

Pneumatic Brake Apply System Response and Aero-Acoustic Performance Considerations

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ABSTRACT

Over the past decade, the automotive industry has seen a rapid decrease in product development cycle time and an ever increasing need by original equipment manufacturers and their suppliers to differentiate themselves in the marketplace. This differentiation is increasingly accomplished by introducing new technology while continually improving the performance of existing automotive systems. In the area of automotive brake system design, and, in particular, the brake apply subsystem, an increased focus has been placed on the development of electrohydraulic apply systems and brake-by-wire systems to replace traditional pneumatic and hydraulic systems. Nevertheless, the traditional brake apply systems, especially vacuum-based or pneumatic systems, will continue to represent the majority of brake apply system production volume into the foreseeable future, which underscores the need to improve the performance and application of these traditional systems in passenger cars and light-trucks. This paper introduces the test methodologies and math models needed to predict, characterize, and improve the response and aero-acoustic performance of pneumatic apply systems during the brake development process.

INTRODUCTION

The automotive brake system design and development process is increasingly executed through a system of "virtual" development activities that are centered around the application of math-based system models and traditional computer-aided engineering tools such as finite-element and computational fluid dynamics models. However, a number of areas still require an almost exclusive use of physical prototypes to accurately quantify and select the optimal performance of the brake system, which includes the brake system response and aero-acoustic performance of pneumatic brake apply systems.

The goal of this paper is to introduce math models and experimental techniques used to predict and characterize the response and aero-acoustic performance of pneumatic apply systems early in the brake development process. In particular, this paper will focus on the effect of a pneumatic apply system and other pertinent system design factors on overall brake system response and, separately, its related aero-acoustic performance. An overview of vacuum-based apply systems will be given along with the outline of a comprehensive brake system response math model. Vacuum-based apply system aero-acoustic performance will be detailed with the introduction of an empirical math model to predict vacuum-based apply system noise performance. Also, brake apply system test methodologies will be reviewed with a focus on response and noise performance. Lastly, an illustration of how the apply system and other brake parameters can impact brake system response performance will be examined using the response math model with an analytical Design for Six Sigma (DFSS) approach. In future work, a comprehensive fully integrated brake response and aero-acoustic model will be detailed that can be used to optimize a brake system design.

PNEUMATIC BRAKE BOOSTER MECHANICS

The basic operation of a typical passenger car or light truck pneumatic booster equipped hydraulic brake system has been covered previously in the literature, so only a brief review will be afforded here.

In the resting (released) position, a source of vacuum (usually the engine, but in some cases a vacuum pump) evacuates both the apply and the vacuum chambers on either side of the booster diaphragm, resulting in a net zero force on the diaphragm. When the driver applies force to the brake pedal, it is amplified by the pedal lever ratio and transmitted through the apply system input rod to the booster air valve. When sufficient force has been applied to overcome the air valve return spring force, the air valve opens and permits ambient air to flow through air passages in the power piston body and into the

chamber on the apply side of the booster diaphragm. Once sufficient pressure across the diaphragm has been accrued to overcome the booster return spring force, the air valve/power piston assembly will move against the return spring, and in turn against the output rod. The output rod is moved against the master cylinder input piston, and when sufficient force is applied, it will move against the fluid in the master cylinder chamber. Fluid is pushed up into the reservoir until the compensating ports are covered, and then fluid is moved through the brake pipes towards the ABS modulator (if equipped) and then towards the brake hoses to the calipers. In a dual-master cylinder, there are two hydraulic circuits separated by a floating piston; this floating piston transmits force from the primary master cylinder chamber to the secondary. There are usually points in the hydraulic circuit of smaller diameter than the brake pipe inner diameter, such as orifices in the modulator, junctions between the pipe and the brake hose, and the brake hose to caliper attachment point. When fluid flows into the calipers, it moves the caliper piston out against the piston seal towards the brake pads and brake rotor.

There is a force-feedback system in the brake booster consisting of a rubber reaction disk, which is seated against the power piston. At rest, it has a small air gap between itself and the tip of the air valve. The power piston transmits its output force through the reaction disk, which causes it to extrude through an opening in the power piston towards the tip of the air valve. When the extruded reaction disk contacts the tip of the air valve, it behaves quite analogously to a hydraulic circuit with two different piston sizes corresponding approximately to the diameter of the seat of the disc in the power piston and the tip of the air valve. (The ratio of force on the reaction disc and force on the air valve tip is the reaction ratio). The reaction disc transmits the reaction force from the booster output back into the air valve after reducing it by the reaction ratio, through the input rod, and to the brake pedal where the driver can feel it.

When the driver stops moving the air valve towards the power piston, air pressure across the diaphragm will move the power piston away from the air valve, which will close it and stop the flow of ambient air into the apply chamber. The power piston will move only far enough to close the air valve passage to the apply chamber, at which point the forces in the system are in equilibrium. It does not move far enough to open the valve. The booster remains in this rest position, with the pneumatic force across the booster balancing the force applied to the hydraulic system, as long as the driver does not release force on the brake pedal.

When the driver releases the brake pedal, the air valve moves back relative to the power piston, opening the valve between the vacuum chamber and the apply chamber. Air is evacuated from the apply chamber until pressure is equalized across the diaphragm in a fully released state, or until the driver ceases the release

motion and the power piston is again moved relative to the air valve, closing the path between the vacuum and apply chambers and balancing force in the system.

PNEUMATIC APPLY SYSTEM RESPONSE & AERO-ACOUSTIC PERFORMANCE TESTING

Apply systems can be evaluated in a laboratory for the purpose of accurately measuring a number of performance characteristics. These characteristics include response time and the corresponding aero-acoustics or characteristic booster “whoosh” noise, which have a direct impact on vehicle stopping distance, pedal feel and cabin interior sound quality. In order to accurately and repeatably quantify pneumatic booster response time and noise, the test bench/set-up simulates the pertinent aspects of the vehicle environment. A test fixture is used to rigidly mount the booster to the test bench, which maintains the location of the apply system throughout the test in an attitude similar to its position in the vehicle. An adjustable vacuum source is used to supply vacuum to the booster, which ensures that the vacuum level and any vacuum level dynamics that appear in the vehicle are accurately reproduced during testing. The brake corner compliance is also simulated with the appropriate hardware (i.e. ABS modulator, brake lines, etc.) plumbed downstream from the master cylinder and a range of simulated compliance levels available, which allow for evaluations under various corner compliance configurations. Lastly, an adjustable force is supplied to the booster input rod, which allows the force input to closely follow a customers’ typical input during both standard and emergency braking situations.

Apply system response and aero-acoustic performance can be measured simultaneously, however, this paper will focus on the response and aero-acoustic performance individually. In order to quantify the aero-acoustic performance of the apply system alone, the pneumatic booster is placed into an anechoic test chamber with a cutoff frequency lower than the lowest frequency of acoustic energy emitted by the booster. Furthermore, the chamber must be able to maintain a low ambient background noise sound pressure level (SPL) sufficient to accurately measure the noise. Since the booster must be held rigidly in order to conduct testing during an apply, a fixture is required within the test chamber in order to properly locate the booster, which is depicted in Figure 1. The booster fixture is designed to allow the unobstructed placement of a microphone near the booster air valve entrance tube, which is also depicted in Figure 1.

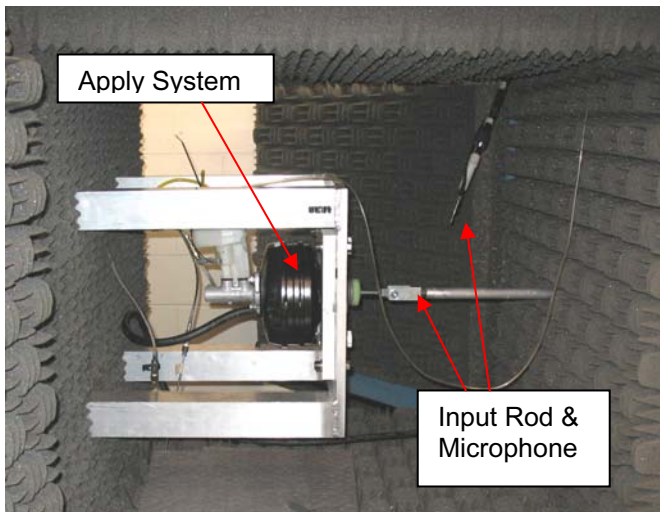


Figure 1. Pneumatic Booster Test Chamber

With the booster properly attached to the fixture within the test chamber, the vacuum source is attached to the booster check valve and adjusted to accurately reflect vacuum levels provided by the vehicle's powertrain. The input force levels are determined from the booster runout levels, which were measured during prior testing. With the input levels set, a step apply and release is made with the booster, and an acoustic quality data acquisition system is used to measure the booster "whoosh" noise. Figure 2 shows a typical pneumatic booster response/aero-acoustic evaluation setup. From these types of applies, the apply system response time can be quantified from the 63.2% master cylinder primary port pressure rise time. The release response time is determined as the time necessary for the master cylinder primary port pressure to reach 36.8% of the steady-state value. During the test, the input force, booster input rod travel, booster vacuum levels, and master cylinder output pressure are digitally recorded. Furthermore, the Zwicker loudness and SPL levels during both the apply and release is computed as well.

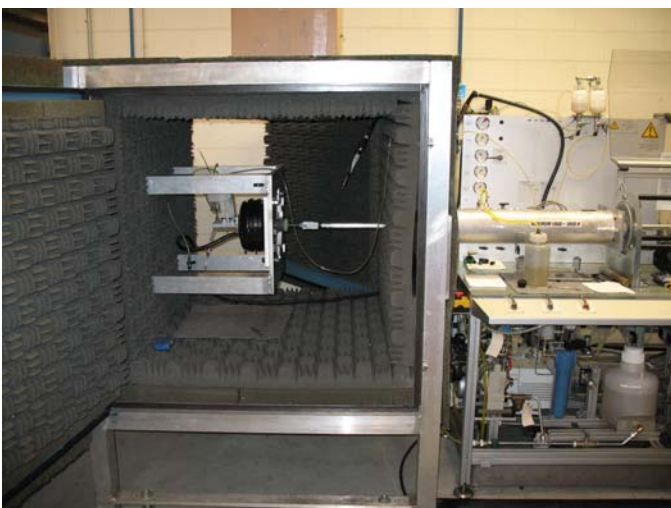


Figure 2. Pneumatic Booster Test Bench and Chamber

BRAKE SYSTEM AND PNEUMATIC BOOSTER RESPONSE MODEL

The ability to mathematically model and predict brake system response early in the vehicle development process is important due to the effect that system response has on overall brake performance, including pedal feel and stopping distance. A simplified diagram of the brake system and associated parameters in the model is shown in Figure 3 below:

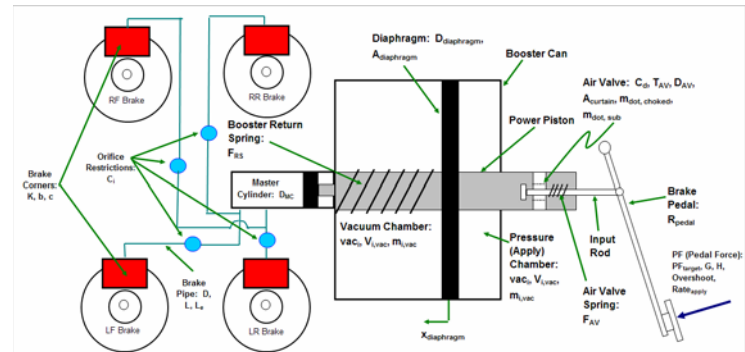


Figure 3: Illustration of Brake System Functional Response Model

For the purposes of modeling brake system response, some simplifying assumptions are made:

- Airflow through the booster air valve is modeled as compressible, 1-dimensional flow. It is choked above the critical pressure ratio and subsonic below. Flow equations for a generic poppet air valve are used.
- Flow losses through the air passages in the booster are neglected; it is assumed that the air valve dominates pneumatic flow losses.
- Inertias of moving components are considered to be negligible with respect to the pneumatic and hydraulic flow losses, and are therefore not modeled.
- The assumption is applied that a brake response-type apply is sufficiently rapid to discourage significant airflow through the check valve back to the vacuum source. In other words, the booster does not recharge significantly during the response apply.
- The booster, whether single or tandem, is represented with a single piston-cylinder system of equivalent working area to the actual booster.
- Bends in the brake pipe are represented with an 'equivalent length' – a length of straight pipe that results in flow losses equivalent to bends.
- The hydraulic flow model features one orifice restriction. The restriction must simulate orifices in the chassis controls and/or at brake pipe to hose junctions and brake hose to caliper junctions (banjo bolt).
- Brake fluid is modeled as incompressible and of uniform properties throughout the brake system.

- Losses in the floating piston in a dual-circuit master cylinder are neglected (the same pressure is applied at the master cylinder junction to each of the four brake pipes), and it is assumed that brake fluid only flows in one direction (pressure is only rising or held steady).
- The brake system is in a steady-state, constant flow condition for each time step the model is solved for.

The flow and force balance equations for the response model are as follows. Starting with the air valve, the curtain area is defined per equation (1) below:

$$(1) \quad A_{curtain} = \pi \cdot D_{AV} \cdot L_{AV}$$

where,

$A_{curtain}$ = air valve curtain area

D_{AV} = air valve curtain area diameter

L_{AV} = lift of the air valve from its seat

Flow through the booster air valve is modeled using the equations for compressible flow through a poppet valve, as shown in reference [1]:

$$(2) \quad \dot{m}_{subcritical} = \frac{C_d \cdot A_{curtain} \cdot P_{in}}{(R \cdot T_{in})^{0.5}} \cdot \left(\frac{P_{vac}}{P_{in}} \right)^{1/\gamma} \cdot \left\{ \frac{2 \cdot \gamma}{\gamma - 1} \cdot \left[1 - \left(\frac{P_{vac}}{P_{in}} \right)^{(\gamma-1)/\gamma} \right] \right\}^{0.5}$$

$$(3) \quad \dot{m}_{choked} = \frac{C_d \cdot A_{curtain} \cdot P_{in}}{(R \cdot T_{in})^{0.5}} \cdot \gamma^{0.5} \cdot \left(\frac{2}{\gamma + 1} \right)^{(\gamma+1)/(2(\gamma-1))}$$

where,

C_d = air valve discharge coefficient

P_{in} = ambient pressure at the booster inlet

T_{in} = temperature of air local to the booster inlet

R = ideal gas constant for air at temperature T_{in}

P_{vac} = air pressure on the vacuum side of the booster diaphragm(s),

γ = specific heat ratio of air

Equation (2) is used for the case of subcritical flow, and equation (3) is used for the case of choked flow. Flow is considered to be choked when the pressure ratio across the air valve is above critical, e.g.

$$(4) \quad \left(\frac{P_{vac}}{P_{in}} \right) \leq \left[\frac{2}{\gamma + 1} \right]^{\gamma/(\gamma-1)}$$

The discharge coefficient of the valve may be estimated by correlating the model to measured data, it may be determined analytically, or it may be measured directly in a flow bench. Figure 4 shows an example of booster

air valve discharge coefficients measured on a subsonic flow bench – hence the relatively low pressure differential across the valve. To fully characterize the air valve flow coefficient via physical testing, it would be necessary to test on a sonic flow bench, but the data from the subsonic bench is sufficient for response modeling:

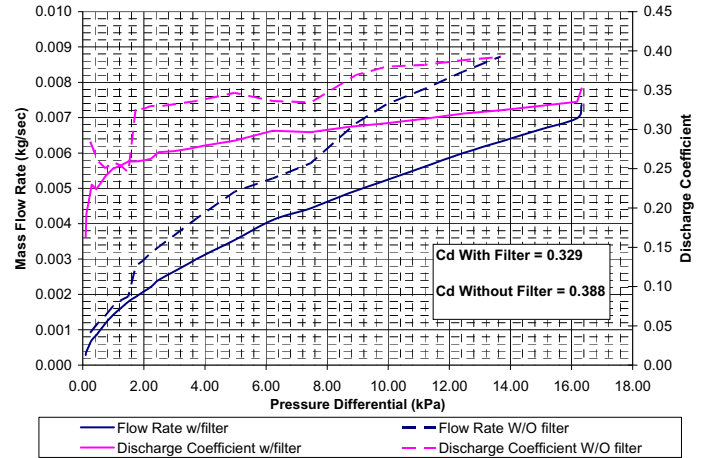


Figure 4: Example Booster Air Valve Discharge Coefficients (with and without foam silencer)

The pedal force input was modeled as a linear rise up to the overshoot value, with an exponentially decaying sinusoidal function approximating the overshoot and subsequent settling behavior. The pedal force model is defined by the following equations:

$$(5): \text{ when } PF \leq PF_{target} + \text{Overshoot} \\ PF = \text{Rate}_{apply} \cdot t$$

$$(6) \text{ when } PF > PF_{target} + \text{Overshoot}$$

$$PF = PF_{target} + \text{Overshoot} \cdot \exp(-H \cdot (t_{ovr} - t_{ovr,0})) \cdot \cos \left[\frac{2 \cdot \pi \cdot (t_{ovr} - t_{ovr,0})}{G} \right]$$

where,

T_{ovr} = Time elapsed since the pedal force target reached overshoot,

$t_{ovr,0}$ = Time elapsed since the initiation of the brake apply at which overshoot first occurs

H = Pedal force setting time factor

G = Period of oscillation factor.

The force input to the booster is taken to be the product of the pedal force and the pedal ratio, with no losses (force losses will be accounted for in an overall apply system efficiency term):

$$(7) \quad F_{in} = PF(t) \cdot R_{pedal}$$

The area of the booster diaphragm may be used if known, or estimated from apply system bench test data.