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РАЗРАБОТКА ИМИТАЦИОННОЙ МОДЕЛИ СИСТЕМЫ ПНЕВМАТИЧЕСКОГО ПРИВОДА ТОРМОЗОВ ТЯЖЕЛЫХ ГРУЗОВЫХ АВТОМОБИЛЕЙ

DEVELOPING A SIMULATION MODEL OF THE AIR ACTUATION SYSTEM FOR PNEUMATIC BRAKES IN HEAVY-DUTY TRUCKS

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Пневматические тормоза и сегодня широко используются в больших грузовиках. Однако одним существенным недостатком этой системы является время ее срабатывания, на которое влияют многочисленные параметры системы из-за сложности пневматических тормозных систем. Чтобы решить эту проблему и поддержать исследовательские процессы, в рамках данного исследования была разработана имитационная модель системы пневматического привода тормоза для грузовиков. Приводится моделирование компонентов пневматического привода по их принципиальной схеме работы в специальном пакете Simcenter-Amesim. Результаты показывают, что модель точно отражает характеристики клапанных узлов и работу системы.

Pneumatic brakes remain widely used in large trucks today. However, one significant drawback of this system is its response time, which is influenced by numerous system parameters due to the complexity of pneumatic brake systems. To address this issue and support research processes, this study developed a simulation model of the pneumatic brake system's air actuation system. Modeling of pneumatic drive components according to their operating principle in the special program Simcenter-Amesim is presented. The results demonstrate that the model accurately reflects the characteristics of the valve assemblies and the operation of the system.

Ключевые слова: *пневматические тормоза, главный пневматический тормозной клапан, время срабатывания, релейный клапан.*

Keywords: *pneumatic brakes, pneumatic dual brake valve, response time, relay valve.*

INTRODUCTION

The pneumatic brake system is widely applied in large trucks due to its high braking force with minimal control force. Hence, there is a continual demand for research and improvement in air brake systems [1–7]. Studies often focus on optimizing the response time of the system [1, 6, 8, 9] as well as the control algorithms of pneumatic ABS. Previous research typically regarded valves as flow passages accompanied by a volume [10]. Nowadays, studies on pneumatic brake systems often incorporate computer simulations to solve complex problems and enhance the accuracy of simulation models [11, 12]. In a study by Zhe Wang and colleagues, it was concluded that the dual brake valve and pipeline accounted for 80 % of the total delay of the entire system. Among these, the delay caused by the pipeline and connectors can be up to 30 % [1, 8]. Some optimization studies on system operation include the use of auxiliary brake chambers [13] to support the main brakes and optimizing ABS control algorithms. Based on these facts [14], we have developed a simulation model to serve research on pneumatic brake systems in heavy-duty vehicles.

METHODS

This study simulates a basic system with the diagram shown in fig. 1. Using this model, it is possible to investigate the impact of valve assemblies as well as the parameters of the air ducts in a basic pneumatic brake system. Additionally, in this system, the dual brake valve also has a complex structure. By using the blocks in the simulation of this valve, other valves in the system can be simulated as well.

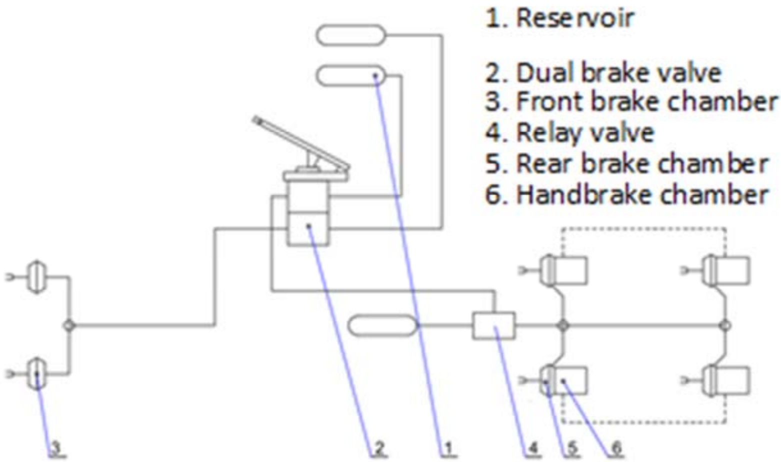


Figure 1 – Diagram of the compressed air brake system used in the study

Modeling the dual brake valve and relay valve.

To model the air brake system, the complex and important step is to model the valves (relay valve and master valve). The type of master valve we use for modeling is the type of dual-chamber master valve commonly used on trucks today, which has a schematic diagram as shown in fig 2.

When simulating the operation of the dual brake valve, it can focus on the main structures that create two chambers: the upper chamber and the lower chamber. The upper chamber connects to the rear axle outlet consisting of the upper piston, upper valve; the lower chamber consists of the lower piston, and lower valve. When braking and holding, the brake pedal presses the cup down to compress the compression spring, which has a very high stiffness compared to the remaining return springs, the compression spring pushes the upper piston down to touch the upper valve closing the exhaust path of the upper chamber, then both go down to open the intake valve supplying air to the upper chamber.

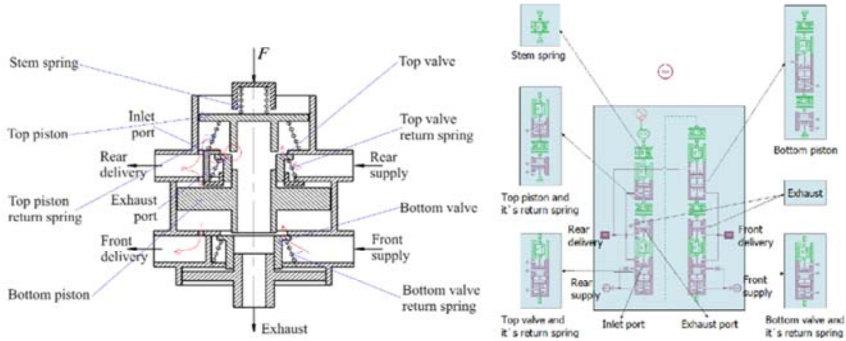


Figure 2 – Schematic diagram of the dual brake valve

For the lower chamber, the operation is similar to the upper chamber: air from the upper chamber flows down to the lower piston pushing the lower piston down to touch the lower valve closing the exhaust path of the lower chamber, and then all goes down to open the intake, at this point the air supply is connected to the lower chamber. As for the operation when holding the brake pedal, the air pressure in the two chambers will gradually increase to a certain value to create a pressure pushing the upper piston back up to balance with the force of the pedal, then both the upper and lower valves balance at the closed position maintaining stable brake pressure. When braking to the maximum, the upper valve will touch and directly push the lower piston down along with the lower valve, at this point both valves open and the pressure increases to the maximum. When releasing the pedal, the return spring pushes the valves and pistons back to the initial position closing the 2 valves and opening the exhaust ports.

The operating principle of the relay valve is also similar to the operating principle of the dual brake valve: when braking, air passes through the dual brake valve into the control port of the relay valve, pushing the piston down to touch the valve closing the exhaust port and both continue down to open the intake valve at this point the air from the air reservoir is connected to the brake chamber outlet.

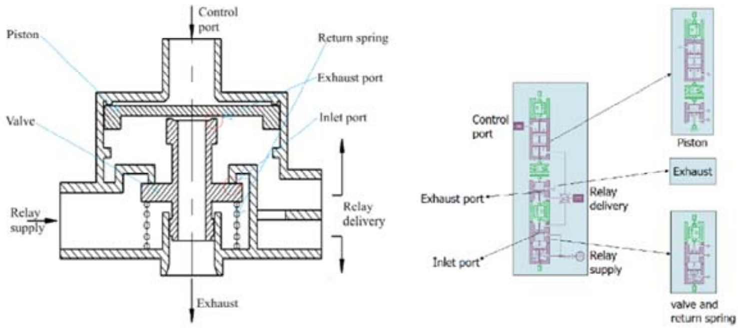
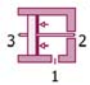
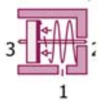
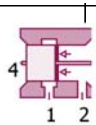
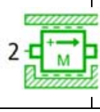


Figure 3 – Schematic diagram of the relay valve

The research team used blocks to simulate the dual brake valve using Simcenter software. In which a branch of the dual brake valve is described by a piston and an air valve. In addition, there are also return springs as well as descriptions for the mass and friction of the components. To solve these problems, the paper used blocks in tabl. 1.

Table 1 – Main blocks used in system simulation

No.	Name	Image	Description	Parameters to determine
1	Pn_psensor (PNPS001)		Compressed air pressure sensor.	Offset gain
2	Pn_bap2 (PNPA002)		Non-return spring pneumatic piston. Combined with 1 block PNPA003 to simulate the lower piston.	Piston diameter Required diameter
3	Pn_bap3 (PNPA003)		Pneumatic piston with return spring simulated for two pistons and two valves of the dual brake valve.	Piston diameter Required diameter Spring stiffness Spring force
4	Pn_bao1 (PNAO011)		The opening and closing of the valve are based on the position of the piston simulating the opening and closing of the two air valves.	Piston diameter Required diameter Initial position
5	Mass_friction _endstops		An object moving straight in one direction simulates the travel distance of the pistons and valves.	Mass Friction Limit of movement

End of Table 1

6	Springdamper 01 (SD0000)		The damping system with stiffness and damping simulating the compression spring effect of the piston.	Stiffness Damping
7	Elasticontact (LSTP00A)		Represents a collision between two objects with mass simulating the distance between the piston and the valve.	Distance,Stiffness Damping, Elastic deformation

The Pn_psensor (PNPA003) block is used to describe a pressure sensor. It is described by the following equation:

$$p_{sig} = \left(press_3 + p_{atm} - offset \right) \cdot gain \,,$$

where p_{sig} is the output pressure signal of the sensor; $press_3$ is the pressure at port 3; p_{atm} is the atmospheric pressure.

The Pn_bap2 (PNPA002) and Pn_bap3 (PNPA003) blocks describe a piston without a return spring and with a return spring. One side of the piston is connected to the atmosphere and is not affected, while the other side is subjected to a gas pressure. When using these two blocks, it is necessary to determine the parameters such as piston diameter, rod diameter, and the parameters of the spring for the Pn_bap3 block. This block is described by the following equations:

– the current chamber length: $length = x_0 + x_3$;

– volume: $vol_1 = length \cdot \pi/4 \cdot \left(dp^2 - dr^2 \right)$;

– derivative of volume: $d_{vol1} = -v_3 \cdot \pi/4 \cdot (dp^2 - dr^2)$;

– the total force on port 3: $f_3 = f_2 + press_1 \cdot \pi/4 \cdot \left(dp^2 - dr^2 \right) + f_{1x}$;

– spring force: $f_{1x} = K \cdot \left(x_3 + x_{s0} \right) + f_0$,

where dp is piston diameter, dr is rod diameter; x_0 is chamber length at zero displacements; x_3 is the displacement of port 3; v_3 is the velocity of port 3; $press_1$ is the pressure at port 1; f_2 is the force at port 2; K is spring rate; f_0 is spring force at zero displacements, x_{s0} is spring compression at zero displacements.

The Pn_bao1 (PNAO011) block is a valve that opens and closes based on the position of the piston simulating the opening and closing of

2 air valves. When using this block, it is necessary to determine 3 parameters: piston diameter, rod diameter, and initial position of the piston. This block is described by the following equations:

– the underlap: $x = x_0 + x_3$;

– the current length of chamber: $length = len_0 + x_3$;

– chamber volume: $vol_2 = length \cdot \pi/4 \cdot \left(dp^2 - dr^2\right)$.

The contribution due to the movement of the spool is passed at current pressure at port 2 to the adjacent PCD chamber: $d_{vol2} = v_3 \cdot \pi/4 \cdot \left(dp^2 - dr^2\right)$.

Force at port 3: $f_3 = f_4 + f_{jet} - press_2 \cdot \pi/4 \cdot \left(dp^2 - dr^2\right)$,

where len_0 is chamber length at zero displacement; x_3 is displacement of port 3; dp is piston diameter; dr is rod diameter; v_3 is velocity of port 3; f_4 is force at port 4; f_{jet} is flow force; $press_2$ is pressure at port 2.

The Mass_friction_endstops (MECMAS21) block is an object with mass moving in one direction under the influence of frictional force simulating the pistons and valves. When using this block, it is necessary to determine the mass, limit of movement, and parameters of the frictional force. This block is described by the following equations:

Force:

$$F_{min} = \begin{cases} Kb_{min} \cdot (x_{min} - x_1) - Db_{min} \cdot (1 - e^{\frac{x_1 - x_{min}}{Pd_{min}}}) \cdot v_1 & if\ x_1 < x_{min}\ ; \\ 0 & if\ x_1 \geq x_{min} \end{cases}$$

$$F_{max} = \begin{cases} Kb_{max} \cdot (x_1 - x_{max}) - Db_{max} \cdot (1 - e^{\frac{x_{max} - x_1}{Pd_{max}}}) \cdot v_1 & if\ x_1 < x_{min}\ . \\ 0 & if\ x_1 \geq x_{min} \end{cases}$$

Acceleration:

$$acc_1 = \frac{1}{mass} . \left(F_{\min} - F_{\max} - F_{fric} \cdot \mathrm{sign}\left(v_1\right) - rvisc \cdot v_1 - wind \cdot v_1 \cdot \left|v_1\right|\right),$$

where $Kb_{\min}$ is lower limit stiffness; $Kb_{\max}$ is higher limit stiffness; $Db_{\min}$ is lower limit contact damping coefficient; $Db_{\max}$ is higher limit contact damping coefficient; $Pd_{\min}$ is lower limit penetration for full damping; $Pd_{\max}$ is higher limit penetration for full damping; $x_{\min}$ is lower displacement limit; $x_{\max}$ is higher displacement limit; x_1 is displacement of port 1; F_{fric} is dry friction force; $v1$ is velocity of port 1; *rvisc* is coefficient of viscous friction; wind is coefficient of windage.

The Springdamper01 block (SD0000) is a simulation block for the influence of the compressed spring on the piston. When using this block, 2 parameters need to be set, which are stiffness and damping. This block is simulated by the following equations:

– the total force: $force_1 = force + damperforce$,

$$damperforce = cdamp \cdot \left(vel_1 + vel_2\right);$$

– derivative of force: $\dfrac{dforce}{dt} = kval \cdot \left(vel_1 + vel_2\right)$.

The spring compression: $x = \dfrac{force}{srate}$,

where *force* is spring force; *damperforce* is damper force; *cdamp* is damper rating; *kval* is spring stiffness value; *srate* is spring rate.

The Elasticendsto block (LSTP00A) represents a collision simulation for the distance between the piston and the valve. When using this block, parameters such as distance, stiffness, damping, and deformation need to be set. This block is described by the following equations:

– gap: $gap = gap_0 - x_1 - x_2$;

– penetration: $pen = -gap$;

– penetration velocity: $dpen = v_1 + v_2$,

where x_1 is displacement of port 1; x_2 is displacement of port 2; v_1 is velocity of port 1; v_2 is velocity of port 2.

Using the structural diagrams of the clusters in Fig 1 system combined with the functional description blocks in Table 1, a simulation diagram of the entire system has been built using Simcenter software. The diagram is described as in fig. 4.

RESULTS AND DISCUSSION

The results in fig. 5 show that when the distance between the bottom piston and the bottom valve of the master brake cylinder is small, it affects the pressure release process due to the smaller throttle area of the exhaust path of the front axle brake branch. Additionally, changing the piston diameter in the dual brake valve does not significantly impact the pressure increase and decrease, as the volume of air contained by the dual brake valve is negligible compared to the amount of air entering the brake chamber. In fig. 6, when the distance between the bottom piston and the bottom valve of the dual brake valve with a piston diameter of 110 mm is changed, the pressure release varies while the pressure increase remains unchanged. This is because the distance does not alter the throttle area of the air valve during pressure increase but only affects the throttle area during pressure release.

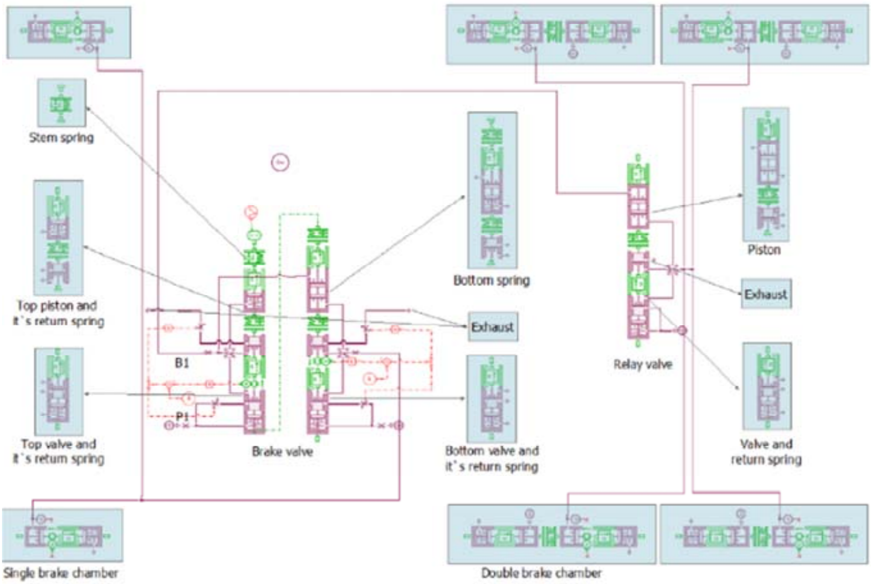


Figure 4 – Brake system diagram constructed in Simcenter

The system also successfully simulated the effectiveness of the dual brake valve, showing that incorporating the booster reduced the system's response time by approximately 20 % and the pressure release time by about 50 %. These results are highly reliable, as the effectiveness of the dual brake valve has been demonstrated in previous studies [15, 16]. When the brake pedal is applied at 50 %, the system pressure only reaches a maximum of about 3,7 Bar. The discrepancy in peak pressure is due to the fixed brake pedal position, while the distance between valves changes, leading to different displacements of the air valves.

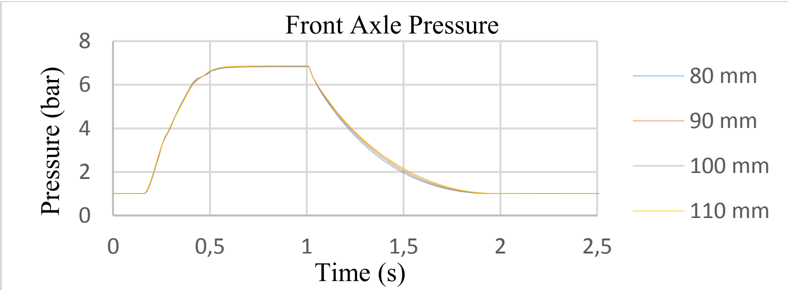


Figure 5 – Front Chamber Pressure at Full Pedal Travel

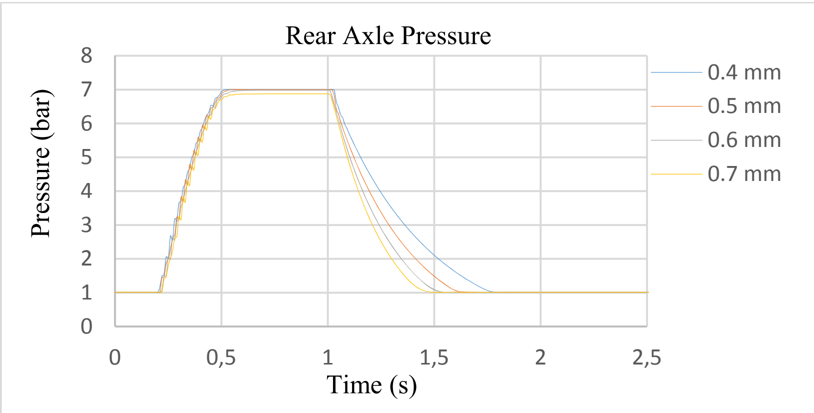


Figure 6 – Rear Chamber Pressure at Full Pedal Travel

CONCLUSION

The results demonstrate that the study successfully simulated the detailed components of the pneumatic brake system as well as the system's operation. The outcomes accurately reflect the pressure increase and decrease in the brake chambers under various conditions, such as changing the brake pedal position and structural parameters of the dual brake valve. This simulation framework can be utilized for investigating and evaluating the operation of pneumatic brake systems in research studies.

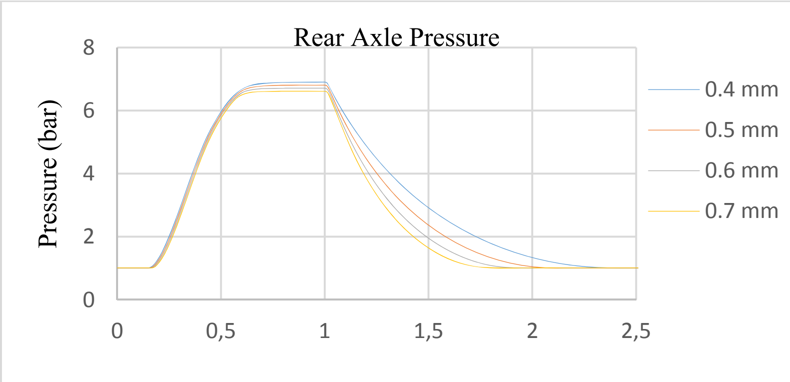


Figure 7 – Rear Axle Brake Chamber Pressure in a System without a Relay Valve at Full Pedal Travel

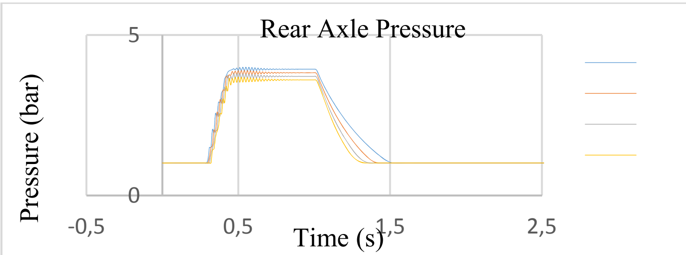


Figure 8 – Rear Axle Brake Chamber Pressure in a System with a Relay Valve at 50 % Brake Pedal Travel

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