MODELING AND SIMULATION OF AUTOMOTIVE HYDRAULIC BRAKE SYSTEMS USING SIMCENTER AMESIM

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Abstract - This study develops a model of and simulates a hydraulic braking system using Simcenter Amesim to analyze the dynamic characteristics of the hydraulic system and evaluate the effects of several structural parameters. The simulation results show that the model can represent system operation, with a maximum pressure of 88 bar, a pressure transmission delay of about 0.003 s, and front axle brake pad clamping forces approximately 2.4 times greater than those at the rear axle. Transient phenomena were observed during emergency braking. Increasing the viscous friction coefficient led to a pressure delay increase of about 0.032 s, while increasing the return spring stiffness reduced the pressure by 2 bar. When the brake pad clearance increased by 0.2 mm, the pressure decreased by about 23%. These findings provide a basis for parameter selection and optimization to improve braking performance and system stability.

Key words - Hydraulic braking system; braking dynamics; Simcenter Amesim; system simulation; lumped parameter modeling

1. Introduction

Hydraulic braking systems are widely used due to their advantages of fast response and high efficiency. However, the design and optimization of these systems face many difficulties because of their strong nonlinearity [1].

The emergence and application of modern computational methods, such as the finite element method (FEM) and computational fluid dynamics (CFD), have provided a foundation for design and simulation [2]. Today, simulation and analysis tools have become an essential part of the product development process. Vu Hai Quan et al. proposed an intelligent braking system model based on Matlab/Simulink [3]. The results showed that the fuzzy logic control algorithm improved braking efficiency and shortened the vehicle stopping distance. However, the research model was mainly based on a quarter-car model; therefore, it did not fully reflect the dynamics of the entire system during braking [3]. Ivan Filipovic et al. developed a detailed hydrodynamic model that considered the compressibility of brake fluid and pressure wave propagation in the system [4]. Although the model achieved high accuracy, the process of formulating equations and performing calculations using the fourth-order Runge–Kutta method was relatively complex, making rapid application in industrial design difficult [4]. Sylvester Emeka Abonyi et al. investigated P, PI, and PID controllers based on Matlab/Simulink to maintain the optimal slip ratio [5]. The results showed that the PID controller reduced the braking time from 3.6 s to 2.3 s. However, the model used was still simple and therefore did not fully reflect the physical phenomena in the hydraulic braking system [5]. Similarly, G. Viselga et al. integrated the dynamics of the wheel, brake pad, and brake drum into a single Simulink model [6]. Although this model covered many components, the use of transfer functions to describe mechanical components may cause the characteristic nonlinearities of an actual hydraulic system to be lost [6]. Concurrently, recent research trends have shifted toward intelligent control (such as Arduino-based ABS) and multi-objective structural optimization (using MOPSO or Autodesk Inventor) to enhance safety and reduce disc brake operating temperatures. To complement these diverse approaches, the application of Simcenter Amesim in this study not only addresses structural considerations but also enables a deeper, quantitative analysis of these non-linear hydrodynamic characteristics and structural parameters within a unified multi-domain simulation environment [7-11].

Among current tools, Simcenter Amesim is a powerful multidomain simulation platform based on bond graph techniques and the lumped-parameter method. Unlike conventional simulation tools, Amesim enables close coupling among translational mechanical, pneumatic, and hydraulic domains [8].

Although current simulation methods such as Matlab/Simulink or ANSYS have addressed individual aspects related to control algorithms or detailed flow characteristics, there is still a lack of an integrated multidomain model capable of simultaneously representing real hydraulic characteristics, including compressibility and pressure waves, as well as specific mechanical structural parameters for a hydraulic braking system. In particular, transient phenomena and the quantitative effects of design parameters have not yet been systematically analyzed in a unified simulation environment. Therefore, this study focuses on developing a purely hydraulic braking system model with a vacuum booster, without considering the control processes of electronic control systems, on the Simcenter Amesim platform. The model aims to analyze the operating characteristics of the fluid flow and the distribution of brake pad clamping force between the two axles. At the same time, it evaluates the effects of structural

parameters using the developed model, thereby providing a basis for selecting and optimizing design parameters to improve the efficiency and stability of the braking system.

2. Schematic diagram of the hydraulic braking system in a passenger car

Within the scope of this study, only the purely hydraulic actuation system is considered, while the operation of electronically controlled braking systems is not taken into account. The schematic diagram of the system is presented in Figure 1.

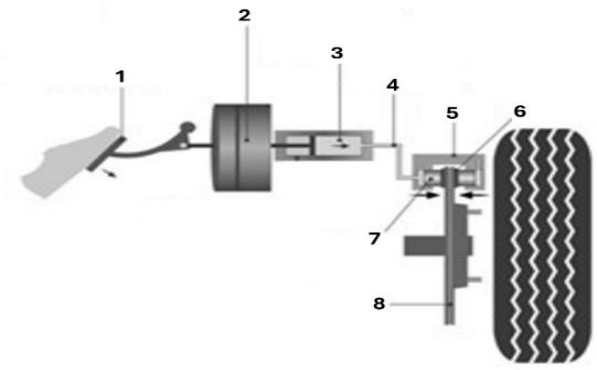


Figure 1. *Schematic diagram of a vacuum-assisted hydraulic braking system*

1. Brake pedal; 2. Vacuum booster; 3. Master cylinder; 4. Brake fluid pipe; 5. Brake caliper; 6. Brake pad; 7. Piston; 8. Brake disc.

When the driver depresses brake pedal 1, the pedal force generated by the driver's foot is amplified through vacuum booster 2. The force from the push rod acts on master cylinder 3, causing the piston in master cylinder 3 to compress the brake fluid at high pressure. The pressure from the brake fluid is transmitted to the braking mechanisms at the wheels, forcing pistons 7 in the wheel brake cylinders against brake pads 6. The brake pads are then pressed against brake disc 8, generating friction and performing the braking process.

3. Model development

The modeling process of the system follows the lumped parameter model approach, in which the system is divided into functional blocks corresponding to actual physical components. The subsystem models of the braking system are developed using elements from the mechanical, hydraulic, pneumatic, and signal control libraries.

The translational mass elements MECMAS21 are used to simulate the motion of pistons and mechanical components. The hydraulic chambers BAP11 and BAP12, combined with the BAP016 element, describe the pressure generation process and the nonlinear characteristics of valves and return springs. The pipeline system and fluid flow are developed through BHC11 connection nodes and the BHORF0 orifice element to simulate flow behavior and pressure losses. In addition, elastic and damping elements such as LSTP00A and SD0000A are used to describe the elastic characteristics, friction, and mechanical clearances of the system. The mechanical, hydraulic, and pneumatic domains are coupled in a multidomain manner, allowing the simulation of the conversion of brake pedal force into hydraulic pressure, vacuum assistance, and brake pad clamping force at the wheels [12].

3.1. Modeling of the master cylinder

The master cylinder model is developed in the Simcenter Amesim environment based on a dual-chamber master cylinder structure to simulate the conversion of mechanical force into hydraulic pressure for two independent brake circuits.

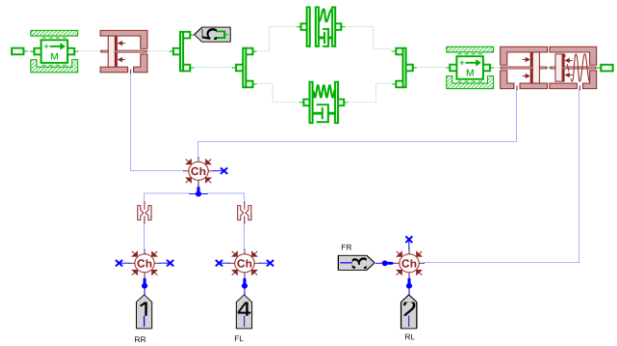


Figure 2. *Master cylinder model in Simcenter Amesim*

The oil flow rate through the port q_{mci} is determined as follows:

$$q_{mci}=v_{pi}\frac{\pi}{4}\left(d_{pi}^2-d_{ri}^2\right)0.06\frac{\rho(p_{mci})}{\rho(0)} \quad (1)$$

The pressure variation in the master cylinder is expressed as:

$$\frac{d_{p_{mci}}}{dt}=\frac{B(p_{mci})}{V_{total,i}}\left(\sum q_{in,i}\frac{\rho(0)}{\rho(p_{mci})}-\frac{dV_{total,i}}{dt}\right) \quad (2)$$

where, v_{pi} is the piston velocity (m/s); d_{pi} and d_{ri} are the diameters of the primary piston and piston rod, respectively (mm); B(p) is the bulk modulus of the brake chamber (bar); p_{mci} is the oil pressure in chamber i (bar); $V_{total,i}$ is the total volume of oil chamber i (cm^3); C_q is the flow coefficient of the orifice; A_{line} is the cross-sectional area of the pipeline (mm^2); $q_{in,i}$ is the brake fluid flow rate entering the i-th port (L/s).

3.2. Modeling of the vacuum booster

The vacuum booster model is developed to enable the analysis of complex multidomain interactions between translational mechanics and vacuum pressure.

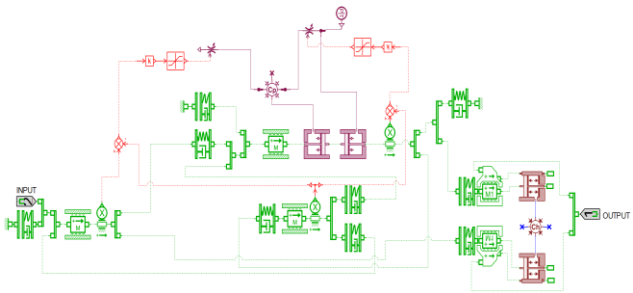


Figure 3. *Vacuum booster model in Simcenter Amesim*

The output force of the booster is expressed as:

$$F_{bst_out}=F_{pedal}+press_{pn}\frac{\pi}{4}(d_{pd}^2-d_{rb}^2)10^{-6} \quad (3)$$

Where, F_{bst_out} is the output force of the vacuum booster

transmitted to the master cylinder (N); F_{pedal} is the force applied to the brake pedal (N); $press_{pn}$ is the pressure difference between the two chambers in the booster (Pa); d_{pb} is the diameter of the booster piston, namely the diaphragm piston (mm); d_{rb} is the diameter of the booster piston push rod (mm).

3.3. Modeling of the braking mechanism

The disc brake mechanism model in Simcenter Amesim is developed to simulate the conversion of hydraulic pressure from the master cylinder into mechanical clamping force, creating contact between the brake pad and the brake disc to generate braking torque.

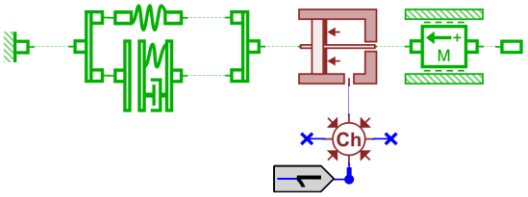


Figure 4. *Disc brake mechanism model in Simcenter Amesim*

The actual braking force is generated when the brake pad moves beyond the free clearance and comes into contact with the brake disc.

The brake pad displacement pen_{brk} (m) is expressed as:

$$pen_{brk}=-(gap_{0_brk}-x_{pc}) \quad (4)$$

The brake pad clamping force $F_{clamping}$ (N) is expressed as:

$$F_{clamping}=K_{brk}pen_{brk}+D_{eff}v_{pc} \quad (5)$$

where, K_{brk} is the contact stiffness of the brake pad material (N/m); D_{eff} is the effective damping coefficient during contact (N/(m/s)); x_{pc} is the piston displacement stroke (m).

The oil pressure at the caliper (p_{cal}) is not constant but varies according to the flow rate transmitted from the master cylinder:

$$\frac{d_{p_{cal}}}{dt}=\frac{B(p_{cal})}{V_{total_c}}\left(\left(q_{line}-q_{cal}\right)\frac{\rho(0)}{\rho(p_{cal})}-\frac{dV_{total_c}}{dt}\right) \quad (6)$$

where: $B(p_{cal})$ is the bulk modulus of the brake fluid (bar); V_{total_c} is the total oil volume in the wheel brake cylinder (cm^3); q_{line} is the oil flow rate supplied from the pipeline (L/s);

3.4. Development of the overall braking system model

The overall hydraulic braking system model is developed by integrating the subsystem model assemblies to form a complete simulation model.

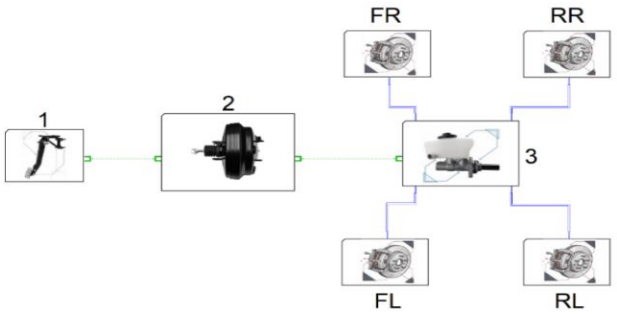


Figure 5. *Overall hydraulic braking system model in Simcenter Amesim*

1. Brake pedal; 2. Vacuum booster; 3. Master cylinder; FL: Front-left braking mechanism; FR: Front-right braking mechanism; RL: Rear-left braking mechanism; RR: Rear-right braking mechanism

The overall model is developed based on the principle of port causality, ensuring that the output variables of one assembly serve as the input variables of the subsequent assembly through power or signal transmission ports.

4. Simulation and model evaluation

The hydraulic braking system of a passenger car was simulated using the developed model. The braking process was simulated with a constant maximum braking force of 400 N, a brake fluid viscosity of 1000 N/(m/s), and a brake force rise time (t_1) varying from 0.2 s to 0.4 s. Varying the brake pedal force rise time enables the sensitivity and effectiveness of the braking system to be evaluated under braking intensity conditions ranging from moderate braking to emergency braking. The brake pedal force profile is shown in Figure 6.

Table 1. *Technical specifications of the hydraulic braking system for a five-seat passenger car*

No.	Parameter	Value	Unit
1	Master cylinder piston diameter	22.2	mm
2	Primary piston mass of the master cylinder	0.3	kg
3	Secondary piston mass of the master cylinder	0.2	kg
4	Vacuum booster diaphragm diameter	250	mm
5	Front wheel brake mechanism piston diameter	57.22	mm
6	Rear wheel brake mechanism piston diameter	33.06	mm
7	Front wheel brake mechanism piston mass	0.75	kg
8	Rear wheel brake mechanism piston mass	0.4	kg
9	Brake fluid line diameter	3.16	mm
10	Brake fluid line length to the front-left wheel	1.1	m
11	Brake fluid line length to the front-right wheel	1.8	m
12	Brake fluid line length to the rear-left wheel	3.8	m
13	Brake fluid line length to the rear-right wheel	4.4	m

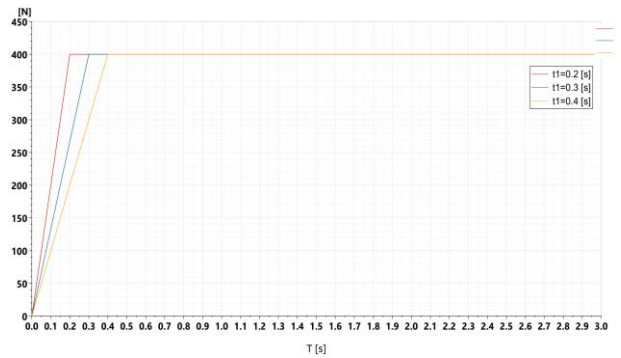


Figure 6. *Brake pedal force profile*

The simulation results show that the oil pressure in the master cylinder (Figure 7) and the wheel cylinders (Figure 8) vary linearly with the brake pedal force and reaches a steady state of approximately 88 bar when the pedal force reaches 400 N. The brake force rise time has a clear influence on the pressure build-up process. During emergency braking, with t_1 = 0.2 s, pressure appears early after approximately 0.006 to 0.007 s, and a transient phenomenon occurs with a peak pressure of about 90 bar before stabilizing. In contrast, with t_1 = 0.4 s, pressure builds up more slowly, after approximately 0.009 to 0.012 s; the pressure increase process is smoother, and oscillations are significantly reduced. This result reflects the effects of fluid

inertia, brake fluid compressibility, and the elastic characteristics of the system on the dynamic response.

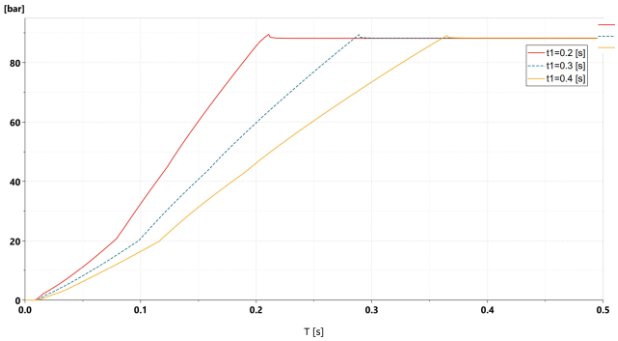


Figure 7. *Pressure build-up process in the master cylinder*

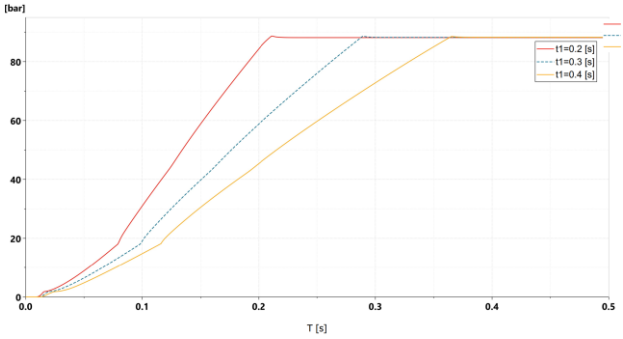


Figure 8. *Oil pressure at the wheel cylinders*

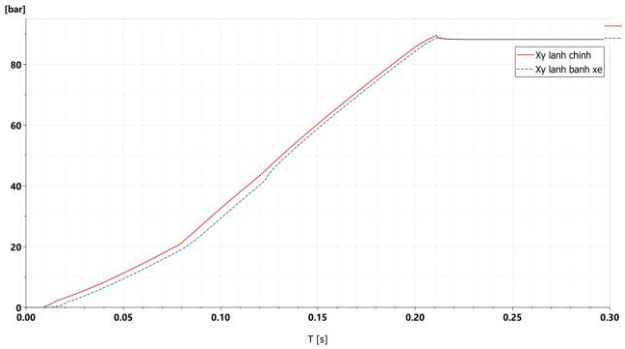
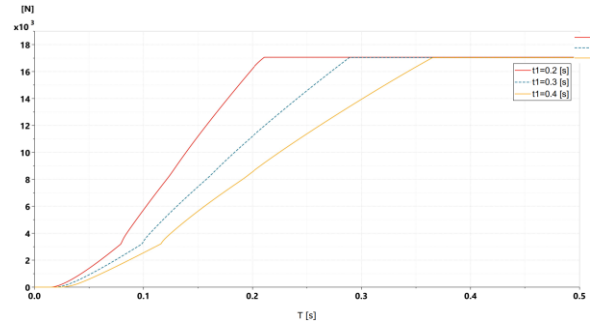


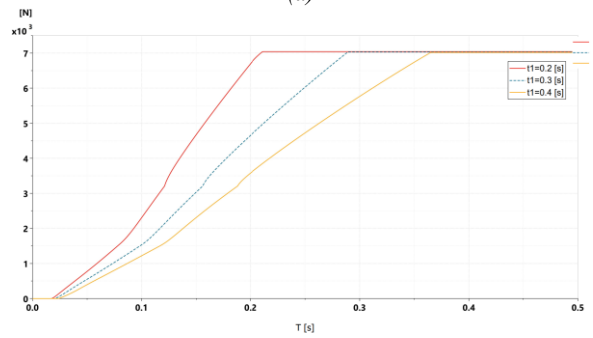
Figure 9. *Pressure delay between the master cylinder and the wheel cylinders*

The pressure at the wheel cylinders reaches values equivalent to that in the master cylinder, indicating that pressure losses in the pipelines are negligible under the investigated conditions, although a small transmission delay of approximately 0.003 s still exists (Figure 9). In addition, there is a clear difference in brake pad clamping force between the front and rear axles. When the brake pedal force reaches 400 N and t_1 = 0.2 s, the maximum brake pad clamping force at the front axle reaches approximately 17 kN, which is about 2.4 times greater than that at the rear axle, approximately 7 kN. This is consistent with the load distribution characteristics during braking and the design requirements for ensuring braking efficiency and vehicle motion stability.

Within the scope of this study, several characteristic parameters of the system, including the viscous friction coefficient r_{visc}, the initial clearance between the brake pad and brake disc gap_0, and the stiffness of the piston return spring k, were varied in a controlled manner to evaluate their effects on the operating process of the system.



(a)



(b)

Figure 10. *Brake pad clamping force at the wheels (a) Front wheels; (b) Rear wheels*

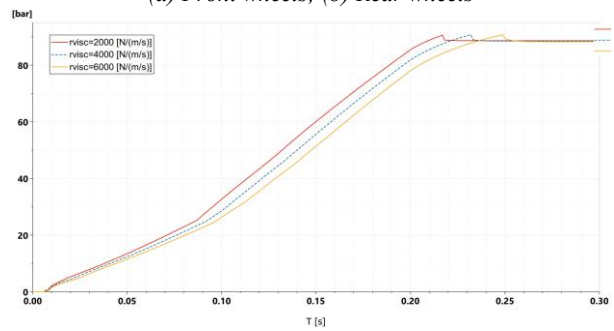


Figure 11. *Oil pressure in the master cylinder with varying viscous friction coefficient*

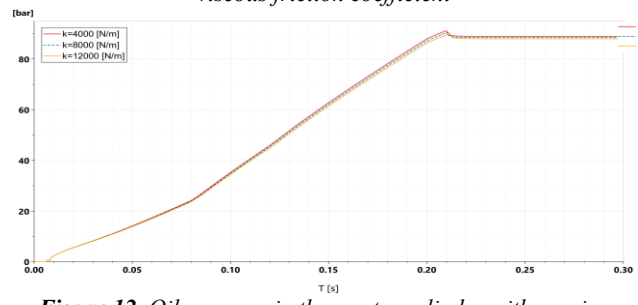


Figure 12. *Oil pressure in the master cylinder with varying spring stiffness*

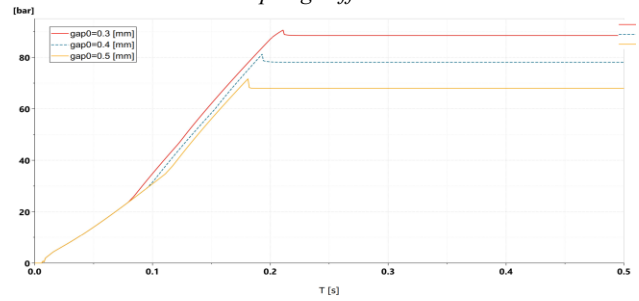


Figure 13. *Oil pressure in the master cylinder with varying clearance between the brake pad and brake disc*

The simulation results show that structural parameters such as the viscous friction coefficient (r_{visc}), the return spring stiffness (k) and the brake pad clearance (gap_0) all significantly affect the pressure response of the system. When r_{visc} increases from 2000 to 6000 N/(m/s), the time required to reach the maximum pressure increases from approximately 0.217 s to 0.249 s. At the same time, the initial pressure oscillations decrease, and the pressure build-up process becomes more stable (Figure 11). For the return spring, when the stiffness (k) increases from 4000 to 12000 N/m, the steady-state pressure decreases slightly from approximately 89 bar to 87 bar because part of the pedal force is consumed to overcome the elastic force of the spring (Figure 12). Meanwhile, the brake pad clearance has a more pronounced effect. When gap_0 increases from 0.3 mm to 0.5 mm, the steady-state pressure decreases from approximately 88 bar to 68 bar, corresponding to a reduction of about 23%, because the system requires a larger oil flow rate to compensate for the free stroke of the piston before pressing the brake pad (Figure 13).

5. Conclusions

This study developed a hydraulic braking system model on the Simcenter Amesim platform using the lumped parameter method, enabling the simulation of pressure build-up, transmission, and distribution in the system. The simulation results show that the model can represent the operating process of the system, with a maximum pressure of approximately 88 bar, a pressure transmission delay within the system of about 0.003 s, and front axle brake pad clamping forces approximately 2.4 times greater than those at the rear axle. Transient phenomena were also observed during emergency braking.

In addition, the study proposed a method for evaluating the effects of several structural parameters, such as the viscous friction coefficient, return spring stiffness, and brake pad clearance, on the system response. When the viscous friction coefficient increases, the pressure delay increases by approximately 0.032 s. When the return spring stiffness increases, the pressure decreases by approximately 2 bar. In particular, when the brake pad clearance increases by 0.2 mm, the pressure decreases by approximately 23%. These results provide a basis for parameter selection and optimization during the design, manufacturing, and operation of the system.

The developed simulation model can be used as an effective tool in the research, design, and optimization of automotive hydraulic braking systems. The obtained results not only confirm the feasibility of the simulation approach using Simcenter Amesim but also contribute to providing a scientific basis for improving the operating efficiency and stability of braking systems under actual conditions.

The current study focuses only on developing a purely hydraulic braking system model and has not yet considered electronic control systems such as ABS and EBD or whole-vehicle dynamics during braking. In future studies, the authors will develop the model by incorporating whole-vehicle dynamics and integrating electronic control systems to more comprehensively evaluate the operating characteristics and stability of the braking system under actual operating conditions.

REFERENCES

[1] S. E. Abonyi, E. E. Ezendiokwelu, and J. A. Obineche, "Modelling and simulation of braking system in automobile," *International Journal of Trend in Scientific Research and Development (IJTSRD)*, vol. 9, no. 3, pp. 7–14, 2025.

[2] Robert Bosch GmbH, *Automotive Handbook*, 11th ed. Hoboken, NJ, USA: John Wiley & Sons, 2022.

[3] V. H. Quan, N. X. Tuan, L. D. Nghiem, N. M. Tien, and T. V. M. Tu, "Building a model of intelligent brake system on vehicle using MATLAB-Simulink," *Journal of Marine Science and Technology*, no. 72, pp. 37–42, 2022.

[4] F. Ivan, P. Boran, and P. Davor, "A detailed model for simulation of hydrodynamic processes in hydraulic braking system with ABS on motor vehicles," *Journal of Scientific Research & Reports*, vol. 9, no. 3, pp. 1–12, 2016, doi: 10.9734/JSRR/2016/20630.

[5] S. E. Abonyi, E. E. Ezendiokwelu, and J. A. Obineche, "Modelling and simulation of braking system in automobile," *International Journal of Trend in Scientific Research and Development (IJTSRD)*, vol. 9, no. 3, pp. 7–14, Jun. 2025.

[6] G. Viselga, E. B. Ugnenko, E. N. Uzhviieva, O. N. Tymchenko, and N. I. Sorochuk, "Simulation modeling of the automobile braking system performance," in *IOP Conference Series: Materials Science and Engineering*, vol. 1021, *International Scientific Conference Energy Efficiency in Transport (EET 2020)*, Kharkiv, Ukraine, Nov. 2020, p. 012051, doi: 10.1088/1757-899X/1021/1/012051.

[7] P. M. Man, N. L. C. Thanh, and N. T. H. Van, "An analysis into the process of enhanced heat loss in disc brake surfaces based on local heat transfer with cooling flutes," *UD-JST*, vol. 11, no. 108.2, pp. 37–42, 2016. [Online]. Available: https://jst-ud.vn/jst-ud/article/view/1790.

[8] N. L. C. Thanh, N. V. Dong, and N. V. Hai, "ABS technology solutions to increase braking performance for motorcycle," *UD-JST*, vol. 21, no. 5, pp. 13–18, 2023. [Online]. Available: https://jst-ud.vn/jst-ud/article/view/8384.

[9] H. Thang and H. Dung, "Optimization design magnetorheological brake with multiple disk," *UD-JST*, vol. 17, no. 1.2, pp. 46–48, 2019. [Online]. Available: https://jst-ud.vn/jst-ud/article/view/1643.

[10] T. C. Chi, "Optimization of brake disc structure for mass and temperature reduction using the MOPSO algorithm," *UD-JST*, vol. 23, no. 5A, pp. 51–56, 2025. doi: 10.31130/ud-jst.2025.138.

[11] P. Q. Thai, P. V. Binh, and V. C. Tai, "Design of an anti-lock braking control system for motorcycles," *UD-JST*, vol. 18, no. 3, pp. 43–47, 2020. [Online]. Available: https://jst-ud.vn/jst-ud/article/view/3082.

[12] Siemens, *Simcenter Amesim Reference Manual*. Siemens Digital Industries Software, 2023.

[13] B. H. Trieu, N. N. Que, Đ. H. Quyet, and N. V. Huu, *Hydraulic and Pneumatic Transmission*. Hanoi, Vietnam: Hanoi Publishing House, 2006.

[14] W. Li, X. Wang, X. Leng, and M. Wang, "Modeling and simulation of automobile braking system based on kinetic energy conversion," in *2008 IEEE Vehicle Power and Propulsion Conference*, Harbin, China, 2008, pp. 1–3, doi: 10.1109/VPPC.2008.4677741.

[15] L. He, X. Wang, Y. Zhang, J. Wu, and L. Chen, "Modeling and simulation vehicle air brake system," in *Proc. 8th International Modelica Conference*, Dresden, Germany, 2011, pp. 430–435, doi: 10.3384/ecp11063430.

[16] M. Mitschke and H. Wallentowitz, *Dynamik der Kraftfahrzeuge*. Wiesbaden, Germany: Springer Vieweg, 2014.

[17] M. Selvaraj, S. Gaikwad, and A. Suresh, "Modeling and simulation of dynamic behavior of pneumatic brake system at vehicle level," *SAE Technical Paper*, no. 2014-01-2494, 2014, doi: 10.4271/2014-01-2494.

[18] F. Yang, G. Li, J. Hua, X. Li, and T. Kagawa, "A new method for analysing the pressure response delay in a pneumatic brake system caused by the influence of transmission pipes," *Applied Sciences*, vol. 7, p. 941, 2017, doi: 10.3390/app7090941.

[19] H. A. Dung, "Building the model of hydraulic brake system with a vacuum amplifier," *Vietnam Mechanical Engineering Journal*, no. 11, pp. 71–73, 2017.

[20] B. Lenzo, Ed., *Vehicle Dynamics: Fundamentals and Ultimate Trends*, 1st ed. Cham, Switzerland: Springer, 2022.