

Robust Design of a Pneumatic Brake System in Commercial Vehicles

Jinglai Wu, Hongchang Zhang, Yunqing Zhang and Liping Chen

Huazhong University of Science & Technology, China

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ABSTRACT

The air brake system has been widely used since its great superiority over many other kinds of brake systems, but the capacity and the stability of air brake system are determined by many factors which are always uncertain and difficult to be evaluated accurately. So it is necessary to improve the robustness of this kind of brake system. In this paper, a physical model of air brake control system is made by a multi-domain physical modeling software-AMESim and the robust design for air brake system is carried out. Firstly, the key design parameters that will greatly affect on the delay time and pressure that leads to the shaking problem are obtained by using the method of design of experiment (DOE). Then, the regress of the response surface based on results of DOE and the robust design using the tolerance design are carried out. The value for those key parameters that can lead to the best performance and robustness of the air brake system are finally determined.

INTRODUCTION

Brake system affects the safety of automotive directly. It is necessary to research robustness of brake system. Compared to hydraulic brake system, the air brake system has many advantages such as air brake system can produce larger brake force without oversize pedal force or pedal stroke, and the disposal of pipelines in the air brake system is much simpler and so on. In fact, most tractor-trailer vehicles with a gross vehicle weight rating (GVWR) over 19000 lb, most single trucks with a GVWR over 31000 lb, most transit and intercity buses, and about half of all school buses are equipped with air brake systems [1]. The air brake system in the most commercial vehicles is mainly made up of two subsystems, namely the pneumatic subsystem and the mechanical subsystem. The pneumatic subsystem

includes the compressor, storage reservoirs, treadle valve (or the brake application valve), brake lines, relay valves, quick release valve (QRV), and brake chambers, etc. The mechanical subsystem starts from the brake chambers and includes pushrods, slack adjusters, S-cams, brake pads, and brake drums, etc[2]. Generally, the air brake system will show longer delay time than the hydraulic brake system as the response of air pressure is more slowly relative to hydraulic pressure. So, in this paper the pneumatic subsystem is mainly studied.

In fact, the delay time for the pressure response will lead to enormous influence on the performance of the vehicle brake system. Shorten the delay time can lead to short brake distance, especially when the initial velocity of automotive is very high. Also the quick pressure response is one of the most important inevitable working conditions for the anti-brake system (ABS) which has been widely used in automotives. Thus, to shorten the time for the pressure response is necessary. On the other hand, the pressure in the air brake system may change sharply and suddenly, which in turn will lead to vehicle seriously shaking problem. We define the swing of pressure as *Shake_pressure*. The *Shake_pressure* should be minimized, because it not only affect the brake system working performance but also leads to much noise sometimes.

The mathematic model for the air brake system robustness analysis is very complicated since many parameters may be correlated with each other especially the parameters for the key components, such as brake valve, relay valve and pipelines. A non-linear model of the pneumatic subsystem of the air brake system which relates the pressure in the brake chamber to the treadle valve plunger displacement and the supply pressure to the treadle valve has been developed and presented in [3]. But the model doesn't contain mathematic model for the pipelines. Bowlin [4] developed the pressure control

scheme which can be used to control the response of air brake system, it can be further used as a part of emerging technologies such as adaptive cruise control (ACC) systems for commercial vehicles.

Fluid power transmission and control problem are extremely complex. Some parameters are just obtained from empirical equations, thus adding inaccuracies to the model [5]. Thus, simulating the system based on this kind of mathematic models is difficult to achieve the accurate results. Even sometimes it is impossible to obtain the results as the mathematic model of pneumatic brake system is very complicate and incomplete.

However, modeling and simulating the air brake system are very convenient by using the AMESim software[6] .

Actually, there are many uncertain factors which will lead to many kinds of errors in product design, such as the manufacture or assemble process, so the research on the robustness of brake system under uncertain factors is necessary. Robust design which has been developed to improve the product quality and reliability in industry can be used to solve the stability problem of pneumatic brake system. The concept of robust design had been initially introduced by Taguchi in the late1940s and this technique has become commonly known as the Taguchi design [7, 8]. Now many new methods have been developed for the robust design such as the Responsible Surface Methodology (RSM), the tolerance designs Methodology and so on.

Robust optimization can be formulated by capturing the design characteristics. However, regardless of the design characteristics, its objective and constraint functions are composed of their statistics. When the RSM for statistics are utilized, all of the functions in the design formulation can be expressed in mathematical forms, which leads to a simple classic optimization problem. The critical step in the procedure is to make a reliable response surface model to replace the true statistics [9]. Once this has been accomplished, the tolerance model can be further used to robust optimization.

In the following pages, we present the model of a pneumatic subsystem which is based on the practical physical structure. The model of key components that include storage reservoirs, brake valve, pipelines, relay valve and brake chambers is made by AMESim. Then the pneumatic subsystem base on those key components model is made and the pressure response time is obtained through simulation. Then find and select the parameters which lead to great influence on the delay time and shake pressure through design of experiment (DOE). DOE can also be used to determine the levels of some of these selected design parameters. Finally, regress the response surface based on the results obtained by DOE and do the robust design using the tolerance design [10-14]. By using the methodology mentioned above, we can get the structure parameters of pneumatic subsystem which can not only meet the

steady request but also can lead to better dynamic performance. Fig. 1 shows the robust design procedure.

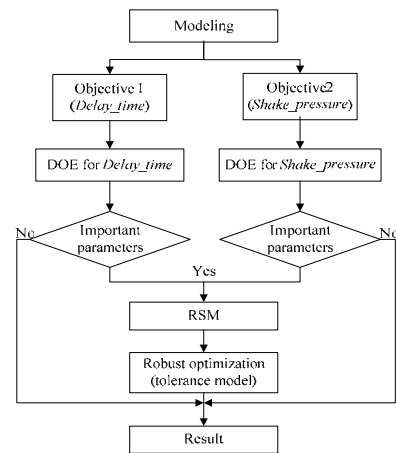


Figure. 1 The robust design procedure

DESCRIPTION OF THE PNEUMATIC BRAKE SYSTEM AND OBJECT OF EXPERIMENT

THE COMPONENTS OF PNEUMATIC SUBSYSTEM

An essential layout of the pneumatic subsystem in the air brake system of a commercial vehicle is shown in Fig.2. An engine-driven air compressor is used to compress air and the compressed air is collected in storage reservoirs. The pressure of the compressed air in the reservoirs is regulated by a governor. Compressed air is supplied from these reservoirs to the brake and relay valves. The driver applies the brake by pressing the brake pedal on the brake valve. This action makes the compressed air flow from the supply port of the treadle valve to its delivery port. Then, the compressed air travels from the delivery port of the treadle valve through air hoses to the brake chambers mounted on the axles, see details in [2].

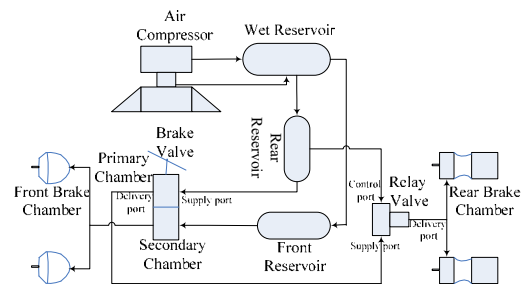


Figure. 2 Schematic representation of a pneumatic brake system (without parking brake system)

It is to note that there are two chambers in the brake valve and each chamber contains both a supply port and a delivery port. The supply port of secondary chamber is connected to the front reservoir while the delivery port is connected to the front service brake chamber. The

supply port of primary chamber connects to the rear reservoir accordingly. However, the delivery port of primary chamber does not connect to the rear brake chamber directly. The delivery port of primary chamber connects to the control port of relay valve which is used to speedup the response velocity. Relay valve contains a supply port, a delivery port and an exhaust port besides the control port. The supply port connects to the rear reservoir and the delivery port connects to the rear brake chamber. The compressed air from the primary chamber of brake valve is just used as pressure signal.

THE MODEL OF PNEUMATIC BRAKE SYSTEM

Brake valve

The structure of tandem dual circuit brake valve is shown in Fig. 3.

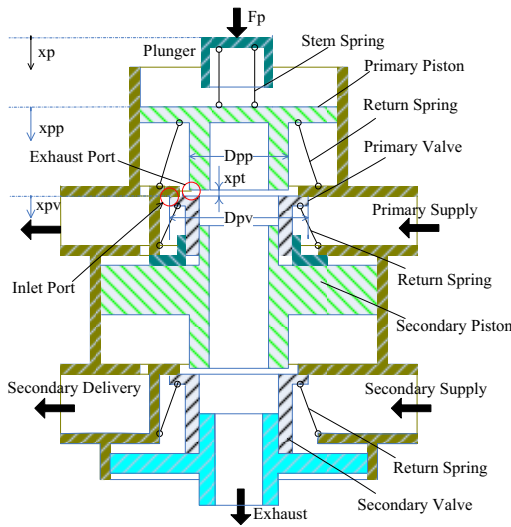


Figure. 3 Structure of tandem dual circuit brake valve

The primary circuit is actuated by the pedal force and the secondary circuit acts essentially as a relay valve. Under normal operating conditions, air that flows from the primary delivery is used to actuate a relay piston which in turn actuates the secondary circuit. When the primary circuit fails, the secondary circuit is actuated directly by pedal force. When the brake pedal is applied, the primary piston first closes the primary exhaust and then opens the primary inlet valve. This phase is called the "apply phase." When the delivery pressure increases to a level where it balances the pedal input force, the primary inlet valve is closed with the primary exhaust also remaining closed. This phase is called the "balance or the hold phase." When the pedal is released, so is the balancing force on the primary piston, which causes the primary piston to lift off from the primary exhaust seat and the air in the delivery circuit is exhausted to the atmosphere thereby releasing the brakes. This phase is referred to as the "exhaust phase." [2] The secondary circuit functions in a similar

manner except for the fact that the relay piston is used to actuate it under normal operating conditions.

The governing equations for the working primary piston of the brake valve are given by

If $x_{pp} \leq x_{pt}$,

$$\begin{cases} M_{pp} \ddot{x}_{pp} = F_p - (F_{kpp} + K_{pp} x_{pp} + P_{pd} A_{pp}) \\ F_p = K_{ss} (x_p - x_{pp}) \\ x_{pv} = 0 \\ V_{pd} \frac{dP_{pd}}{dt} = mR \frac{dT_{pd}}{dt} + RT_{pd} \dot{m} \\ C_v m \frac{dT_{pd}}{dt} = \alpha s (T_0 - T_{pd}) - RT_{pd} \dot{m} \\ \dot{m} = A_{xpv} C_q C_m \frac{P_{pd}}{\sqrt{T_{pd}}} = \pi D_{pv} (x_{pt} - x_{pv}) C_q C_m \frac{P_{pd}}{\sqrt{T_{pd}}} \end{cases}, \quad (1)$$

where

$$C_m = \begin{cases} \sqrt{\frac{2}{R} \cdot \frac{\gamma}{\gamma-1}} \cdot \sqrt{\left(\frac{P_{atm}}{P_{pd}}\right)^2 - \left(\frac{P_{atm}}{P_{pd}}\right)^{\frac{\gamma+1}{\gamma}}}, & \text{if } \frac{P_{atm}}{P_{pd}} > P_{cr} = 0.528 \\ \sqrt{\frac{2}{R} \cdot \frac{\gamma}{\gamma-1}} \cdot \left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}, & \text{if } \frac{P_{atm}}{P_{pd}} \leq P_{cr} = 0.528 \end{cases}$$

If $x_{pp} > x_{pt}$,

$$\begin{cases} M_{pp} \ddot{x}_{pp} = F_p - (F_{kpp} + K_{pp} x_{pp} + P_{pd} A_{pp} + F_{pv}) \\ F_p = K_{ss} (x_p - x_{pp}) \\ M_{pv} \ddot{x}_{pv} = F_{pv} + P_{pd} A_{pv} - (F_{kpv} + K_{pv} x_{pv} + P_{ps} A_{pv1}) \\ x_{pv} = x_{pp} - x_{pt} \\ V_{pd} \frac{dP_{pd}}{dt} = mR \frac{dT_{pd}}{dt} + RT_{pd} \dot{m} \\ C_v m \frac{dT_{pd}}{dt} = \alpha s (T_0 - T_{pd}) + RT_{pd} \dot{m} \\ \dot{m} = A_{xpv} C_q C_m \frac{P_{ps}}{\sqrt{T_{ps}}} = \pi D_{pv} x_{pv} C_q C_m \frac{P_{ps}}{\sqrt{T_{ps}}} \end{cases}, \quad (2)$$

where

$$C_m = \begin{cases} \sqrt{\frac{2}{R} \cdot \frac{\gamma}{\gamma-1}} \cdot \sqrt{\left(\frac{P_{pd}}{P_{ps}}\right)^2 - \left(\frac{P_{pd}}{P_{ps}}\right)^{\frac{\gamma+1}{\gamma}}}, & \text{if } \frac{P_{pd}}{P_{ps}} > P_{cr} = 0.528 \\ \sqrt{\frac{2}{R} \cdot \frac{\gamma}{\gamma-1}} \cdot \left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}, & \text{if } \frac{P_{pd}}{P_{ps}} \leq P_{cr} = 0.528 \end{cases}$$

In Eqs.(1) and (2), M_{pp} and M_{pv} denote the mass of the primary piston and the primary valve assembly gasket respectively. x_p , x_{pp} and x_{pv} denote, respectively, the displacement of the treadle valve plunger, the primary piston and the primary valve from their initial position, x_{pt} is the distance traveled by the primary piston before it

makes the primary exhaust close and is also the exhaust clearance of primary valve, K_{ss} , K_{pp} , and K_{pv} denote, respectively, the spring constants of the stem spring, the primary piston return spring, and the primary valve assembly return spring, F_{kpp} , and F_{kpV} denote, respectively, the preloads on the same, A_{pp} is the net area of the primary piston exposed to the pressurized air at the primary delivery port, A_{pv} is the net cross-sectional area of the primary valve assembly gasket exposed to pressurized air at the primary delivery port, A_{pv1} is the net cross-sectional area of the primary valve assembly gasket exposed to the pressurized air at the primary supply port, F_p is the force transmitted from the plunger to the primary piston by the stem spring and rubber graduating spring, F_{pv} is the force transmitted from the primary piston by the primary valve, T_0, T_{pd} and T_{ps} is the temperature of the environment, air at the primary delivery port and air being supplied to the primary circuit respectively, D_{pp} and D_{pv} is the diameter of exhaust valve and primary valve respectively, m is the mass of flow air, s is the thermal exchange area of primary chamber, P_{ps} is the pressure of air being supplied to the primary circuit, P_{pd} is the pressure of air at the primary delivery port, and P_{atm} is the atmospheric pressure, other signs are some coefficients for thermodynamics [6]. The governing equations for secondary chamber are similar to the primary chamber. Fig. 4 shows the tandem dual circuit brake valve model in AMESim software.

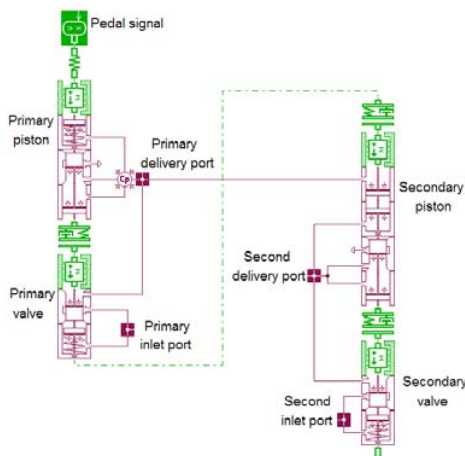


Figure. 4 The tandem dual circuit brake valve Model

From Eq. (1), if the area of piston and the stiffness of valve return spring have been determined, we can find out the key parameters which will affect the pressure responses and stabilization are the stiffness of primary piston return spring, the exhaust clearance of the primary valve and the second valve.

Relay valve

The structure of relay valve is similar to treadle valve, as shown in Fig. 5.

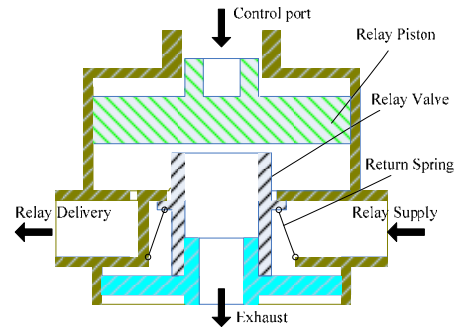


Figure. 5 Structure of relay valve

The difference between the brake valve and relay valve is that the control signal for brake valve is the force supplied by pedal while the control signal for relay valve is the compressed air from the brake valve. Relay valve can shorten the time of response of pressure for the rear brake chambers which are far from the brake valve. When the compressed air in brake valve flows to the control port of relay valve, the diaphragm and mandril open the valve which make the compressed air in the rear reservoir flow to relay valve directly. The compressed air that flows to the brake chamber gets to a steady pressure along with air pressure in control port got the stabilization. When the pressure of air in control port gets low with the pedal releasing, the diaphragm and mandril retoured while the valve is closed and the air in the brake chamber is exhausted to the atmosphere thereby releasing the brakes. Fig.6 shows the relay valve model in AMESim software.

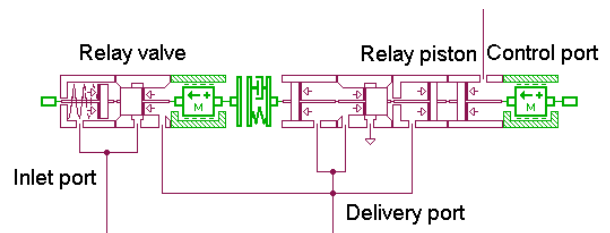


Figure. 6 The relay valve model

Thus, if the area of piston and the stiffness of relay valve return spring have been determined, the exhaust clearance of relay valve can be obtained.

Brake chamber

In this paper, we simplify the rear service and spring brake chamber as the front service brake chamber (see Fig. 7) Set the volume of the brake chambers as our research focus.