ is the ideal gas constant; } T_0 \text{ is supply air temperature; } P_0 \text{ is supply pressure; and } P_{\text{out}} \text{ is brake chamber pressure. Finally, the entire non-linear mathematical model architecture is shown in Figure 6. The details of the modeling process can be found in our early work [28].}
\]

**B. SYSTEM LINEARIZATION**

The pneumatic brake system is inherently nonlinear due to spool movement and nonlinear spring characteristics. Designing a nonlinear controller to meet tight performance requirements is very challenging. It is decided to design a linear controller based on the linearized system model at each operational condition, and the designed set of the controller at different operational conditions can be interpolated or switched under different operational conditions.

Since linear controllers will be designed based on the linearized models, it is crucial to obtain a set of linearized mathematical models.
models. The \(q\)-Markov COVariance Equivalent Realization (\(q\)-Markov Cover) method for system identification is applied to the pneumatic brake system to obtain linearized models under different operational conditions (different brake chamber pressure) in this article. A PRBS (pseudo-random binary signal) is used as a disturbance input and added to the brake chamber reference signal, where the air brake system is controlled in a closed-loop with a proportional controller. The system output is the brake chamber pressure minus the steady-state pressure due to input reference signal without the PRBS. Note that closed-loop system identification is conducted to obtain the closed-loop transfer function, and then, the open-loop transfer function is solved based on the known controller transfer function. The PRBS is generated based on the maximum length sequences (also called m-sequence [15]), where the PRBS length is \(m = 2^n - 1\) and \(n\) is an integer called PRBS order. A PRBS contains \(m = 2^{n-1}\) ones and \(m = 2^{n-1} - 1\) zeros. In this article, the zero-mean inversed PRBS is used; see [15] for details. The complete linearization process can be found in our early publication [28].

In this article, a 10th order inversed PRBS signal is generated. Under reference target pressure of 2 bar and supply pressure of 5.8 bar, the PRBS input signal with a magnitude of 0.4 bar and 0.12s sample period is used for system identification, where both system input and output are sampled at 0.01s. After the input and output signals were collected, they are fed into the PRBS system identification GUI (graphic user interface) [15], and the discrete-time state-space model (linearized) of the second-order closed-loop system can be obtained. Then the discrete-time transfer function of the air brake system can be solved from the identified closed-loop system model. Under the different reference pressures, the form of the system transfer function does not change, but the coefficients vary. As a result, the model can be simplified as below.

\[
G_P(z) = \frac{\theta_1 z + \theta_2}{z^2 + \theta_3 z + \theta_4} \tag{3}
\]

Then, the system plant model is constructed under different operational conditions as a function of reference pressure for a given supply pressure; see Table 1. From Table 1, a trend is found after data analysis, that is, \(\theta_1 \approx -0.0078\theta_3\) and \(\theta_4 \approx -0.48\theta_3\). As a result, the model can be simplified as below.

\[
G_P(z) = \frac{-0.0078\alpha_1 z + \alpha_2}{z^2 + \alpha_1 z - 0.48\alpha_1} \tag{4}
\]

The simulation results comparing nonlinear and linear models with 2 bar target pressure are shown in Figure 7.

**C. DISCRETE-TIME STATE-SPACE MODEL**

Consider the discrete-time linear system below

\[
\begin{align*}
    x(k+1) &= Ax(k) + Bu(k) + Dw(k) \\
    y(k) &= Cx(k) \\
    z(k) &= Mx(k) + v(k)
\end{align*} \tag{5}
\]

where \(x(k)\) is the state vector; \(u(k)\) and \(w(k)\) are the control input and system disturbance input, respectively; \(y(k)\) and \(z(k)\) are system performance output and measurement,
respectively; \(v(k)\) is measurement noise; matrices \(A, B, D, C,\) and \(M\) are the system dynamic matrices, the system input matrix, system noise input matrix, the system output matrix, and measurement matrix, respectively.

Based on the discrete-time state-space equations of the plant system obtained from the system identification under different reference and supply pressure levels, the plant system is shown below in equation (6).

\[
A = \begin{bmatrix} -\alpha_1 & 0.48\alpha_1 \\ 1 & 0 \end{bmatrix} \\
B = [1 \ 0]^T, \quad D = B \\
C = [-0.0078\alpha_1 \ \alpha_2], \quad M = C \quad (6)
\]

**IV. LINEAR QUADRATIC TRACKING (LQT) CONTROL**

To improve braking performance, the infinite horizon LQT control strategy [31] is proposed to make the brake chamber pressure track the reference pressure. To be more specific, based on the simulation model, the control target is to make the system output track the desired pressure as close as possible. Since the state vector cannot be measured, a Kalman filter is used as an optimal state observer.

**A. INFINITE HORIZON LQT**

The control objective is to minimize the tracking error \(e(k)\) between the desired system output \(r(k)\) and actual system output \(y(k)\), assuming that the pair \((A, B)\) is stabilizable and the pair \((A, \sqrt{Q}C)\) is detectable. The tracking error is defined in equation (7) below.

\[
e(k) = y(k) - r(k) = C(k)x(k) - r(k) \quad (7)
\]

The performance cost function of the infinite horizon LQT controller is shown in equation (8) below.

\[
J = \frac{1}{2} \sum_{k=0}^{\infty} [e^T(k)Qe(k) + u^T(k)Ru(k)] \quad (8)
\]

where \(Q = Q^T \geq 0\) and \(R = R^T > 0\). The optimal tracking problem can be solved by following the minimum principle in [31]. With the Hamiltonian equation, the optimal control \(u(k)\) can be obtained; see the following equation (9).

\[
u(k) = -L_{FBg}x(k) + LF_{gg}(k + 1) \quad (9)
\]

where \(L_{FBg} = [R + B^T PB]^{-1}B^T PA\) is the feedback gain; \(L_{Fgg} = [R + B^T PB]^{-1}B^T\) is the feedforward gain; and matrix \(P = P^T \geq 0\), can be obtained by solving the following Algebraic Riccati Equation (ARE).

\[
P = A^T P[I + B R^{-1} B^T]^{-1} A + C^T Q C \quad (10)
\]

and \(g(k)\) is a function of \(r(k)\) and defined as below.

\[
g(k) = [A^T - A^T PB(R + B^T PB)^{-1} B^T] g(k + 1) + C^T Q r(k)
\]

\[
= A^T[I + PBR^{-1}B^T]^{-1} g(k + 1) + C^T Q r(k) \quad (11)
\]

In equation (9), the first term is for feedback control that depends linearly on the system state vector, and the second term is for reference track that depends on the reference trajectory \(r(k)\). Since \(x(k)\) cannot be measured, it needs to be estimated using a Kalman filter, provided that the system is detectable.

The Kalman state estimation is used to provide an optimally estimated state vector \(\hat{x}(k)\) based on the linear quadratic estimation algorithm with white Gaussian noises, where the process disturbance is denoted by \(w(k)\) and the measurement noise \(v(k)\); see Figure 8.

**FIGURE 8. Work principle of LQT.**

The input system noise \(w(k)\) and measurement noise \(v(k)\) are assumed to be zero mean and independent random vectors with the following properties.

\[
E[w(k)] = 0, \quad W = E[w(k)w^T(k)] \quad (12)
\]

\[
E[v(k)] = 0, \quad V = E[v(k)v^T(k)] \quad (13)
\]

where \(E[\cdot]\) is an expectation operator, \(W\) and \(V\) are the corresponding covariance matrices of input and measurement noises, respectively.

The Kalman estimated state \(\hat{x}(k + 1)\) can be obtained through measurement \(y(k)\) and the control \(u(k)\) shown in Figure 8. The observer structure is in the form of (14).

\[
\hat{x}(k + 1) = A\hat{x}(k) + Bu(k) + L(z - M\hat{x}(k)) \quad (14)
\]

where \(\hat{x}(k)\) is an estimation of \(x(k)\); \(L\) is the Kalman filter gain matrix defined by (15).

\[
L = AKM^T(MKM^T + V)^{-1} \quad (15)
\]
where $K$ can be obtained by solving the following algebraic Riccati equation (16)

$$K = AKAT + DWD^T - LMKA^T$$ (16)

Using the estimated state vector in (14), the system control can be constructed as (17).

$$u(k) = -R^{-1}B^TP\hat{x}(k) + R^{-1}B^Tg(k)$$ (17)

Note that due to separation theory [31], stability of the LQT controller with the Kalman state estimate is guaranteed.

V. THE PID CONTROLLER

A PID controller, based on the error between the reference and system output pressure levels, is used in this article for comparison purposes. The PID controller has the following form:

$$u(k) = K_P e(k) + K_I \sum_{i=0}^{k} e(i)T_s + K_D \frac{e(k) - e(k - 1)}{T_s}$$ (18)

where $u(k)$ is the control output, $e(k)$ is the error between the reference and system output pressure levels, $k$ is the sample index, $T_s$ is the sample period, $K_P$ is the proportional gain, $K_I$ is the integral gain, and $K_D$ is the derivative gain. The PID controller used in the system is shown in Figure 9.

![FIGURE 9. Closed-loop control system with PID controller framework.](image)

VI. SIMULATION VALIDATION

The LQT controller is designed based on the discrete-time dynamic model and implemented in Simulink. Under different reference pressure levels, the closed-loop system performance can be validated through a simulation study. To achieve better tracking performance and system stability, the control design parameters, including weighting matrices $Q$ and $R$, are tuned in the simulation study.

By analyzing the state-space model, the system is controllable and observable. Solving the LQT and Kalman filter related AREs with known $Q$, $R$, $W$, and $V$ leads to solutions of equations (10) and (11); and the Kalman filter gain $L$ can be solved through equation (14). Simulation results under different reference and supply pressure levels are obtained, and the values of tuned $Q$ and $R$ are shown in Table 2. By varying the LQT weighting matrices $Q$ and $R$, the system tracking performance, response time and overshoot can be optimized.

With the LQT controller, the simulation results for both linear and nonlinear systems are obtained. By tuning the PID parameters of $K_P$, $K_I$, and $K_D$ in Matlab/Simulink, simulation results are obtained. Comparing with the results from this well-tuned PID controller, the simulation results under the reference pressure of 2 bar with step input and Sinusoid input are shown in Figures 10 and 11, respectively.

From Figures 10 and 11, it can be observed that the rates of charge and discharge processes with the LQT controller are faster than those of the well-tuned PID controller. We believe that this is due to the fact that the nonminimal phase characteristics of the air brake system prevents having high PID control gains. Also, the tracking performance of the LQT controller is better than that with the PID controller, and oscillations at the beginning of the charging process with the LQT controller is smaller than that of the PID controller.

| Reference pressure/bar | Q   | R   |
|------------------------|-----|-----|
| (1)                    | 2   | 24  |
| (2)                    | 3   | 22  |
| (3)                    | 4   | 20  |

![FIGURE 10. Simulation results comparison between LQT controller and PID with step input.](image)

![FIGURE 11. Simulation results comparison between LQT controller and PID with Sinusoid input.](image)
VII. EXPERIMENT VALIDATION

Experimental validation was conducted on an air brake system test bench. The test bench was constructed with a switch valve, a pressure regulator, an air supply tank, a reserve compact pneumatic pressure sensor PSe540, a USB data acquisition card (USB 6009), an air chamber, and a dSPACE microcontroller with associated amplifier. The proposed discrete-time LQT controller with a Kalman Filter is implemented into the dSPACE microcontroller in Simulink. The control and mechanical parts of the proportional valve are independent. Since the dSPACE control output voltage range is between 0 and 5V and the solenoid valve control input voltage is 24V, an amplifier (see Figure 2 (b)) (under the red board) is required to drive the solenoid. To prevent the interference between the control signals of the input and exhaust valves, the input and exhaust amplify circuits are independent. Through both amplifiers, control signals are provided to the two control solenoids, respectively. The system output pressure is measured by a pneumatic pressure sensor and the data is recorded by the USB 6009 data acquisition card.

A comparison between simulation and experiment pressures of both controllers is shown in Figure 12. From Figure 12, the simulation and experiment pressures track well during the charging process and at the steady-state, and the simulation response rate of the discharge process is slower than that of the experiment.

VIII. INTERPOLATED CONTROLLER DESIGNING

Based on the controllers designed for reference pressure levels of 2, 3, and 4 bar, the goal is to generate a compound control scheme that can be used to achieve the reference pressure at any level from atmosphere to supply pressure. The controller with the reference pressure of 2 bar is used in the system with the target pressure below 2 bar; and the controller with the reference pressure 4 bar is used in the system with the target pressure above 4 bar. Between 2 and 3 bar the controller is a convex combination of the controllers designed for 2 and 3 bar; and between 3 and 4 bar the controller is a convex combination of the controllers designed for 3 and 4 bar. For example, for the target pressure of 2.3 bar, 70% of the controller under 2 bar target pressure and 30% of the controller under 3 bar target pressure are used in the system with 2.3 bar target pressure. The control gain is expressed as equation (19).

\[
L = \begin{cases} 
L_2 & r \leq 2 \\
(3 - r)L_2 + (r - 2)L_3 & 2 < r \leq 3 \\
(4 - r)L_3 + (r - 3)L_4 & 3 < r < 4 \\
L_4 & r \geq 4 
\end{cases} \quad (19)
\]

where \( L \) is the selection of the controller; \( L_2, L_3, L_4 \) is controllers designed at 2 bar, 3 bar, and 4 bar, respectively; and \( r \) is the reference pressure.

With this compounded controller, the comparison results between simulation and experimental results under 1.42 bar, 2.58 bar, 3.67 bar, 4.72 bar reference pressure and 5.8 bar supply pressure are shown in Figure 14. Considering a practical signal with varying brake pressure, a simulation study is conducted with the LQT controller and the associated simulation results are shown in Figure 15. From Figures 12, 13, 14 and 15, it can be observed that the performance with the interpolated controllers is better than...
these for individual controller designed at 2, 3, and 4 bar. With the interpolated controllers, the tracking performance is improved.

IX. CONCLUSION

In this article, a linear-quadratic tracking (LQT) controller with a Kalman state estimator is designed based on a set of linearized models obtained using q-Markov Cover (q-Markov COVariance Equivalent Realization) system identification of a nonlinear model developed for an air brake system. It is necessary to establish an accurate model for the LQT controller design. The linear-quadratic tracking (LQT) controller is constructed based on the linearized pneumatic brake system model. Comparing with the well-tuned PID controller, the proposed LQT controller improves the system response and tracking performance with the small oscillations under different supply and reference pressure levels. To expand the application scope and improve the controller usability for the system, the interpolated LQT controller at different reference pressure levels based on the set of individual LQT controller is designed and experimentally validated on a test bench. The validation results match with simulation ones very well. The interpolated LQT controller is able to track the desired input with minimal tracking error. Experimental data illustrated the effectiveness of the proposed controller design scheme.

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