AN OPTIMAL FEEDBACK LINEARIZATION APPROACH FOR THE SOFT LANDING OF ELECTROMECHANICAL VALVE ACTUATOR IN CAMLESS INTERNAL COMBUSTION ENGINES

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***Abstract*— Recently, the fuel efficiency, maximum torque and power, and pollutant emissions of internal combustion engines have improved more remarkably compared to conventional crankshaft systems as a result of replacing the crankshaft with the camless actuation system. Some of the main conflicting objectives in camless engines include soft landing, tracking of the desired trajectory and valve opening and seating in very fast time. Due to the nonlinear model of the system, the present study aimed to propose feedback linearization control to manage the challenges of the system. In order to optimize the system response, the concept of feedback linearization control is incorporated into the linear quadratic tracker, resulting in the proposed structure known as FLQT. The simulation results in the normal case, as well as in the presence of external disturbance, indicated that the optimal feedback linearization controller had proper performance in the realization of the desired system objectives.**

***Keywords*— Feedback linearization control; Linear quadratic tracker; Camless engine; Electromechanical valve actuator; Internal combustion engine.**

I. INTRODUCTION

In conventional crankshaft valve actuation systems, a fixed valve lift is realized in a constant timing for any engine speed and all load ranges (Haus *et al*., 2017). Although these systems are cost-efficient and have a simple structure, their performance is not optimal in all circumstances (Qiu *et al*., 2012; Zhao and Seethaler, 2011). In camless internal combustion engines, the performance of the system consists of fuel economy, torque output, and lower NOx emissions, which could be enhanced owing to the variations in the time of valve opening and closing (Hoffmann *et al*., 2003).

An electromechanical valve (EMV) actuator could be constructed with several structures. The system could be developed in various models depending on the selected structure and utilized equipment. In the present study, a nonlinear model was considered based on the states of the system as the armature position, velocity and coil current, as well as the system input as the applied voltage and system output as the armature position. The model could be used to describe the specifications of the system and overcome the mentioned challenges (Samadi and Saif, 2011; Samani and Khodadadi, 2017).

According to the literature, several techniques could be applied to overcome the mentioned challenges, including iterative learning (Hoffmann *et al*., 2003), nonlinear self-tuning control (Peterson *et al*., 2002a), optimal approaches (Fabbrini *et al*., 2008), predictive control method (Zhu *et al*., 2009), output observer-based feedback (Peterson *et al*., 2002b), feedback linearization approach (Haskara *et al*., 2004), feedforward and linear quadratic regulator (LQR) (Chun and Tsu-Chin, 2003), adaptive control strategies (Mercorelli and Werner, 2017; Ma *et al*., 2011), cycle adaptive feedforward approach (Tsai *et al*., 2008), sliding mode and optimal sliding mode controller (Samani and Khodadadi, 2017).

In terms of industrial applications, studies have been focused on electromagnetic valve actuators (Mercorelli, 2014; Braune *et al*., 2006), proposing theoretical and experimental results for the operation of linear electromagnetic motors as an engine valve actuator. The control structure consists of two proportional-derivative controllers, including position control and current control.

Although the more researchers have considered the system model in the linear or linearized forms around the equilibrium points (Mercorelli, 2014; Braune *et al*., 2006), we aimed to investigate the nonlinear model of the EMV actuator in the present study. The feedback linearization control approach is applied to manage the system challenges. In order to optimize the system performance, the concept of feedback linearization is incorporated into the linear quadratic tracker (LQT) and referred to as FLQT. To the best of our knowledge, the optimal feedback linearization approach for trajectory tracking has not been adequately studied in the EMV actuators of camless engines. According to the simulation results, the optimal feedback linearization controller had proper performance in the realization of the desired system objectives, including soft landing, short transient time, and trajectory tracking.

The outline of the paper is as follows: the EMV actuator model has been described in Section II, the proposed controller for the tracking of the desired trajectory of the valve position has been elucidated in Section III, which is composed of developing the feedback linearization method and its combination with the optimal strategy. Section IV has been dedicated to the simulation results in two cases of normal status and the presence of external disturbance. The paper concludes in Section V.

II. EMV ACTUATOR MODEL

The EMV actuator in the present study is similar to the system proposed by Chung *et al.* (2007), in which two opposing electromagnets provide the required force for valve movement. Moreover, two springs are utilized to store the mechanical energy. The schematic view of the EMV actuator is depicted in Fig. 1.

The nonlinear model proposed by Samadi and Saif (2011) and Samani and Khodadadi (2017) which was obtained by the combination of electrical and mechanical equations is employed for the evaluation of the system. The system model could be derived in the state space form, as follows:

$$\begin{cases} \dot{X} = F(X,t) + G(X,t)U \\ Y = H(X,t) \end{cases} \tag{1}$$

where

$$(X,t) = \begin{bmatrix} x_2 \\ \frac{1}{m}\left[\frac{\lambda_s f(x_1)}{f^2(x_1)}\left(1-(1+x_3 f(x_1))e^{-x_3 f(x_1)}\right) - K_s x_1 - B x_2\right] \\ -R_c \frac{e^{x_3 f(x_1)}}{\lambda_s f(x_1)} x_3 - \frac{\dot{f}(x_1)}{f(x_1)} x_3 x_2 \end{bmatrix} \tag{2}$$

$$f(x_1) = \frac{2c_1}{c_2 - x_1} + c_3 \tag{3}$$

$$X = [x_1, x_2, x_3] = [x, v, i]^T \tag{4}$$

$$G(X,t) = \begin{bmatrix} 0 \\ 0 \\ \frac{e^{x_3}}{\lambda_s f(x_1)} \end{bmatrix} \tag{5}$$

In the above equations, the position (x) and velocity (v) of the electromagnet armature and the coil current (i) are composed of the state space variables (X), and U and Y are the vectors of input and output of the system, respectively. Due to the considered SISO model of the EMV actuating system, these vectors are converted to $U = u$ and $Y = y$. The applied voltage and armature position represent the system input and output, respectively. The flux saturation $f(x_1)$ is calculated as (3), and c_1, c_2 and c_3 are the parameters that could be estimated through least squares method for the curve fitting of the experimental force data. Moreover, the coil resistance and spring constant are denoted as Rc and K_s, respectively. Equations 6 and 7 are also applied to represent the magnetic flux and system output, respectively. The numerical values of the system parameters are presented in Table 1 (Samadi and Saif, 2011; Samani and Khodadadi, 2017).

$$\lambda(x_1, x_3) = \lambda_s\left(1 - e^{-x_3 f(x_1)}\right), \quad x_3 \geq 0 \tag{6}$$

$$H(X,t) = x_1 \tag{7}$$

III. CONTROLLER DESIGN

A. Feedback Linearization

Feedback linearization (FL) is an effective control approach that could be applied to nonlinear systems (Shulong *et al.*, 2014). The method is primarily based on the conversion of nonlinear dynamics into linear dynamics and applying linear control techniques. The method is based on two operations including the nonlinear change of the coordinates and nonlinear state feedback.

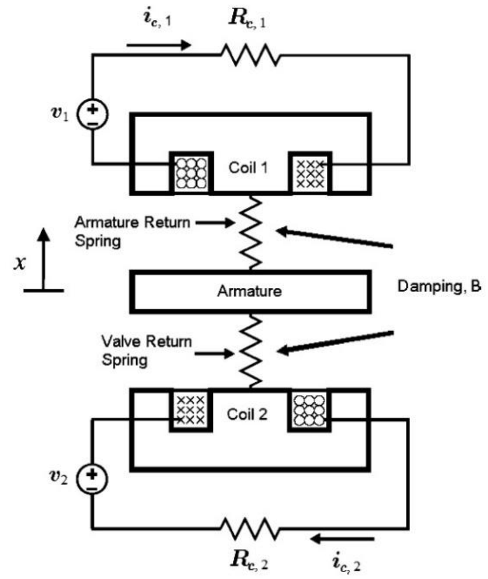

Fig. 1. Schematic of a two-spring valve system (Chung *et al.*, 2007).

Table 1. Numerical Values of System Parameters (Samadi and Saif, 2011; Samani and Khodadadi, 2017)

Parameter	Value	Parameter	Value
x_0	-4mm	c_1	0.0232 mm/A
M	0.28Kg	c_2	4.04 mm
K_s	250.98 N/mm	c_3	4.18×10^{-4} A^{-1}
B	12.75 Ns/m	R_c	0.52 Ω
λ_s	0.076 Wb	T_s	2×10^{-5} s

For the EMV actuator, the FL design is as follows. Considering the system output as $y = x_1$, the output derivatives could be calculated as follows:

$$\dot{y} = \dot{x}_1 = x_2 \tag{8}$$

$$\ddot{y} = \dot{x}_2 = \frac{1}{m}\left[\frac{\lambda_s \dot{f}(x_1)}{f^2(x_1)}\left(1-(1+x_3 f(x_1))e^{x_3 f(x_1)}\right) - K_s x_1 - B x_2\right] \tag{9}$$

$$\dddot{y} = \ddot{x}_2 = \frac{1}{m}\left[-k_s \dot{x}_1 - B\dot{x}_2 + \lambda_s \dot{A}(X)\right] \tag{10}$$

where

$$A(X) = \frac{\dot{f}(x_1)}{f^2(x_1)}\left(1-\left(1+x_3 f(x_1)e^{x_2 f(x_1)}\right)\right) = \frac{\dot{f}(x_1)}{f^2(x_1)} - \frac{e^{x_2 f(x_1)}\dot{f}(x_1)}{f^2(x_1)} + \frac{x_3 e^{x_2 f(x_1)}\dot{f}(x_1) f(x_1)}{f^2(x_1)} \tag{11}$$

$$\begin{aligned} \dot{A}(X) = {} & \frac{\ddot{f}(x_1)\dot{x}_1 f^2(x_1) - 2f(x_1)\dot{x}_1\dot{f}(x_1)}{(f^2(x_1))^2} - \\ & \frac{\left[\left(\dot{x}_2 f(x_1) + x_2\dot{x}_1\dot{f}(x_1)\right)e^{x_2 f(x_1)}\dot{f}(x_1) + \ddot{f}(x_1)\dot{x}_1 e^{x_2 f(x_1)}\right] f^2(x_1)}{f^4(x_1)} \\ & - \frac{2\dot{x}_1 f(x_1)\dot{f}(x_1)e^{x_2 f(x_1)}\dot{f}(x_1)}{f^4(x_1)} \\ & + \frac{f^2(x_1)\left[\dot{x}_3 e^{x_2 f(x_1)}\dot{f}(x_1) f(x_1) + x_3\left[\left(e^{x_2 f(x_1)}\dot{f}(x_1) f(x_1)\right)\right]\right]}{f^4(x_1)} - \\ & \frac{2f(x_1)\dot{x}_1\dot{f}(x_1)x_3 e^{x_2 f(x_1)}\dot{f}(x_1) f(x_1)}{f^4(x_1)} \end{aligned} \tag{12}$$

and

$$\dot{x}_3 = -R_c \frac{e^{x_3 f(x_1)}}{\lambda_s f(x_1)} x_3 - \frac{\dot{f}(x_1)}{f^2(x_1)} x_3 x_2 + \frac{e^{x_3}}{\lambda_s f(x_1)} u \tag{13}$$

Accordingly, the relative degree of the system is equal to three. The systems with the relative degree of equal to the order of the system have proper inversion and stabilization qualities. In a nonlinear system, the zero dynamic is the uncontrollable segment of the system, which may cause system instability (Mercorelli, 2009). Since the relative degree of the system is equal to its order, the system does not have zero dynamic, and there is no need to assess its stability. As such, considering Eq. (12), Eq. (10) could be rewritten as follows:

$$\dddot{y} = \frac{-k_s}{m}\dot{x}_1 - \frac{B}{m}\dot{x}_2 + Z(X) + \frac{\dot{x}_3 E(X)}{f^4(x_1)} \tag{14}$$

The Eq. (14) is written as Eq. (15) employing Eq. (16).

$$\dddot{y} = \frac{-k_s}{m}\dot{x}_1 - \frac{B}{m}\dot{x}_2 + I(X) + \frac{E(X)e^{x_3}\,u}{\lambda_s f(x_1) f^4(x_1)} \tag{15}$$

$$I(X) = Z(X) + \frac{E(X)}{f^4(x_1)}\left[R_c \frac{e^{x_3 f(x_1)}}{\lambda_s f(x_1)} x_3 - \frac{\dot{f}(x_1)}{f^2(x_1)} x_3 x_2 + \frac{e^{x_3}}{\lambda_s f(x_1)}\right] \tag{16}$$

Moreover, by defining the $D(X) = \frac{E(X)e^{x_3}}{\lambda_s f(x_1) f^4(x_1)}$, Eq. (15) could be converted into Eq. (17).

$$\dddot{y} = \frac{-k_s}{m}\dot{x}_1 - \frac{B}{m}\dot{x}_2 + I(X) + D(X) \times u \tag{17}$$

Considering $\dddot{y} = V$, the control signal for the EMV system could be calculated, as follows:

$$u = \frac{1}{D(X)}\left[V + \frac{k_s}{m}\dot{x}_1 + \frac{B}{m}\dot{x}_2 - I(X)\right] \tag{18}$$

Using the pole placement method and by defining the error signal as $e(t) = y(t) - y_d(t)$, V could be derived as Eq. (19).

$$V = \dddot{y}_d + k_1(\ddot{y}_d - \ddot{y}) + k_2(\dot{y}_d - \dot{y}) + k_3(y_d - y) \tag{19}$$

Selection of appropriate values for k_1, k_2, and k_3 guarantees that the system error converges to zero. In this system, the gains are selected as $k = [k_1, k_2, k_3]^T = [27,27,9]^T$ to place all the system poles on -3.

B. Optimal Control

The main challenge associated with the EMV system is trajectory tracking. As such, the designed controller should be able to track the desired position of the valves (Samani and Khodadadi, 2017). In the current research, Linear Quadratic Tracker (LQT) is the selected optimal strategy to optimize the opening and closing of the EMV actuators. The method is combined with feedback linearization in order to enhance system robustness.

The main purpose of LQT is to provide an optimal control law to minimize the objective function and cause the system to track a predefined trajectory within a determined time interval (Mansouri and Khaloozadeh, 2002). There are two types of LQT approaches, including finite-time interval and infinite-time interval, which have been elucidated in the literature to solve optimization problems. When limited tracking time in the system is required and tracking should be performed within a limited time, the infinite-time interval cannot yield an appropriate response (Naidu, 2003). Therefore, the finite-time interval approach is incorporated into the feedback linearization method in the present study. In this strategy, the system is assumed to be in the linear form of Eq. (20).

$$\begin{aligned}\dot{x} &= Ax + Bu + d\\ y &= Cx\end{aligned} \tag{20}$$

The objective function that should be minimized is as follows:

$$J = \frac{1}{2}(Cx(T) - r(T))^T P (Cx(T) - r(T)) + \frac{1}{2}\int_{t_0}^{T} [(Cx - r)^T Q (Cx - r) + u^T R u]\,dt \tag{21}$$

where the R matrix should be positive definite, while the P and Q matrices are positive semi-definite, and r is the desired output. The parameters Q and R could be used as the design parameters to penalize the state variables and control signals. Selection of a large R value indicates that the designer attempts to stabilize the system with less energy (expensive control strategy), while selecting a small R value highlights the lack of need to penalize the control signals (cost-efficient control strategy). Similarly, if a large value is preferred for Q, it indicates an attempt to stabilize the system with the least possible changes in its states (tracking strategy). Therefore, there is a trade-off between Q and R, and the designer could select a large R value if there is a limit on the control output signal (e.g., if large control signals cause actuator saturation or sensor noise), and a small R values is selected if a large control signal causes no problems in the system. The control signal could be obtained as follows (Lewis *et al.*, 2012):

$$u = -Kx + R^{-1}B^T v \tag{22}$$

where K, S, and v could be obtained by solving Eqs. (23-25).

$$K(t) = R^{-1}B^T S(t) \tag{23}$$

$$-\dot{S} = A^T S + SA - SBR^{-1}B^T S + C^T Q C,\quad S(T) = C^T P C \tag{24}$$

$$-\dot{v} = (A - BK)^T v + Sd,\ \ v(T) = C^T \mathrm{Pr}\,(T) \tag{25}$$

The matrix $S(t)$ is independent of the state trajectory; as such, by solving the Riccati equation, the $S(t)$ and $K(t)$ feedback gain could be obtained. If the reference track $r(t)$ is known, the auxiliary function $v(t)$ could be precomputed. By substituting the attained matrices in Eq. (22), the optimal control policy could be achieved.

IV. SIMULATION RESULTS

In this section, the simulations of the EMV actuator have been discussed for the proposed controllers. The dynamic model of the EMV system (Eqs. 1-7) was applied, and considering the numerical values in Table 1, the numerical model could be used for this purpose. The desired profile for valve movement has been presented in Equation 26, in which, a typical 2-mm displacement at 1800 rpm is constructed (Samani and Khodadadi, 2017).

$$\begin{cases} xref = sat(8\sin\left(\frac{2\pi}{T}\right)) & 0 < t < 0.033 \\ \text{lower band} = 2,\ \text{upper band} = 4 \end{cases} \tag{26}$$

In Eq. (26), T shows the period and is equal to 33 milliseconds, and $sat(.)$ represents the saturation function with the lower and upper bands of two and four, respectively.

Regarding the nonlinear model proposed for the EMV actuator, application of the feedback linearization method resulted in a linear model that is described in Eqs. (27) and (28).

$$\dot{x} = Ax + Bu \tag{27}$$

$$A = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & 0 & 0 \end{bmatrix}, \qquad B = \begin{bmatrix} 0 \\ 0 \\ 1 \end{bmatrix} \tag{28}$$

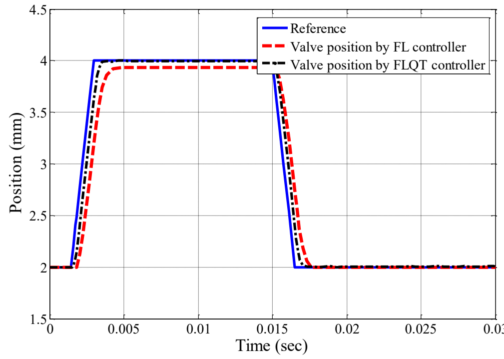

Fig. 2. Desired and valve positions for FL and FLQT controllers.

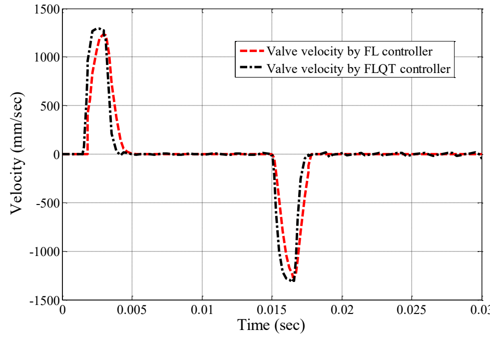

Fig. 3. Valve velocity profiles for FL and FLQT controllers.

In addition, the R, P and Q weighting matrices could be selected, as follows:

$$Q = \begin{bmatrix} 10 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}, \qquad R = [1], \qquad P = [1] \quad (29)$$

In the current research, the simulations are performed in two cases, including the normal case and disturbance case. Figure 2 depicts the results of the desired valve position and EMV actuator movement in the feedback linearization method and feedback linearization method combined with LQT, respectively. Figures 3 and 4 show the other system states, including the velocity and current for both controllers.

As can be seen in Fig. 2, the suggested structures for the controllers had appropriate performance in the tracking of the desired reference. As one of the foremost features of the camless engine, soft landing could be realized for both controllers. Additionally, the opening and closing of the EMV actuator are obtained within the short transient time, and the system had no overshoot in the system output for both controllers. On the other hand, the combination of the optimal structure and feedback linearization caused the system to have a less significant increment and settling time compared to FL. Moreover, the optimal structure of the FLQT infinite horizon resulted in valve position tracking, which is performed without steady-state errors. Accordingly, the proposed controller could meet the design performance requirements as expected. Furthermore, the valve velocity and current profiles of the FL and FLQT controllers demonstrated the landing velocity and the armature current of the EMV are in the permissible limits (Figs. 3 and 4).

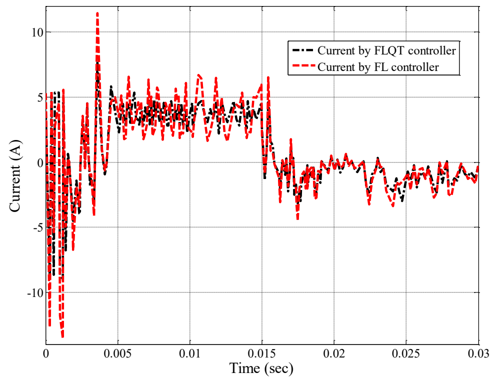

Fig. 4. Current profiles of FL and FLQT controllers.

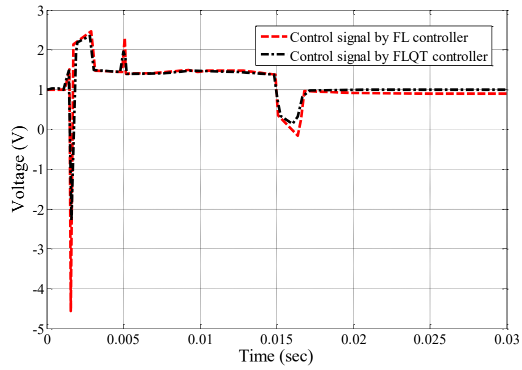

Fig. 5. Control efforts of FL and FLQT controllers.

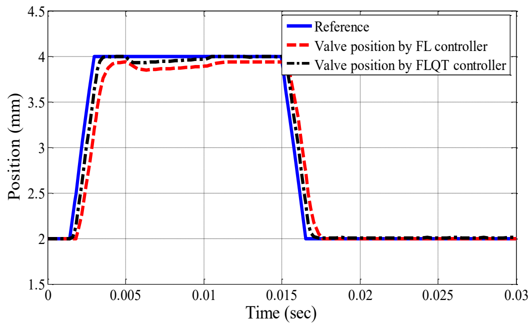

Fig. 6. Desired and valve positions for FL and FLQT controllers in presence of disturbance.

Figure 5 depicts the control signals of the proposed controllers in the form of motor voltage. Correspondingly, the obtained control signals for FL and FLQT are completely acceptable and could be applied to the actuator. It is notable that the motor voltage of the FLQT controller had fewer variations and was smoother compared to the control signals of the FL approach despite its higher final value.

In order to assess the capability of the proposed controllers for attenuating the disturbance effects, an output disturbance is considered in Eq. (30).

$$\begin{cases} d(t) = -20e^{-1000t} & 0.005 < t < 0.01 \\ d(t) = 0 & \text{else} \end{cases} \quad (30)$$

Figures 6 and 7 illustrate the desired valve position, EMV actuator positions, and control signals for both controllers in the presence of the disturbance. As is observed, the effect of the external disturbance on the sys-

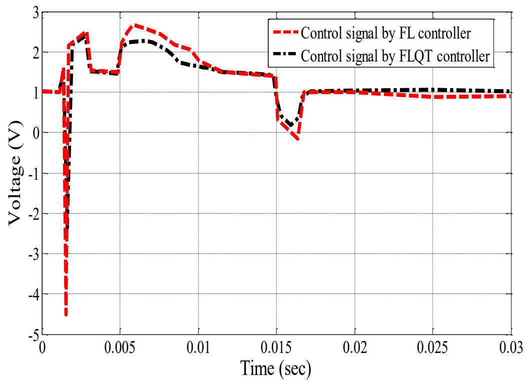

Fig. 7. Control efforts of FL and FLQT controllers in presence of disturbance.

tem performance is attenuated by the proposed controllers.

V. CONCLUSIONS

Camless engines provide more advantages compared to conventional crankshaft systems due to the elimination of the camshaft; such examples are the fewer moving parts, improved fuel efficiency, maximum torque and power, and reduced pollutant emissions. Fundamentally, the camshaft should be replaced by an actuating system, in which the electromechanical valve actuator is selected for this purpose in the present study. In order to overcome the challenges of the system, the FL and FL controller combined with finite time LQT method named as FLQT are designed for the EMV actuator. The simulation results demonstrated that the proposed controllers could realize the design requirement in term of soft landing and reference tracking. In addition, the optimal structure of FLQT resulted in zero steady-state error and faster valve displacement.

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