Modeling and Simulation of an Electromechanical Brake System

Farag Mahel Mohammed¹, Jamal A.-K. Mohammed², Haider Faisl³

Department of Electromechanical Engineering/ University of Technology / Baghdad-Iraq
¹ Email: drfaragmahel@yahoo.com, 50127@uotechnology.edu.iq
² Email: 130015@uotechnology.edu.iq
³ Email: head.face@hotmail.com

Abstract. The effect of brake pad wear on the electromechanical disc brake system response using MATLAB/Simulink software was investigated in this paper. The analysis of the pad wear involves building thermal model and electromechanical brake model, and calculating the required braking force at a constant and variable coefficient of friction between the pad and disc. A comparison between the electromechanical brake responses without wear and during wear at a constant and variable coefficient of friction was carried out. The results show that the coefficient of friction has a major role in calculating the required braking force, and cumulated wear during breaking. The required braking force starts higher in the case of using a variable coefficient of friction by about 0.06%. The variable coefficient of friction presents higher wear than the constant coefficient of friction by about 30%.

1. Introduction

Brakes are one of the most essential components and main safety system in the car. Regardless of other causes, 22% of the auto accidents are caused by the failure of brakes [1]. Brakes are mechanical or sometimes electrical devices or components that help to decelerate the vehicle and eventually stop the vehicle at a certain time and certain distance [2]. Both of the drum brakes and disc brakes are a friction-based braking system. Due to the problems of hydraulic brakes concerning the maintenance, the Electromechanical Brake (EMB) is drawing more attention as shown in figure 1. The EMB is a type of Brake-by-wire (BBW) where the driver’s brake command results in an electric signal that will be communicated via micro-controllers to the actuator [3].

![Figure 1. EMB](image)

It has many advantages over conventional brake such as; reducing mechanical parts, nature friendly, and less maintenance. The EMB mechanical part performs by transforming motor rotational motion...
into linear displacement by which it controls the braking force. The electrical part processes feedback signals transmitted to the control chip via sensors to ensure optimal response time. Friction brakes operate by converting the kinetic and potential energy into thermal energy or heat. The temperature can exceed 500 degrees, and 90% of the heat generated is transferred to the rotor before dissipating to the air [4] and the force can reach to 30KN [5]. Such amount of temperature and force during braking leads to wear over time causing the brake to fade. Brake pad wear is more to be considered because of its material. Before the pad is completely wearied, the gap between the pad and rotor becomes greater than when pads are new. Such clearance will affect the response time in the normal brake as well as the EMB.

Jaaskelainen (2009) [6] had calculated the heat-transfer coefficients analytically and numerically. It was shown that the empirical and numerical values of the heat transfer coefficients were close to each other in the air gap. Talati and Jalalifar (2009) [7] extracted the governing heat equations for the disk and pad. It was showed that there is a heat partition between the two components in sliding contact. Belhocine et al. (2014) [8] investigated the temperature distribution of a rotor disc during braking operation using finite element analysis (FEA) techniques. The temperature distribution on the brake disc was predicted and the critical temperature of the rotor was identified. Lee and Kang (2015) [9] had introduced a mathematical formula for variable friction coefficient computation. The effectiveness of the formula is verified experimentally.

Osunbor and Koya (2015) [10] determined the coefficient of friction and the wear characteristics of the materials. The wear properties of four brands of brake pad were evaluated and a pin-on-disc test rig was conducted. It was shown that the brake pads exhibited different wear rates as load increases. Verma (2016) [11] investigated two cases of wear for two friction materials. It showed there is an exponential relationship between the specific wear rate coefficient and the temperature of the disc for both pads materials. Kalhapure and Khairnar (2018) [1] summarized and analyzed the influence of operating parameters on the surface interface behavior of different brake materials. The study concluded that there was a rapid process of wear with increasing temperature and slow process wear with an increase in pressure. Soundhararajan (2019) [12] developed an algorithm to measure and estimate the brake gain on-line. Also, they presented a temperature model to estimate the disc temperature then the disc-pad interface friction coefficient. It was approved that an improved temperature estimator can improve the performance of the brake capability estimator.

In present study, a thermal model was built to estimate the temperature and the required braking force in an EMB system. The required braking force was calculated at a constant and variable coefficient of friction. The EMB is modeled using the principles used by Ludemann (2002) [3]. In which the inputs are the actuating current, acceleration, angular velocity, angular displacement, and linear (piston) displacement. The wear model is built using Archard's equation (1953) [14], where the inputs are the specific wear rate coefficient, pressure, and sliding distance.

In some literary work, the EMB system was built while in other literatures, the pad wear was calculated experimentally. Since the brake piston displacement is in millimeters and 1mm of displacement can generate force up to 10kN, the pad wear even in micro millimeter would affect the generated force. However, in this work, the pad wear model was built. The resulted wear was converted into negative force using the stiffness of the brake caliper to be subtracted from the required braking force to get the actual braking force.

2. The Methodology Consideration

2.1 Pad Wear

The pad wear is depending on the sliding speed, sliding time, temperature, and normal force acting upon it. The volume wear rate computed as [13]:

\[ \text{Volume Wear Rate} = \text{Specific Wear Rate} \times \text{Pressure} \times \text{Sliding Distance} \]
Where; $h$ is the thickness wear rate (m), $K$ is the specific wear rate coefficient (m$^2$/N), $F$ is the normal force acting on surfaces (N), and $S$ is the sliding distance (m). According to [11], the $K$ coefficient is mainly dependent on surface temperature and an exponential relationship with disc temperature ($T$) observed as shown in figure 2. An exponential relationship between pad wear and temperature can be derived through an exponential regression fit to this disc wear data.

$$A = 17.7e^{0.0087\cdot10^{-15}\left(\frac{m^2}{N}\right)}$$

2.2 Braking Force

The braking force is the generated normal force to the tangential friction force between the pad and disc surfaces that can bring a running vehicle to lower velocity or to stop via pressure applied on the disc by two pads, which is the amount of resistance to the kinetic energy of vehicle during braking. The rate of kinetic energy can be estimated as [15]:

$$\dot{E} = \frac{0.2M(V_i^2-V_f^2)}{t_b}$$

Where, $\dot{E}, M, V_i, V_f,$ and $t_b$ are the braking kinetic energy, vehicle mass, initial velocity, final velocity, and braking time. The 0.2 value is a result of multiplication of 0.5, 0.5, and 0.8 (or 80%) of vehicle weight considered on the front axle and the first 0.5 is for half axle while the second 0.5 is related to the kinetic energy equation. The power required to absorb the kinetic energy is [16]:

$$P = 2\mu \cdot p \cdot A_{pad} \cdot \Delta \omega \cdot r_e$$

Where $\omega$ is the change of angular velocity, $p$ is the pressure applied by a single pad, $\mu$ is the coefficient of friction between disc and pad, $r_e$ is the effective radius of the brake torque on the disc, and $A_{pad}$ is the contact surface area of the pad. Equating equations (3) and (4) and isolating the force yields:

$$F = \frac{0.1M(V_i^2-V_f^2)}{\mu \Delta \omega \cdot r_e \cdot t_b}$$

Assuming pressure constant, the effective radius is the mean radius between the inside and outside radii of the pad:

$$r_e = \frac{2(R_o^3-R_i^3)}{3(R_o^2-R_i^2)}$$
The friction coefficient is mainly dependent on the temperature and sliding velocity; therefore it varies upon them and can be calculated as follows [9]:

\[ \mu_d = 0.38 \left( 0.184 e^{-0.1V_d(t)} + 1 \right) \left( 0.105 e^{-0.0147T_d(t)} + 1 \right) \]  

(7)

Where, \( V_d(t) \) and \( T_d(t) \) are velocity and temperature of the disc at a given time.

\[ \dot{C}_p D_m \frac{dT}{dt} = \dot{Q}_{Din} - \dot{Q}_{Dout} \]  

(8)

Where, \( m_{disc}, C_p D, \dot{Q}_{Din}, \dot{Q}_{Dout} \) are the disc mass, the disk specific heat, the disc heat generation, and the heat dissipation, respectively.

The heat is generated due to kinetic energy and dissipated to air due to convection and radiation, but only 90.88% of it is absorbed by the disc as follows [4]:

\[ \dot{Q}_{Din} = 0.908 \* \dot{E} \]  

(9)

\[ \dot{Q}_{Dout} = \dot{Q}_{rad} - \dot{Q}_{conv} \]  

(10)

Where, \( \dot{Q}_{rad} \) and \( \dot{Q}_{conv} \) are the heat loss rate caused by radiation and convection, respectively. The specific heat capacity is dependent on temperature [8] as shown in figure 4.a.

\[ \text{(a) Specific heat versus temperature} \]  

\[ \text{(b) GCI emissivity versus temperature} \]  

Figure 3. Braking force, friction, and effective radius

Figure 4. Specific heat capacity as [8], Interpolating of Emissivity and Temperature [18]
A linear relationship between material specific heat and temperature can be derived using a linear regression fit. The equations obtained using Excel fit tool are shown as follows:

$$C_p = 0.262T + 459.4\left(\frac{W}{m.C}\right)$$  

(11)

Heat loss due to convection can be calculated using the equation presented as [17]:

$$\dot{Q}_{\text{conv}} = h_{\text{conv}} A_{\text{conv}} (T - T_{\text{amb}})$$  

(12)

Where, $h_{\text{conv}}$, $A_{\text{conv}}$, $T$, and $T_{\text{amb}}$ are the convection heat transfer, convection area, disc temperature, and ambient temperature, respectively.

Convection heat transfer is dependent on air speed and can be calculated as [6]:

$$h = \frac{(N_u d_2)}{k_{\text{air}}^0}$$  

(13)

$$N_u = C_1 R_e C_2 N_u P_r^{0.33}$$  

(14)

$$R_e = \frac{\rho \nu d_2}{\alpha_v}$$  

(15)

Where, $k_{\text{air}}$, $d_2$, and $P_r$ are the density, thermal conductivity, dynamic viscosity, and Prandtl Number of air. Air properties are required to be taken at 552°C under atmosphere pressure as shown in Table-1 [2]. $N_u$ and $R_e$ are the Nusselt Number and Reynold’s Number, respectively. The coefficients $C_1$ and $C_2$ are related to Reynold’s Number, and can be taken from Table-1 [6]:

| Temperature (°C)  | 552 | $Re$ | $C1$ | $C2$ |
|-------------------|-----|------|------|------|
| Density (kg/m$^3$) | 0.43 | 0.4 - 4 | 0.989 | 0.33 |
| Viscosity (kg/m-s) | 37.66×10$^{-6}$ | 4 - 40 | 0.911 | 0.385 |
| Prandtl Number     | 0.693 | 40 - 4000 | 0.683 | 0.466 |
| Specific heat (J/kg-k) | 1103.5 | 4000 - 40000 | 0.193 | 0.618 |
| Thermal conductivity | 0.059835 | 40000 -400000 | 0.027 | 0.805 |

Heat loss due to radiation can be calculated using the equation [17]:

$$\dot{Q}_{rad} = \varepsilon \sigma A_{rad} (T^4 - T_{amb}^4)$$  

(16)

The $\sigma$ is the emissivity and is the Stephan-Boltzmann constant: $(5.67 \times 10^{-8} \frac{W}{m.K^4})$.

Putting the radiation heat equation loss in the form of:

$$\dot{Q}_{\text{conv}} = h_{\text{rad}} A_{\text{rad}} (T - T_{\text{amb}})$$  

(17)

The radiation heat transfer becomes:

$$h_{\text{rad}} = \varepsilon \sigma (T^2 + T_{amb}^2)(T + T_{\text{amb}})$$  

(18)

It’s seen that radiation heat transfer is dependent on emissivity and temperature as shown in figure 2.b [18]. After interpolating of readings, the radiation heat transfer equation for the disc be described as:

$$h_{\text{rad}} = 1.25 \times 10^{-4} T^2 - 0.01 T + 5.1$$  

(19)
Sliding distance is defined as the distance that the disc travels across the pad surface along the effective radius on disc and can be obtained as follows:

\[ S = S_{dt} * \left( \frac{r_T}{r_e} \right) \]  
\[ (20) \]

Where \( S_{dt} \) is the stopping distance, \( r_T \) the tyre radius, and \( r_e \) the disc effective radius.

### 2.4 Motor Modeling

A Permanent Magnet Synchronous Motor (PMSM) is chosen for braking applications due to its higher torque, faster response, and ability of the EMB control unit to control both motor velocity and motor current accurately [19].

In the PMSM model, three-phase vectors (a-b-c) concerning a voltage or a current are transformed to rotor coordinates; the direct-quadrature (d-q) axis. Clarke's and Park's Transformation is used to build the mathematical model [21]. Park's transformation matrix and its inverse are used to transform the reference frame fixed on the rotor (d-q), to a stationary reference frame of \( \alpha \) and \( \beta \) and vice versa as shown in figure 3.

![Clarke's and Park's transformation between phases](image)

The overall transformation matrix and its inverse between the three-phase stationary frame and d-q reference frame are [21]:

\[
\begin{bmatrix}
X_d \\
X_q
\end{bmatrix} =
\begin{bmatrix}
\cos \theta_r & \cos(\theta_r - 120) & \cos(\theta_r + 120) \\
-\sin \theta_r & \sin(\theta_r - 120) & -\sin(\theta_r + 120)
\end{bmatrix}
\begin{bmatrix}
X_a \\
X_b \\
X_c
\end{bmatrix}
\]

Where, \( X \) represents flux, current, and voltage of phases. The stator voltage equations become:

\[
u_{sd} = R_s i_{sd} + L_{sd} \frac{di_{sd}}{dt} - \omega_r i_{sq} + \omega_r L_{sd} i_{sd} - \omega_L \psi_m
\]

\[ (21) \]

\[
u_{sq} = R_s i_{sq} + L_{sq} \frac{di_{sq}}{dt} - \omega_r i_{sd} + \omega_r \psi_m
\]

\[ (22) \]

Where, \( \omega_L \) is the angular velocity, \( R_s \) is the resistance of stator, \( L_{sd} \) and \( L_{sq} \) are d-q axis inductance of the stator, \( i_{sd} \) and \( i_{sq} \) are synchronous d-q axis current of the stator, \( \psi_m \) is the flux linkage. From equations (2) and (3), the torque equation can be expressed as:

\[
T_e = \frac{P_n}{2} \left( \frac{L_{sd} - L_{sq}}{2} \right) i_{sq} i_{sd} + \psi_m i_{sq}
\]

\[ (23) \]

The PMSM is designed with the same inductances, \( L_{sd} = L_{sq} \) and then \( i_{sd} \) does not affect the output, so, the torque equation is re-writing as:

\[
T_e = \frac{P_n}{2} \psi_m i_{sq}
\]

\[ (24) \]
Where, $T_e$ is the motor electrical (actual) torque and $P_n$ is the number of poles.

### 2.5 EMB Mathematical Model

Building an EMB mathematical model has some requirements which are as follows [20]:

1. Motor torque is a multiplication of motor current and torque constant.
2. Net torque is the result of torque balance around the motor axle considering motor torque and load torques (friction torque and clamping torque).
3. Acceleration is a division of net torque by total inertia.
4. Velocity is the integration of acceleration.
5. Piston displacement is a multiplication of motor position and effective gear ratio.
6. Clamping force is a multiplication of piston displacement and caliper stiffness.
7. Clamping torque is a multiplication of clamping force and effective gear ratio.

The equation of motion for the EMB can be expressed as [5]:

$$\ddot{T}_e = T_m - T_{load} = T_m - T_f - T_{clamp} \quad (25)$$

$$J\ddot{\theta} = T_m - T_f - F_{clamp}K_{gear} \quad (26)$$

$T_m$ is equal to the motor electrical torque $T_e$ and for the motor with 2 poles it becomes:

$$T_e = k_tI_sq \quad (27)$$

Where, $k_e$ is the motor torque constant, $\psi_m$ represents the back EMF constant and $\psi_m = 2/3k_e$. $T_f$ is the friction torque and can be regarded as a kinetic coulomb friction model, whose sign is changed depending on the direction of movement, positive for forward and negative for backward [22]:

$$T_f = D\dot{\theta} \cdot \text{sign}(\dot{\theta}) \quad (28)$$

Where, $D$ is the viscous friction coefficient ($N.m/rad$).

Clamping torque $T_{clamp}$ is dependent on clamp force and effective gear ratio $K_{gear}$. $K_{gear}$ can be calculated as [23]:

$$K_{gear} = \frac{1}{GR_{tot}2\pi} \quad (29)$$

Where, $P_s$ is the ball screw pitch and $GR_{tot}$ is the total gear reduction ratio.

The clamping force is a result of the relationship between piston displacement and caliper stiffness. Piston displacement is calculated by:

$$X = K_{gear}\cdot\theta \quad (30)$$

Caliper stiffness is a non-linear curve, which is a characteristic curve that links displacement to clamp force [24]. Different characteristic curves are presented in [5]. In this study, the following polynomial curve will be used for its smoothness and ability to respond faster.

$$F_{cl} = \begin{cases} 
-7.23x^3 + 33.7x^2 - 3.97x \\
0.1295x 
\end{cases} \quad x > 0.125$$

$$F_{cl} = \begin{cases} 
\text{otherwise} 
\end{cases} \quad \text{otherwise} \quad (31)$$

The stiffness curve is approximated for force range of (0-30)kN and displacement range of (0-3) mm.
2.6 Pulse Width Modulation and Inverter

Most controller units use Pulse Width Modulation (PWM) to control the speed of the DC motor. The PWM and Inverter equations that are used to generate a sinusoidal PWM and to invert voltages are as follow [21]:

\[
S_a = \begin{cases} 
1 & \text{if } U_{a,ref} \geq \text{triangular wave} \\
0 & \text{if } U_{a,ref} < \text{triangular wave}
\end{cases}
\]

\[
S_b = \begin{cases} 
1 & \text{if } U_{b,ref} \geq \text{triangular wave} \\
0 & \text{if } U_{b,ref} < \text{triangular wave}
\end{cases}
\]

\[
S_c = \begin{cases} 
1 & \text{if } U_{c,ref} \geq \text{triangular wave} \\
0 & \text{if } U_{c,ref} < \text{triangular wave}
\end{cases}
\]

The inverter equations are:

\[
\begin{align*}
    u_a &= V_{Am} = V_{AM} - V_{MM} \\
    u_b &= V_{Bm} = V_{BM} - V_{MM} \\
    u_c &= V_{Cm} = V_{CM} - V_{MM}
\end{align*}
\]

2.7 Controllers

Three cascade controllers represented by current, speed and force controller are used in the EMB system model as shown in figure 7. The current and speed controllers were of PI controller type with back calculation anti-windup as shown in figure 8, while the force controller is of only P type (Proportional) controller as shown in figure 9, also a feed forward block is used to enhance the set-point of current and speed controller. Tuning of the controller’s gains was done by using trial and error methods. Gains, motor parameters used in the EMB model are shown in tables 2 and 3.

![Figure 6. Blocks for generation Sinusoidal PWM](image)

![Figure 7. EMB Cascade Controllers with Feed-forward](image)
3. Results and Discussion
Models were simulated at a period of 30 seconds including three brake applications of one second. The vehicle speed and heat loss due to convection and radiation are shown in figure 10. As can be seen, the heat loss due to convection is increasing during acceleration, while the heat loss due to radiation increasing during deceleration, also the rate of heat loss due to convection is much higher than due to radiation. This because the heat loss due radiation is mainly depending on the airflow caused by the vehicle acceleration and its duration, while the heat loss due to radiation mainly depends on its duration and the difference in the temperature between the air and the disc. In our simulation model, the duration of acceleration is assumed longer. The highest difference in the temperature reached, was only about 430°C.

Figure 10. Speed vs. heat loss (convection and radiation)
Figure 11 represent the vehicle speed concerning the net heat and temperate at the disc. It's obvious both of net heat and temperature have the same path as the temperature is proportional to heat. Also, both heat and temperature rise during deceleration at the braking period and decrease otherwise. Due to that the temperature is calculated using the net heat, mass, and specific heat of the disc as in Eq.8. While the biggest variable in this equation is the heat, then the temperature is mainly varying upon the net heat.

The relationship between force, deceleration, and coefficient of friction are shown in figure 12. At a constant coefficient of friction of 0.4, the required braking force also constant, while at a variable coefficient of friction (range between 0.38 and 0.45), the force follows a curvy path by starting high, then decreasing gradually.

Figure 13 shows the wear model result and how wear is related to temperature and force. It is noted from the figure that there is a difference in wear. Also, it is shown that wear is greater by 0.002mm when force is calculated using a variable coefficient of friction. Because the variable coefficient of friction starts higher at 0.45 with a longer curvy path than for the constant coefficient of friction, which causes the force to start higher also with a longer path and then causes more wear in a shorter time.
The EMB model result in the first case of braking is shown in figure 14, where the coefficient of friction is considered as constant value of (0.4). The green line represents the force signal without wear, while the blue line represents the signal during wear. The responses are marked with dotted lines, respectively. As can be seen, there is a difference of 200 Newton for wear of $6 \times 10^{-3}$ millimeters.

![Figure 14. EMB response, before and during wear (at constant coefficient of friction)](image)

While, figure 15 shows the second case, where the coefficient of friction is assumed variable. The red line represents the force signal without wear, while the brown line represents the signal during wear. The responses are marked with dotted lines. It shows that the difference in force is similar as in case 1 despite wear is higher.

![Figure 15. EMB response, before and during wear (at variable coefficient of friction)](image)

In figure 16, the two cases are shown together, force calculated using the variable coefficient of friction starts at $25.5 \times 10^3$ and smoothly drops to $22 \times 10^3$ Newton, while force calculated using the constant coefficient of friction is $24 \times 10^3$ Newton.

![Figure 16. EMB responses (at all cases)](image)
4. Conclusions
The effect of brake pad wear on the electromechanical disc brake system response was investigated using MATLAB/Simulink software. A comparison between the electromechanical brake responses without wear and during wear at a constant and variable coefficient of friction has been carried out. The following conclusions are drawn:

1. The coefficient of friction has a major role in calculating the required braking force, and cumulated wear during breaking.
2. The required braking force starts higher in case of using a variable coefficient of friction by about 0.06%.
3. The variable coefficient of friction yields higher wear than the constant coefficient of friction by about 30%.
4. The effect of wear is proportional to the force applied.

5. References
[1] V. A. Kalhapure and D. H. P. Khairnar, “Wear mechanism and modelling for automotive brakes with influence of pressure, temperature and sliding velocity”, 2018.
[2] P. N. Amrish, “Computer aided design and analysis of disc brake rotors,” Advances in Automobile Engineering, vol. 05, no. 02, 2016.
[3] J. Ludemann, “Heterogeneous and hybrid control with application in automotive systems,” Ph.D. dissertation, ProQuest Dissertations & Theses, 2002.
[4] H. Mazidi, S. Jalalifar, S. Jalalifar, and J. Chakhou, “Mathematical modeling of heat conduction in a disk brake system during braking”, Asian Journal of Applied Sciences, vol. 4, no. 2, 2011.
[5] C.L.J. Line, “Modelling and control of an automotive electromechanical brake”, Ph.D. dissertation, 2007.
[6] M. Jaaskelainen, “Determination of coefficients of thermal convection in a high-speed electrical machine”, Helsinki University Technology, Faculty of Electronics, Communications and Automation Department of Electrical Engineering, 2009.
[7] F. Talati and S. Jalalifar, “Analysis of heat conduction in a disk brake system”, Heat and Mass Transfer, vol. 45, no. 8, pp. 1047–1059, Jan. 2009.
[8] A. Belhocine, C.D. Cho, M. Noubry, Y.B. Yi, A. Bakar, and A. Rahim, “Thermal analysis of both ventilated and full disc brake rotors with frictional heat generation”, 2014.
[9] N.-J. Lee and C.-G. Kang, “The effect of a variable disc pad friction coefficient for the mechanical brake system of a railway vehicle”, PLOS ONE, vol. 10, no. 8, p. e0135459, Aug. 2015.
[10] R. S. Fono-Tamo, O. O. Osunbor, and O. A. Koya, “Weibull approach to brake pad wear analysis in the nigerian market”, Friction, vol. 3, no. 3, pp. 228–233, Jul. 2015.
[11] P. C. Verma, “Automotive brake materials: Characterization of wear products and relevant mechanisms at high temperature”, Ph.D. dissertation, University of Trento, 2016.
[12] R. Soundhararajan and P. D. K. Chandrashekar, “Online monitoring of brake capability for heavy vehicles”, Master’s thesis, 2019.
[13] S.-K. Baek, H.-K. Oh, S.-W. Kim, and S.-I. Seo, “A clamping force performance evaluation of the electro mechanical brake using pmsm”, Energies, vol. 11, no. 11, p. 2876, 2018.
[14] J. F. Archard, “Contact and rubbing of flat surfaces”, Journal of Applied Physics, vol. 24, no. 8, pp. 981–988, Aug. 1953.
[15] A. Stephens, “Aerodynamic cooling of automotive disc brakes”, 2006.
[16] K. Bill and B. J. Breuer, “Brake technology handbook”, SAE Technical Paper, Tech. Rep., 2008.
[17] F. P. Incropera, A. S. Lavine, T. L. Bergman, and D. P. DeWitt, Fundamentals of heat and mass transfer. Wiley, 2017.
[18] V. Schweiz, “Table of emissivity of various surfaces”, Mikron Instrument Company Inc, Tech. Rep., Jun. 2014.
[19] L. Prokop, M. Stulrajter, and R. Radhostem, “3-phase pmsm vector control using the pxs20 and tower system”, Freescale Semiconductor Application Note, AN4537, 2012.
[20] C. Maron, T. Dieckmann, S. Hauck, and H. Prinzler, “Electromechanical brake system: Actuator control development system”, SAE Technical Paper, Tech. Rep., 1997.
[21] W. Han, “Simulation model development of electric motor and controller”, Master’s thesis, 2017.
[22] R. Schwarz, R. Isermann, J. Böhm, J. Nell, and P. Rieth, “Clamping force estimation for a brake-bywire actuator”, SAE Technical Paper, Tech. Rep., 1999.
[23] G. Park, S. Choi, and D. Hyun, “Clamping force estimation based on hysteresis modeling for electromechanical brakes”, Int. Journal of Automotive Technology, vol. 18, no. 5, pp. 883–890, 2017.
[24] R. Schwarz, J. Isermann, and P. Rieth, “Modeling and control of an electromechanical disk brake”, SAE Technical Paper, Tech. Rep., 1998.