Simulated Braking Performance Comparison of an Electric Drum Brake and a Hydraulic Drum Brake Systems

Kittirattanachai A, and Watechagit S*

1 Department of Mechanical Engineering, Faculty of Engineering, Mahidol University, Nakornpathom, 73170, Thailand

* Corresponding Author: sarawoot.wat@mahidol.ac.th

Abstract. The most important safety equipment for the operation of a vehicle is the brake equipment. This research attempts to develop an all-electric braking system that can be used in an electric vehicle. Since the drum brake requires lower actuation input than the disk brake in order to deliver the same amount of braking force. It seems logical to be used in the electric vehicle as far as minimizing energy usage is concerned. The proposed electric drum brake system conceptually is not new as it has been used as a trailer brake for a tractor-trailer vehicle. Adopting the trailer brake for a passenger vehicle requires proper downsizing and different braking control scheme. This paper presents the modelling and simulation results of the proposed electric drum brake. The main interest is its braking performance which can be investigated by the brake force profile or the brake force characteristic. The results are compared with the braking performance of the conventional hydraulic drum brake system which is using validated data from the literature. The simulation results show that the braking force of the electric drum brake system is two times higher than that of the hydraulic drum brake system subjected to the same actuation force. The results from this paper will be valuable for the development of the controller in the future.

Keywords: Electromagnetic trailer brake, Electric drum brake, Brake performance and Mathematic models.

1. Introduction

The working principle of a drum brake has not been changed since its first commercial use in 1900. It has still been commonly used in commercial vehicles. One to of the main benefits of the drum brake is its high brake factor, which is defined by the ratio of the output braking force to the application force that activates the braking action. By comparing with other types of vehicle brake systems, e.g. a disk brake, subjected to the same application force, the drum brake generally produces higher braking force to the vehicle’s wheel [1]. For the development of an electric braking system for an electric vehicle, and since the electric vehicle has limited resource for its operation, the drum brake seems to be a good reference system as it requires less actuation effort for the same braking force output. This research, therefore, investigates the development of the electric brake from the conventional drum brake system.

Reviewing pieces of literature related to the electric brake found that there have already been a few electric brake designs, including the magnetorheological brake [2-3], the electronic wedge brake [4-5] and the electromagnetic trailer brake. While the previous/former two are under their development phase,
the latter has been used widely in the commercial trailer truck. Principally, the electromagnetic trailer brake is the drum brake with its application force or the brake actuation action which is produced by an electromagnetic mechanism. In other words, without the electromagnetic parts, this trailer brake would look similar to the general drum brake. It is used as an auxiliary brake rather than the main or the master brake. The adoption of the trailer brake to be used in a passenger vehicle is not as straightforward, especially regarding the controller design. Unfortunately, the detailed design and development of the electromagnetic trailer brake cannot be found in previous studies such as brake characteristic profile and brake mechanism theory.

In this paper, the mathematical models of both conventional hydraulic brake and the electromagnetic brake are hence developed. Only the conventional hydraulic drum brake model can be validated using data from the literature including brake characteristic and parameter of hydraulic drum brake for developing mathematic models. The resulted model will be used as a reference for the model of the electromagnetic trailer brake. To investigate the braking performance of the electromagnetic brake, brake force from both braking systems are simulated and compared with brake force profile. The results can help us to better understand the physics behind the operation of the electromagnetic brake. The developed model will also be used for controller design in the near future.

2. The mathematic model of the hydraulic drum brake and electromagnetic trailer brake systems

2.1 Hydraulic drum brake system
The hydraulic drum brake system works by the interaction of five parts. The pedal force from the driver actuates the braking action. The vacuum booster amplifies the pedal force. The master cylinder transmits force to the wheel cylinder via a hydraulic system. And the brake shoes which generates the braking force to the wheel. The schematic diagram of the hydraulic brake system and the flow of the braking power are shown in Figures (1) and (2).

![Figure 1. Hydraulic drum brake system [6].](image1)

![Figure 2. The transmissible force in the hydraulic drum brake system.](image2)

2.1.1. Model of the brake pedal
As can be seen in Figure 1, the pedal force is the transmitted force from the brake pedal, generated by the driver, to actuate the vacuum booster. The pedal force is calculated from taking moment at the clevis pin [7] as shown in equation (1).

\[ F_n = F_d \frac{l_1}{l_2} \]  

(1)

Here, \( F_n \) is the pedal force (N), \( F_d \) is the force input from the driver at the brake pedal (N), \( l_1 \) is the perpendicular distance from the pedal pivot to the center of input brake pedal (0.36 m) and \( l_2 \) is the perpendicular distance from the pedal to the clevis pin (0.04 m).

2.1.2. Model of the vacuum booster
In Figure 2, the pedal force (\( F_n \)) actuates the vacuum booster piston and vacuum diaphragm, which increases the force to push the master cylinder piston. The total force at the master cylinder piston, the
so called $F_{pedal}$ which is the summation of the force from the pedal and the vacuum booster as shown in equation (2) [8].

$$F_n + \left( \frac{\pi}{4} D^2 - \frac{\pi}{4} d^2 \right) \cdot \left( \text{atm pressure} - \text{vacuum pressure} \right) = F_{pedal} \quad (2)$$

Here, $D$ is the diameter of the vacuum diaphragm (m), $d$ is the diameter of vacuum booster piston, atm pressure is $101,315 \left( \frac{N}{m^2} \right)$ and the vacuum pressure is $55,000 \left( \frac{N}{m^2} \right)$. The vacuum booster is connected with the master cylinder.

2.1.3. **Model of the master cylinder**

The vacuum booster is connected to the master cylinder, which transmits the vacuum force to actuate the hydraulic pressure in the master cylinder. The pressure in the master cylinder ($P_{mcp}$) is calculated by considering all the forces acting across the master cylinder’s piston as described by equation (3). This takes into account also the spring force and seals-friction force at the master cylinder’s piston [9].

$$P_{mcp} = \frac{F_{pedal} - F_{spring0} - K_{spring}(x_1 - x_2) - F_{seal}}{A_{mc}} \quad (3)$$

Here, $x_1$ is the piston displacement in the primary line (m), $x_2$ is the piston displacement in the second line. Assuming here $x_1 - x_2$ is 0.04 [1]. $F_{spring0}$ is the master cylinder spring pre-tension (138 N). $F_{seal}$ is the seal friction force (80 N). $A_{mc}$ is the master cylinder piston area (0.000491 m$^2$) and $K_{spring}$ is the master cylinder spring constant (175 $\frac{N}{m}$).

2.1.4. **Model of the drum brake**

The drum brake model takes the force from the brake shoe [10] and calculates brake torque and brake force. Forces and dimensions inside the drum brake are shown in Figure 3.

Figure 3 illustrates the schematic diagram of the rear right wheel of a vehicle. The pressure from the master cylinder actuates the wheel cylinder, which then creates the actuation force ($F$). This is described by equation (4). This actuation force activates brake force at the leading and trailing shoes.

$$F = P_{mcp} \cdot A_{wc} \quad (4)$$
Here, $A_{wc}$ is the wheel cylinder piston area (0.00045238 m$^2$). The normal force ($dN$) occurs at any angle $\theta$ from the hinge pin, which moves brake lining to touch midpoint of wheel hub. The proportional of pressure and maximum pressure are substituted in the magnitude of the pressure formula, which is described by equation (5).

$$dN = \frac{p_0 b \sin \theta}{\sin \theta_a} d\theta$$

(5)

$p_0$ is the maximum pressure at an angle $\theta_a$ (1.5708 rad) of the right-hand shoe, which can be found by equation (6). $b$ is the width of the shoe (0.05 m) and $r$ is the radius of the drum (0.160 m).

$$\theta_a = 1.5708 \text{ (rad)} = \frac{\theta_1 + \theta_2}{2}$$

(6)

By applying the condition of static equilibrium, the maximum pressure of the right-hand shoe ($p_0$) can be calculated by taking clockwise ($M_f$) and anticlockwise ($M_N$) force moment at point A. From Figure 3 one can see that, each force vector has components in the x-axis and the y-axis term, which includes force in wheel cylinder ($F$), spring force ($Kx_1$), friction force ($f \, dN$) and normal force ($dN$). These forces are used for the calculation of clockwise ($M_f$) and anticlockwise ($M_N$) moment. Hence, $p_0$ can then be written as shown in equation (8). Here it should be noted that the integral terms are the averaged friction areas.

$$F = \frac{M_N - M_f}{c}$$

(7)

$$p_0 = \frac{Fc + (Kx_1 \cos(\theta_5))y - (Kx_1 \sin(\theta_5))x_3}{br(\int_{\theta_1}\theta^2 \sin \theta d\theta - \int_{\theta_1} \theta \sin \theta (r - a \cos \theta) d\theta)}$$

(8)

Based on the above equation, $c$ is the distance between the apply force ($F$) and the hinge (0.243 m). $K$ is the spring constant of the leading shoe and trailing shoe (3.394.85 N/m). $x_1$ is the right-hand shoe’s spring extending range (0.01 m). $\theta_5$ is the directional angle of the spring force (0.05236 rad), $y$ is the length of the spring to the hinge (0.125 m), $x_3$ is the point of spring to the hinge (0.07 m), $a$ is the distance between the drum brake center and the hinge (0.120 m). And $f$ is the friction coefficient between the lining and the drum surface (0.48).

The brake torque of the right-hand shoe ($\tau_{right}$) is represented by equation (9), which is the sum of the friction force $f \, dN$ times the radius of the drum.

$$\tau_{right} = f p_0 br^2 (\cos \theta_4 - \cos \theta_2)$$

(9)

The maximum pressure of the left-hand shoe ($p_a$) can be developed by similar fashion and can be described by a similar equation as (8). In this case, the brake torque of the left-hand shoe ($\tau_{left}$) is then described by a similar equation as (9). Furthermore, the total braking force can be described by equation (10).

$$Total \ braking \ force = \frac{\tau_{right} + \tau_{left}}{r}$$

(10)

2.2 Electric drum brake system

The mechanism of the input force applying at the brake shoe is different from the case of the hydraulic drum brake as it is now generated by the electromagnetic subsystem. The magnet sticks to the drum surface by the application of electric current from the trailer brake control. As the drum moves, the lever hence moves which causes the shoes to push against the drum [12]. The schematic of the electric drum brake is shown in Figure 4.
2.2.1 Model of electric drum brake

As shown in Figure 4, the electromagnetic mechanism is the schematic with the dark-black line, overlaid onto the schematic of the hydraulic drum brake. The magnet system consists of the electric circuit subsystem and the electromagnetic subsystem. The latter one produces the actuation force to push the shoe. The electric circuit subsystem can be formulated by equation (11). This electric circuit generates a magnetic flux. The electromagnetic circuit subsystem generates a magnetic suction force as shown in equations (12) and (13)[13].

\[
U = iR + N \frac{d\phi}{dt}
\]  
(11)

\[
Ni = \frac{\phi l_m}{\mu_0 S_0}
\]  
(12)

\[
F_m = \frac{\phi^2}{\mu_0 S_0}
\]  
(13)

\(U\) is voltage (Volt). \(i\) is the coil current (Ampere). \(R\) is the resistance in the coil (3 ohms). \(N\) is the coil number of turn. \(\phi\) is magnetic flux (Wb). \(l_m\) is the effective length of the magnetic conductor (0.003 m). \(\mu_0\) is the permeability of vacuum (\(4\pi \times 10^{-7}\)) and \(S_0\) is the cross-section area of the air gap (0.0003 m²). The friction force between the magnet and the wheel hub (\(f_m\)) is represented by equation (14) [13].

\[
f_m = \frac{\mu_m \phi^2}{\mu_0 S_0}
\]  
(14)

Here, \(\mu_m\) is the friction coefficient between the magnet and the wheel hub assuming value 0.48.

The force that generates the magnet movement is the effective driving force, \(F_{\text{eff}}\), which takes into account the maximum driving force, rolling resistance force and aerodynamic force in the vehicle model. The actuation force that pushes the brake shoe (\(F\)) can be found by considering the moment by the effective force (\(F_{\text{eff}}\)), the friction force between the magnet and the drum surface (\(f_m\)), and force pushed brake shoe (\(F\)), around point B in Figure 4. The result shows in equation (15).

\[
F = \left\lfloor \frac{-(F_{\text{eff}} + f_m) \times x_4}{c - x_4 - x_5} \right\rfloor
\]  
(15)

\(x_4\) is the distance from point of spring to the magnet (0.02648 m) and \(x_5\) is the distance between the magnet and the hinge (0.005 m). \(F\) from equation (15) is used to calculate the brake force and the brake torque in a similar fashion as shown in section 2.1.4. In particular, \(F\) in equation (15) will be substituted in equation (7) and the term \((-F_{\text{eff}} x_5 - f_m x_5)\) will be added in the numerator of equation (8).

3. The validation of the hydraulic drum brake

The hydraulic brake model is validated by comparing the contact force (\(N\)) of the leading shoe with the data from the literature [11]. The input in the model is the actuation force \(F\). The model then generates the normal force \(F_{\text{nd}}\) and hence the contact force. Table 1 shows the data from hydraulic drum brake testing obtained from the literature [11].
3.1 Validation results of the hydraulic drum brake

Table 2 and Figure 5 show the simulation results as compared to the brake testing data from the literature.

Table 1. The constraint parameter of validation.

| Cases | F (N) | \( F_d \) (N) | \( f \) | Contact force (N) |
|-------|-------|---------------|-------|-------------------|
| 1     | 4,119 | 365.30        | 0.48  | 32,656            |
| 2     | 4,217 | 377.10        | 0.48  | 32,754            |
| 3     | 4,315 | 388.90        | 0.50  | 32,852            |
| 4     | 4,413 | 400.72        | 0.50  | 32,950            |
| 5     | 4,511 | 412.60        | 0.50  | 33,048            |
| 6     | 4,609 | 424.36        | 0.50  | 33,146            |
| 7     | 4,707 | 436.18        | 0.51  | 33,244            |
| 8     | 4,805 | 448.00        | 0.50  | 33,342            |
| 9     | 4,903 | 459.81        | 0.48  | 33,440            |

Table 2. The comparison of simulation results.

| Cases | Contact force (N) from literature | Contact force (N) from simulation | % error |
|-------|-----------------------------------|-----------------------------------|---------|
| 1     | 32,656                            | 32,658                            | 0.006   |
| 2     | 32,754                            | 32,755                            | 0.003   |
| 3     | 32,852                            | 36,457                            | 9.888   |
| 4     | 32,950                            | 36,467                            | 9.644   |
| 5     | 33,048                            | 36,488                            | 9.427   |
| 6     | 33,146                            | 36,509                            | 9.211   |
| 7     | 33,244                            | 38,420                            | 13.472  |
| 8     | 33,342                            | 36,581                            | 8.854   |
| 9     | 33,440                            | 33,522                            | 0.244   |

Figure 5. Contact force vs friction coefficient.

Based on the results shown, the contact force calculated by the hydraulic model deviates from the testing data by 6.75% on average. The error is shown here mainly due to the uncertainty of the shoe’s geometry, i.e. \( \theta_1 \) and \( \theta_2 \), as well as the non-uniform contacting surface between the wheel hub and the shoe. The error though seems to be much, but, when considering the general magnitude of the brake force in the real application, 5%-10% brake force difference does not produce much difference in terms of braking performance, i.e. stopping distance. Therefore, the developed hydraulic model is then considered acceptable to be used as a reference model for the electric drum brake development here.
4. Simulation result and analysis of hydraulic drum brake and electric drum brake
The simulated actuation force that pushes the brake shoe from the hydraulic brake and the electric drum brake are compared in this section. The simulation is setup to brake the vehicle with its mass of 2,400 kg, and its initial speeds of 14, 25, and 36 m/s. Using equations (7), (8), (11), (12), (13), (14) and (15) the actuation force of the electric drum brake can be calculated.

4.1 The brake actuation force simulation
The simulation results showing the brake actuation force for both hydraulic drum brake and electric drum brake for each initial vehicle speeds are shown in Figure 6.

![Figure 6. Force push shoe of hydraulic drum brake vs electric trailer brake](image)

From figure 6, the response time of the hydraulic drum brake performs around 0.01 second. This is because the model does not take into account the dynamic of the hydraulic system during the filling phase. Therefore, the force quickly increases to its steady state. In other words, it is assumed that the hydraulic fluid is filled in the system. Hence the wheel cylinder can be actuated instantaneously.

The response times of the electric drum brake is around 0.9 second. The steady state actuation forces in this case are 4,118.08, 4,074.46, 4,024.68 N for the vehicle speed 14, 25, 36 m/s, respectively. Higher actuation force is produced at low vehicle speed because the dry road friction coefficient at low speed is higher than the one that is of the high vehicle speed. This affects the maximum driving force for effective force \(F_{eff}\) input in equation (15).
4.2 The braking force simulation

![The braking force simulation](image)

Figure 7. The braking force of hydraulic drum brake vs electric drum brake

Figure 7 shows the simulated braking force as a result of the actuation force from section 4.1. The braking force from the hydraulic drum brake is 18,283 N at all vehicle speed. Though, the braking force from the electric drum brake is 45,032.99, 44,867.59, 44,678.83 N at a vehicle speed of 14, 25, 36 m/s respectively. These are higher than the hydraulic drum brake by 146.31, 145.41, 144.37 % respectively. The electric drum brake gains additional brake force from the frictional force between the magnet and the drum or the wheel hub \((f_m)\). The magnet, which is powered by the electric circuit, tries to stick itself with the wheel hub, hence retards the wheel vehicle speed.

4.3 The braking performance discussion

While there are many ways to compare the brake performance of one braking system to another, due to limited resources and information of the electric drum brake from the literature, the braking force from the hydraulic drum brake and the electric drum brake are considered. From the simulation results, the braking force of the electric drum brake generates brake force two times more than that is of the hydraulic drum brake under the same input actuation force. In reality, the benefit of having higher braking force will have to be investigated further as the vehicle’s wheel can be locked any time when the braking force is higher than the friction force between the tire and the road. Figure 7 shows the friction force between the tire and the road for the simulated case, which is 20,012.40 N.

5. Conclusions

This research investigates the development of the electric drum brake from the conventional drum brake system. The main reason is that the drum brake has a higher brake factor than the disk brake which seems appropriate for the electric vehicle as far as minimizing the energy usage is concerned. The development adopted from the already existing electromagnetic trailer brake system is commonly found in the commercial trailer truck. The model of both hydraulic drum brake and the electric drum brake are developed in this paper. The hydraulic drum brake model is validated using the brake testing data from the literature. The validation results show that the model can simulate the brake force with acceptable accuracy and hence it can be used as a reference for the electric drum brake development.

The simulation of the actuation force that pushes the brake shoe and the braking force at the wheel of both hydraulic drum brake and electric drum brake are shown in this paper. While the actuation force
for both braking systems are about the same, the resulted braking force at the wheel for the case of electric drum brake is about two times higher than that is of the hydraulic drum brake. This is due to the fact that the electric drum brake gains additional braking effort from the frictional force acting between the magnet and the wheel hub. Moreover, the simulation does not take into account the wheel locking situation. Therefore, in reality, the benefit of having a higher braking force will have to be further investigated. The developed model will also be further refined for controller design and implementation.

References

[1] Day, A. J. (2014). Braking of road vehicles. Butterworth-Heinemann.
[2] Jerzy, B., & Mateusz, B. J. (2013). Influence of Temperature and Rotational Speed on the Properties of Magnetorheological Brake. In Design and Modeling of Mechanical Systems (pp. 143-149). Springer, Berlin, Heidelberg.
[3] Zheng, J., Li, Y., Wang, J., Shiju, E., & Li, X. (2018). Accelerated thermal aging of grease-based magnetorheological fluids and their lifetime prediction. Materials Research Express, 5(8), 085702.
[4] Ahmad, F., Mazlan, S. A., Zamzuri, H., Hudha, K., & Jamaluddin, H. (2015). Study on the potential application of electronic wedge brake for vehicle brake system. International Journal of Modelling, Identification and Control, 23(4), 306-315.
[5] Aparow, V. R., Hudha, K., Ahmad, F., & Jamaluddin, H. (2014). Development of Antilock Braking System using Electronic Wedge Brake Model. Journal of Mechanical Engineering and Technology (JMET), 6(1), 37-63.
[6] Hydraulic drum brake system, accessed 27 February 2019, https://www.freepik.com/premium-vector/hydraulic-brake-system_2195999.htm
[7] Ivan, F., Boran, P., & Davor, P. (2016). A Detailed Model for Simulation of Hydrodynamic Processes in Hydraulic Braking System with ABS on Motor Vehicles. Journal of Scientific Research and Reports, 1-12.
[8] Gajjar VY, Jaiveshkumar, Gandhi D 2011 International Journal of Environment, Agriculture and Biotechnology vol 2 p 301
[9] Geromel, N. (2014). Modelling and control of the braking system of the electric Polaris Ranger all-terrain-vehicle.
[10] Shigley, J. E. (2011). Shigley’s mechanical engineering design. Tata McGraw-Hill Education.
[11] Khairnar, H. P., Phalle, V. M., & Mantha, S. S. (2015). Estimation of automotive brake drum-shoe interface friction coefficient under varying conditions of longitudinal forces using Simulink. Friction, 3(3), 214-227.
[12] Braking system-electric, USA, accessed 19 September 2018, https://saubermfg.com/wp-content/uploads/2013/02/electric_brake_system_manual.pdf.
[13] Li, S., Guo, P., Jiang, W., Ding, H., & Yu, D. (2015, July). Research on response characteristics and parameters optimization of high-speed solenoid valve. In 2015 34th Chinese Control Conference (CCC) (pp. 2327-2332). IEEE.