

Physical Characteristics of Twin-Tube Shock Absorber

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ABSTRACT

In an automotive suspension, a shock absorber plays a significant role to enhance the vehicle performances, particularly ride comfort and road holding. Because of its important influences on the overall vehicle performances, the understanding of its physical characteristics is essential. Thus, this paper develops a mathematical model of twin-tube shock absorber that is widely used in modern production cars. The model is derived based on a rational polynomial formulation. This formulation generally represents the flow behaviors of fluid across a restriction. Further, simulation results are compared to those obtained from experiments to determine the model accuracy. The result comparison illustrates that the model is able to describe the behavior of shock absorber with slight discrepancies.

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INTRODUCTION

An automotive suspension system is designed to isolate a vehicle body from road noises and improve road handling capability. Shock absorbers installed in parallel with suspensions are one of the key components to enhance these desired performances. The shock absorbers are to dampen any oscillations generated from road disturbances by converting kinetic energy into thermal energy. Typically, there are various types of shock absorbers. A hydraulic shock absorber with the telescopic design is the most widely used forms in automotive suspensions because of its compact size, precise damping, and simple construction [1]. This design allows a piston to move inside a cylinder filled with hydraulic fluid. The telescopic shock absorber can be divided into two categories: mono-tube shock absorber (Figure 1) and twin-tube shock absorber (Figure 2). In a mono-tube shock absorber, the valves that the oil flows through during both rebound and compression stages are integrated into the piston body. Besides, a high-pressure gas chamber is serially appended to the oil chamber to collaboratively absorb some vibrations. In a

twin-tube damper, the piston valve plays a primary role to generate a damping force during the rebound stage. During the compression stage, this role is switched to the base valve. Additionally, because of its better ride quality, lower friction, and higher flexibility, the twin-tube shock absorber has presently become the standard on production cars.

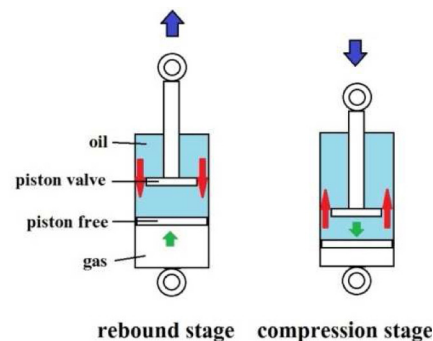


Figure 1. Mono-tube shock absorber.

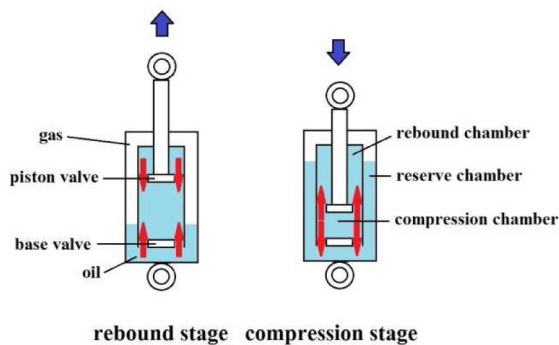


Figure 2. Twin-tube shock absorber.

As previously mentioned, a shock absorber makes a significant contribution to a suspension system in terms of ride comfort and road holding. To achieve a higher-performance damper, the understanding of its physical characteristics is essential. There have been extensive studies to investigate the behaviors of both mono-tube and twin-tube shock absorbers. Examples of mono-tube damper modeling are presented in the works of Reybrouck [2], Talbot and Starkey [3], etc. Herr et al. [4] introduced a computational fluid dynamics model of twin-tube shock absorber. This model is rather complex. Lee and Moon [5] proposed and validated the model of displacement-sensitive shock absorber. Patten et al. [6] and Mo et al. [7] defined a full model of semi-active vibration absorbers by using the effects of fluid compressibility and bulk modulus. However, the models in these works is unlikely to be fundamentally correct, since a damping force is theoretically proportional the velocity, not the displacement. Meissen [8] developed a physical model of twin-tube damper based on the Hagen-Poiseuille equation. This particular equation is typically valid for laminar flow; however, the flow in the shock absorber generally is not laminar flow due to the frequent change of flow. Hence, this model may not be able to naturally describe the complete characteristics of the damper. Heo et al. [9] studied a modeling technique to simulate the behaviors of a continuously variable damper. In this work, valves in the damper were assumed to be orifices. This assumption may be valid to a certain extent, since the valve construction is more comparable to a fluid passage than an orifice.

Referring to these previous studies, it is clear to see that an accurate analytical model of twin-tube shock absorber is still in need, particularly a low-order model. Therefore, the focus of this work is on the modeling of a modern twin-tube shock absorber to describe its physical characteristics. The shock absorber model is derived from a rational polynomial formulation [10], which is used to represent the oil flow through a valve inside the damper. Originated from this rational formulation, nonlinear terms are approximated by the Taylor series expansion. Moreover, the prediction accuracy of the model is validated with experimental results. The validation results confirm the fidelity of the proposed model.

The remainder of the paper is organized as follows: The next section describes the modeling of twin-tube shock absorber. The structure and operation of shock absorber are first explained. Further, valve characteristics are defined, and later applied to determine a force generated by the shock absorber. The effectiveness of model is examined in the next section. The last section concludes the key contributions of this paper.

TWIN-TUBE SHOCK ABSORBER MODEL

A. Valve Characteristics

As shown in Figure 2, a typical twin-tube shock absorber is configured with three chambers: a rebound chamber, a compression chamber and a reserve chamber. The rebound chamber and compression chamber are filled with oil. The reserve chamber partially contains a compressible gas that is usually nitrogen charged under high pressure. The main function of reserve chamber is to compensate a change in volume of the oil caused by the difference of cross-sectional areas between the rebound chamber and compression chamber. Further, there are two main valves: piston valve and base valve, embedded in the twin-tube shock absorber. While the oil is flowing from one chamber to another through these valve restrictions, a damping force is generated. Each valve contains two flow circuits as illustrated in Figure 3. One is an intake valve circuit that consists of a check valve. The other is a serial combination of a port restriction and a parallel flow passage. This parallel flow passage is made of a leak restriction and a blow-off valve.

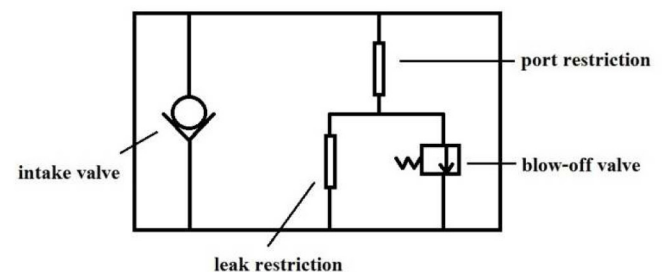


Figure 3. Schematic of piston and base valves.

The characteristics of the port restriction and leak restriction can be characterized by a passive restriction codified as a rational polynomial [10]. The flow in the shock absorber body is represented by a mix of laminar and turbulent flow with an exponent of 1.7 [10, 11]. Thus, the flow behaviors in these restrictions can be mathematically expressed as:

$$\Delta p_{\text{port}} = K_{\text{port}} v^{0.25} Q_{\text{port}}^{1.75} \quad (1)$$

$$\Delta p_{\text{leak}} = K_{\text{leak}} v^{0.25} Q_{\text{leak}}^{1.75} \quad (2)$$

where Δp is the pressure drop across a valve, K is the stiffness of restriction, ν is the fluid viscosity, Q is the flow rate through a valve. The blow-off valve contains a tension spring and a flat disc. The spring is preloaded and placed against the disc that acts as a valve. The blow-off valve remains closed until the blow-off pressure is greater than the preloaded spring pressure ($\Delta p_{\text{blow-off}} \geq \Delta p_0$). The blow-off valve is modeled as a function of spring stiffness and preloaded spring pressure:

$$K_{\text{spring}} y = \Delta p_{\text{blow-off}} A_v - \Delta p_0 A_v \quad (3)$$

where K_{spring} is the stiffness of the blow-off valve's spring, y is the opening height of blow-off valve, A_v is the cross-sectional area of blow-off valve. Furthermore, once the disc is lifted, the flow through the blow-off valve can be characterized by that in an orifice. The flow rate through the blow-off valve is described in terms of valve parameters, the pressure drop across itself, and the opening height of blow-off valve (y):

$$Q_{\text{blow-off}} = C_d y \pi d_v \sqrt{\frac{2 \Delta p_{\text{blow-off}}}{\rho}} \quad (4)$$

Then, Eqs. (3) and (4) are combined to obtain the characteristic of the blow-off valve:

$$\overline{K_{\text{spring}}} Q_{\text{blow-off}} = (\Delta p_{\text{blow-off}} - \Delta p_0) \sqrt{\Delta p_{\text{blow-off}}} \quad (5)$$

where $\overline{K_{\text{spring}}}$ is $\frac{K_{\text{spring}}}{C_d \pi d_v A_v} \sqrt{\frac{\rho}{2}}$

B. Shock Absorber Force

A shock absorber force or damping force is calculated based on the difference between the forces exerting on a piston in a rebound chamber and compression chamber. Since the magnitude of a friction force in a shock absorber is relatively small, it is typically ignored [11]. Thus, the damping force can be written as:

$$F_{\text{sa}} = A_r p_r - A_c p_c \quad (6)$$

where F_{sa} is the shock absorber force, A is the cross-sectional area of a chamber, p is the pressure in a chamber, the subscript r denotes the rebound chamber, and the subscript c denotes the compression chamber. At rest, the pressures in all chambers are equal to the static gas pressure, which is also the same as that in the reserve chamber ($p_{\text{gas}} = p_s$). However, due to the difference between the cross-sectional areas of the rebound chamber and compression chamber, there exists an initial shock absorber force that can be expressed as:

$$F_{\text{ini}} = p_s (A_r - A_c) \quad (7)$$

During the rebound stage, the piston moves upward, and it forces the oil to flow from the rebound chamber to the compression chamber through the piston valve. Further, the difference in chamber volume causes an oil shortage in the compression chamber. That shortage is supplemented by the oil flowing from the reserve chamber through an intake passage of the base valve. In the case that the piston is compressed, the oil is restricted to discharge from the compression chamber to the rebound and reserve chambers. A slight pressure drop across an intake valve is typically neglected [12]. Thus, given these assumptions, the pressure in the compression chamber is the same as that in the reserve chamber during the rebound stage, and the pressure in the rebound chamber and compression chamber are equivalent during the compression stage, which lead to:

$$p_c = p_s \quad \text{for rebound stage (8)}$$

$$p_r = p_c \quad \text{for compression stage (9)}$$

By substituting Eqs. (7), (8), and (9) into Eq. (6), the equation of shock absorber force can be obtained as:

$$\begin{aligned} F_{\text{sa}} &= p_s (A_r - A_c) + A_r (p_r - p_s) \\ &= F_{\text{ini}} + A_r (p_r - p_s) \\ &= F_{\text{ini}} + A_r \Delta p_{\text{piston valve}} \end{aligned} \quad \text{for rebound stage (10)}$$

$$\begin{aligned} F_{\text{sa}} &= p_s (A_r - A_c) + (A_c - A_r) (p_c - p_s) \\ &= F_{\text{ini}} - (A_c - A_r) (p_c - p_s) \\ &= F_{\text{ini}} - (A_c - A_r) \Delta p_{\text{base valve}} \end{aligned} \quad \text{for compression stage (11)}$$

From these equations, it is essential to obtain the pressure drops across the piston valve and base valve. As described in the previous section and illustrated in Figure 3, their structures are similar, consisting of two flow paths. Based on this construction, the following relationships can be expressed:

1. The total pressure drop across the parallel flow path is the same as the individual pressure drops across the leak valve and blow-off valve:

$$\Delta p_{\text{parallel}} = \Delta p_{\text{blow-off}} = \Delta p_{\text{leak}} \quad (12)$$

2. The total pressure drop across the whole path is the summation of the individual pressure drops across the valves:

$$\Delta p_{\text{total}} = \Delta p_{\text{port}} + \Delta p_{\text{parallel}} \quad (13)$$