

Combination of Test with Simulation Analysis of Brake Groan Phenomenon

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ABSTRACT

During a car launch, the driving torque from driveline acts on brake disk, and may lead the pad to slip against the disk. Especially with slow brake pedal release, there is still brake torque applies on the disk, which will retard the rotation of disk, and under certain conditions, the disk and pad may stick again, so the reciprocated stick and slip can induce the noise and vibration, which can be transmitted to a passenger by both tactile and aural paths, this phenomenon is defined as brake groan.

In this paper, we propose a nonlinear dynamics model of brake for bidirectional, and with 7 Degrees of Freedom (DOFs), and phase locus and Lyapunov Second Method are utilized to study the mechanism of groan. Time-frequency analysis method then is adopted to analyze the simulation results, meanwhile a test car is operated under corresponding conditions, and the test signals are sampled and then processed to acquire the features. Finally, the results of the simulation and those of the test are compared, and the mechanism of groan and its contributory factors are revealed.

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INTRODUCTION

With the rapid growth of vehicles, urban traffic conditions are getting worse. In stop and go traffic, brake vibrations and noises are becoming one of the most important factors influencing the ride and comfort.

Brake groan is very common in the vehicle equipped automatic transmission, and shows a fundamental frequency between 50~500Hz [1]. When a car starts, the brake torque will be negotiated by driving torque and the brake disk will start to slip against the brake pad. This initial stick-slip of disk excites vibration in the driveline and brake, allowing a condition where the disk and pad may stick-slip again [2].

It is suggested that brake groan is induced by stick-slip motion between pad and disk, and it occurs most readily under the lower speed and lighter brake pedal release [3]. H. Jang et al. [4] investigate groan propensity of 29 friction material specimens with different relative amounts of the ingredients, and suggest a new material of low propensity of groan phenomena during braking. A caliper-slider experimental model and corresponding theoretical model for groan analysis is proposed [5], but the motion directions of both models are inconsistent. A mechanical model with 2 DOFs is built [6], a pad and a disc is connected together through a sliding friction interface, and the rotation between the pad and disk is just simplified to tangential motion. A torsional model with 4 DOFs

is built to investigate brake groan [3], which is composed of two dynamic sub-systems coupled at the friction interface, meanwhile two brake pads are simplified to one. The pad and the disk are separated, impacted by the same brake torque.

In the recent researches, different models are built to study brake groan, while they are almost all unidirectional, and neglect the dynamic change of normal force between pad and disk. To fill this void, in this paper a nonlinear dynamics model of brake for bidirectional, and with 7 DOFs is created, in which brake disk, tire, inner & outer pad and caliper are considered. Disk, pads and tire can rotate around x axis, while pads and caliper can move along x axis. An exponential expression of friction coefficient is adopted, then its corresponding mathematical model is derived based on Lagrange's Equation of Second Kind. Phase locus and Lyapunov Second Method are utilized to study the mechanism of groan and its contributory factors, which include vehicle velocity and friction coefficient etc., moreover from the point of nonlinear system stability.

Furthermore, the dynamic responses under different operating scenarios are studied numerically, namely slow & quick brake pedal release and constant brake pedal. Time-frequency analysis method is used to analyze the features of brake groan and the effect of brake pedal release rate on it.



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Meanwhile the experimental investigation for a typical passenger car is performed to verify the simulation results. The test signals are sampled and processed with Short-Time Fourier Transform (STFT) and Power Spectral Density (PSD) Estimation. Finally the results of simulation are compared with those of test. From the study, the mechanism of groan can be found out, conclusions can be drawn, or it is the stick-slip motion that causes groan, and the friction-induced motion is highly dependent on brake pedal force, friction coefficient, and vehicle velocity.

DYNAMICS MODEL

A nonlinear model of brake for bidirectional, 7 DOFs, with brake disk, tire, inner & outer pad and caliper is considered in this section, and illustrated in Figure 1.

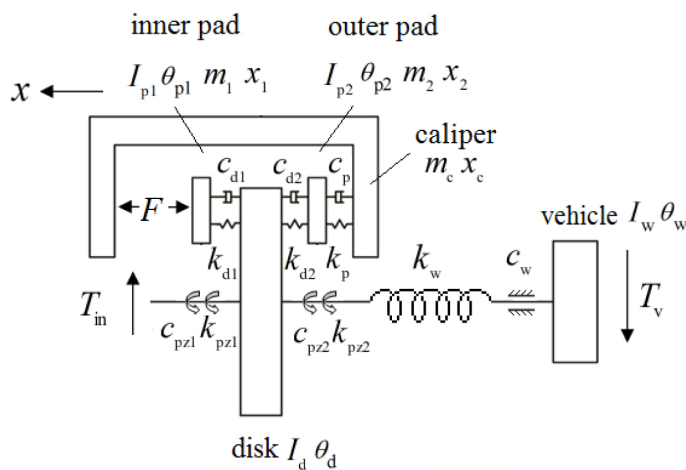


Figure 1. Dynamics model for groan analysis

There are 7 DOFs, and they are:

$$q_1 = \theta_{p1}, q_2 = \theta_{p2}, q_3 = \theta_d, q_4 = \theta_w, q_5 = x_1, q_6 = x_2, q_7 = x_c.$$

Based on Lagrange's Equation of Second Kind:

$$\frac{d}{dt} \left(\frac{\partial E_T}{\partial \dot{q}_j} \right) - \frac{\partial E_T}{\partial q_j} + \frac{\partial E_V}{\partial q_j} + \frac{\partial E_D}{\partial \dot{q}_j} = Q_j$$

$$Q_j = \sum_{k=1}^{n'} F_k \frac{\partial v_k}{\partial \dot{q}_j} + \sum_{k=1}^{m'} T_k \frac{\partial \dot{\theta}_k}{\partial \dot{q}_j}$$
(1)

Where, q_j is generalized coordinates; E_T is total kinetic energy; E_V is total potential energy; E_D is total dissipative function; Q_j is generalized forces; F_k is active force; T_k is active moment; $\frac{\partial v_k}{\partial \dot{q}_j}$ is partial velocity; $\frac{\partial \dot{\theta}_k}{\partial \dot{q}_j}$ is partial angular velocity.

In this system, E_T , E_V , E_D , Q_j can be expressed as:

$$E_T = \frac{1}{2} I_{p1} \dot{q}_1^2 + \frac{1}{2} I_{p2} \dot{q}_2^2 + \frac{1}{2} I_d \dot{q}_3^2 + \frac{1}{2} I_w \dot{q}_4^2 + \frac{1}{2} m_1 \dot{q}_5^2 + \frac{1}{2} m_2 \dot{q}_6^2 + \frac{1}{2} m_c \dot{q}_7^2$$
(2)

$$E_V = \frac{1}{2} k_{pz1} q_1^2 + \frac{1}{2} k_{pz2} q_2^2 + \frac{1}{2} k_{d1} q_3^2 + \frac{1}{2} k_{d2} q_6^2 + \frac{1}{2} k_p (q_6 - q_7)^2 + \frac{1}{2} k_w (q_3 - q_4)^2$$
(3)

$$E_D = \frac{1}{2} c_{pz1} \dot{q}_1^2 + \frac{1}{2} c_{pz2} \dot{q}_2^2 + \frac{1}{2} c_{d1} \dot{q}_3^2 + \frac{1}{2} c_{d2} \dot{q}_6^2 + \frac{1}{2} c_p (\dot{q}_6 - \dot{q}_7)^2 + \frac{1}{2} c_w (\dot{q}_3 - \dot{q}_4)^2$$
(4)

$$Q_j = T_{in} \frac{\partial \dot{q}_3}{\partial \dot{q}_j} - T_v \frac{\partial \dot{q}_4}{\partial \dot{q}_j} - T_{fl} \frac{\partial (\dot{q}_3 - \dot{q}_1)}{\partial \dot{q}_j} - T_{f2} \frac{\partial (\dot{q}_3 - \dot{q}_2)}{\partial \dot{q}_j} - F \frac{\partial (\dot{q}_5 - \dot{q}_7)}{\partial \dot{q}_j}$$
(5)

From Eq.1, 2, 3, 4, 5, equations of motion with respect to x axis can be get:

$$m_1 \ddot{x}_1 + c_{d1} \dot{x}_1 + k_{d1} x_1 = -F$$

$$m_2 \ddot{x}_2 + c_p (\dot{x}_2 - \dot{x}_c) + k_p (x_2 - x_c) + c_{d2} \dot{x}_2 + k_{d2} x_2 = 0$$

$$m_c \ddot{x}_c + c_p (\dot{x}_c - \dot{x}_2) + k_p (x_c - x_2) = F$$
(6)

Equations of motion around x axis can be yielded:

$$I_{p1} \ddot{\theta}_{p1} + c_{pz1} \dot{\theta}_{p1} + k_{pz1} \theta_{p1} = T_{f1}$$

$$I_{p2} \ddot{\theta}_{p2} + c_{pz2} \dot{\theta}_{p2} + k_{pz2} \theta_{p2} = T_{f2}$$

$$I_d \ddot{\theta}_d + c_w (\dot{\theta}_d - \dot{\theta}_w) + k_w (\theta_d - \theta_w) = T_{in} - T_{f1} - T_{f2}$$

$$I_w \ddot{\theta}_w + c_w (\dot{\theta}_w - \dot{\theta}_d) + k_w (\theta_w - \theta_d) = -T_v$$
(7)

And friction torque can be denoted:

$$T_{fi} = \begin{cases} \mu F_{ni} R & \dot{\theta}_d > \dot{\theta}_{pi} \\ T_{in} - c_w (\dot{\theta}_d - \dot{\theta}_w) - k_w (\theta_d - \theta_w) - T_{fj} & \dot{\theta}_d = \dot{\theta}_{pi} \end{cases}$$

(i = 1, 2; j = 2, 1)

F_n is the normal force applied on disk:

$$F_{ni} = (k_{di}x_i + c_{di}\dot{x}_i)\text{Sgn}(x_i), i=1,2 \quad (9)$$

Where, T_{in} is driving torque from driveline; T_v is vehicle resistance; T_{f1} is friction torque between inner pad and disk; T_{f2} is friction torque between outer pad and disk; F_{n1} and F_{n2} are the normal force applied by inner pad and outer pad respectively.

In this paper, an exponential expression of friction coefficient is adopted, and it is defined as: $\mu = \mu_d + \Delta\mu e^{-\alpha\Delta v}$.

SIMULATION APPROACH

Dynamic responses of the system are studied numerically in this section, including the mechanism of brake groan, its contributory factors and the effect of brake pedal release rate.

System Stability

From Lyapunov Second Method, it is known that system stability can be determined by eigenvalues of its Jacobian Matrix. To maintain groan phenomenon and determine its mechanism, let angular velocity of disk, $\dot{\theta}_d$, be a constant. Thus equations of motion for this system become:

$$\begin{cases} I_{p1}\ddot{\theta}_{p1} + c_{pz1}\dot{\theta}_{p1} + k_{pz1}\theta_{p1} = -\mu_1 R(c_{d1}\dot{x}_1 + k_{d1}x_1) \\ I_{p2}\ddot{\theta}_{p2} + c_{pz2}\dot{\theta}_{p2} + k_{pz2}\theta_{p2} = \mu_2 R(c_{d2}\dot{x}_2 + k_{d2}x_2) \\ m_1\ddot{x}_1 + c_{d1}\dot{x}_1 + k_{d1}x_1 = -F \\ m_2\ddot{x}_2 + c_p(\dot{x}_2 - \dot{x}_c) + k_p(x_2 - x_c) + c_{d2}\dot{x}_2 + k_{d2}x_2 = 0 \\ m_c\ddot{x}_c + c_p(\dot{x}_c - \dot{x}_2) + k_p(x_c - x_2) = F \end{cases} \quad (10)$$

Where,

$$\mu_i = \mu_d + \Delta\mu e^{-\alpha(\dot{\theta}_d - \dot{\theta}_{pi})} \approx \mu_d + \Delta\mu e^{-\alpha\dot{\theta}_d}(1 + \alpha\dot{\theta}_{pi}), \quad i=1,2.$$

Let $\theta_{p1} = q_1$, $\theta_{p2} = q_2$, $x_1 = q_3$, $x_2 = q_4$, $x_c = q_5$, $\dot{\theta}_{p1} = q_6$, $\dot{\theta}_{p2} = q_7$, $\dot{x}_1 = q_8$, $\dot{x}_2 = q_9$, $\dot{x}_c = q_{10}$.

Then:

$$\mathbf{Q} = [q_1 \ q_2 \ q_3 \ q_4 \ q_5 \ q_6 \ q_7 \ q_8 \ q_9 \ q_{10}]^T \quad (11)$$

and

$$\dot{\mathbf{Q}} = \begin{bmatrix} \dot{q}_1 \\ \dot{q}_2 \\ \dot{q}_3 \\ \dot{q}_4 \\ \dot{q}_5 \\ \dot{q}_6 \\ \dot{q}_7 \\ \dot{q}_8 \\ \dot{q}_9 \\ \dot{q}_{10} \end{bmatrix} = \begin{bmatrix} q_6 \\ q_7 \\ q_8 \\ q_9 \\ q_{10} \\ \frac{-\mu_1 R(c_{d1}q_8 + k_{d1}q_3) - c_{pz1}q_6 - k_{pz1}q_1}{I_{p1}} \\ \frac{\mu_2 R(c_{d2}q_9 + k_{d2}q_4) - c_{pz2}q_7 - k_{pz2}q_2}{I_{p2}} \\ \frac{-F - c_{d1}q_8 - k_{d1}q_3}{m_1} \\ \frac{-c_{d2}q_9 - k_{d2}q_4 - c_p(q_9 - q_{10}) - k_p(q_4 - q_5)}{m_2} \\ \frac{F - c_p(q_{10} - q_9) - k_p(q_5 - q_4)}{m_c} \end{bmatrix} \quad (12)$$

Equilibriums of the system are:

$$\begin{bmatrix} q_1 & q_2 & q_3 & q_4 & q_5 & q_6 & q_7 & q_8 & q_9 & q_{10} \end{bmatrix}^T = \begin{bmatrix} \frac{\mu_0 RP}{k_{pz1}}, \frac{\mu_0 RP}{k_{pz2}}, -\frac{F}{k_{d1}}, -\frac{F}{k_{d2}}, -\frac{F}{k_{d2}}, -\frac{F}{k_p}, 0, 0, 0, 0 \end{bmatrix}^T \quad (13)$$

Where, $\mu_0 = \mu_d + \Delta\mu e^{-\alpha\dot{\theta}_d}$.

Then the derived system is:

$$\dot{\mathbf{Q}}' = \begin{bmatrix} q_6 \\ q_7 \\ q_8 \\ q_9 \\ q_{10} \\ \frac{-\mu_0 R c_{d1} q_8 + (RF \Delta\mu \alpha e^{-\alpha\dot{\theta}_d} - c_{pz1}) q_6 - \mu_0 R k_{d1} q_3 - k_{pz1} q_1}{I_{p1}} \\ \frac{\mu_0 R c_{d2} q_9 - (RF \Delta\mu \alpha e^{-\alpha\dot{\theta}_d} + c_{pz2}) q_7 + \mu_0 R k_{d2} q_4 - k_{pz2} q_2}{I_{p2}} \\ \frac{-c_{d1} q_8 - k_{d1} q_3}{m_1} \\ \frac{-c_{d2} q_9 - k_{d2} q_4 - c_p(q_9 - q_{10}) - k_p(q_4 - q_5)}{m_2} \\ \frac{-c_p(q_{10} - q_9) - k_p(q_5 - q_4)}{m_c} \end{bmatrix} \quad (14)$$