Mechanism Analysis of Automobile Brake Groan Based on Seven Degrees of Freedom Modal

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***Abstract*—Based on the review of researches on the vibration and noise related to automobile brake,a seven degrees of freedom nonlinear dynamics model is established,including powertrain system,brake subsystem and tire system. the numerical method is used for this study,calculation and analysis results show that the mechanism of brake groan is the stick-slip phenomenon between brake disk and friction pads.Brake groan can intensify corresponding to the increase of pressure in normal direction.**

Keywords- Modeling, Analysis ,Brake groan,Brake disk and pad,Nonlinear dynamics,Powertrain,vibration and noise.

I. INTRODUCTION

Brake system has long been known to have significant influence on the safety of vehicles,vibration and noise generated in the braking process can be divided into three main categories: judder, groan and squeal. It is shown by many experimental researches that the brake groan is most likely to occur under two circumstances. One case is that when the brake pad is released slowly and the vehicle is started from static, another case is when brake pressure is loaded gradually on the low-speed vehicle to make a completely stop. Brake groan is often happened on the vehicles with automatic transmission, occasionally it also occurred on the ones with manual transmission.

The mechanism of groan generation is the stick-slip phenomenon between brake disk and friction pads.Currently,most of researches focus on the vibration characteristics of brake system and its components.In fact,brake groan generated in starting process is not only related to the characteristics of brake system but also to the characteristics of powertrain system and tire system.Moreover,many other factors can also influence the generation of brake groan,for example,brake pedal pressure has significant influence on brake groan.In this paper,brake groan and its influence factors have been investigated in order to reveal the essence of brake groan.

II. THE SEVEN DEGREES OF FREEDOM SYMMETRICAL STRUCTURE MODEL

The seven degrees of freedom nonlinear dynamics model including powertrain system,brake subsystem and tire system is shown in Fig1 and Fig2.Parameters are defined as following.

j_d 、 j_{r1}/j_{r2} 、 j_{b1}/j_{b2} 、 j_{t1}/j_{t2} --inertia of powertrain section, rotation parts (brake disc and wheel rim and wheel) , brake assembly (brake caliper and friction pad) and tire assembly ,respectively.

θ_d 、 θ_{r1}/θ_{r2} 、 θ_{b1}/θ_{b2} 、 θ_{t1}/θ_{t2} -- angular displacement of these sections.

k_{d1}/k_{d2} -- torsion stiffness between the powertrain and rotation part.

k_{d1}/k_{d2} -- torsion stiffness between the powertrain and rotation part.

k_{t1}/k_{t2} --torsion stiffness between the rotation part and tire assembly .

k_{b1}/k_{b2} --torsion stiffness between the brake assembly and connection part.

c_{d1}/c_{d2} -- damping coefficient between the powertrain and rotation part.

c_{t1}/c_{t2} --damping coefficient between the rotation part and tire assembly.

c_{b1}/c_{b2} --damping coefficient between the brake assembly and connection part

M_d -- driving torque.

M_{b1}/M_{b2} -- braking torque.

M_{t1}/M_{t2} -- rolling resistance moment parts.

Subscripts

1 – left part

2 – right part

Let：

$$j_{r1}=j_{r2}=j_r, \quad j_{b1}=j_{b2}=j_b$$

$$j_{t1}=j_{t2}=j_t, \quad k_{d1}=k_{d2}=k_d$$

$k_{b1}=k_{b2}=k_b$, $k_{t1}=k_{t2}=k_t$

$c_{d1}=c_{d2}=c_d$, $c_{b1}=c_{b2}=c_b$

$c_{t1}=c_{t2}=c_t$

This paper considers only symmetrical situation., the system kinetic energy ,potential energy and energy dissipation functions are as follows:

$$T=\frac{1}{2}j_d\dot{\theta}_d^{\,2}+\frac{1}{2}j_r(\dot{\theta}_{r1}^{\,2}+\dot{\theta}_{r2}^{\,2})+\frac{1}{2}j_b(\dot{\theta}_{b1}^{\,2}+\dot{\theta}_{b2}^{\,2})+\frac{1}{2}j_t(\dot{\theta}_{t1}^{\,2}+\dot{\theta}_{t2}^{\,2})$$

$$U=\frac{1}{2}k_d(\theta_d-\theta_{r1})^2+\frac{1}{2}k_d(\theta_d-\theta_{r2})^2+\frac{1}{2}k_t(\theta_{r1}-\theta_{t1})^2+\frac{1}{2}k_t(\theta_{r2}-\theta_{t2})^2+\frac{1}{2}k_b\theta_{b1}^{\,2}+\frac{1}{2}k_b\theta_{b2}^{\,2}$$

$$D=\frac{1}{2}c_d(\dot{\theta}_d-\dot{\theta}_r)^2+\frac{1}{2}c_t(\dot{\theta}_r-\dot{\theta}_t)^2+\frac{1}{2}c_b\dot{\theta}_b^{\,2}$$

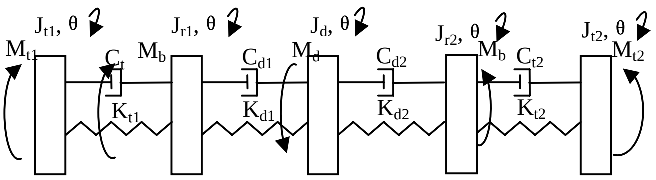

Figure1 Nonlinear dynamics model including powertrain system and tire system

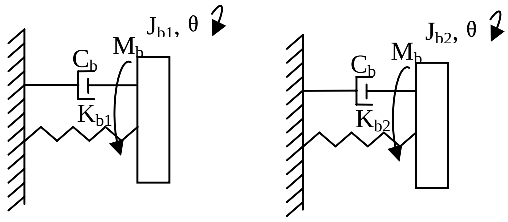

Figure2 Nonlinear dynamics model of brake subsystem

According to Lagrange's law,the following equation can be obtained:

$$j_d\ddot{\theta}_d+k_d(2\theta_d-\theta_{r1}-\theta_{r2})+c_d(2\dot{\theta}_d-\dot{\theta}_{r1}-\dot{\theta}_{r2})=M_d \qquad (1)$$

$$j_r\ddot{\theta}_{r1}-k_d(\theta_d-\theta_{r1})+k_t(\theta_{r1}-\theta_{t1})-c_d(\dot{\theta}_d-\dot{\theta}_{r1})+c_t(\dot{\theta}_{r1}-\dot{\theta}_{t1})=-M_{b1}$$
(2)

$$j_r\ddot{\theta}_{r2}-k_d(\theta_d-\theta_{r2})+k_t(\theta_{r2}-\theta_{t2})-c_d(\dot{\theta}_d-\dot{\theta}_{r2})+c_t(\dot{\theta}_{r2}-\dot{\theta}_{t2})=-M_{b2}$$
(3)

$$j_b\ddot{\theta}_{b1}+k_b\theta_{b1}+c_b\dot{\theta}_{b1}=M_{b1} \qquad (4)$$

$$j_b\ddot{\theta}_{b2}+k_b\theta_{b2}+c_b\dot{\theta}_{b2}=M_{b2} \qquad (5)$$

$$j_t\ddot{\theta}_{t1}-k_t(\theta_{r1}-\theta_{t1})-c_t(\dot{\theta}_{r1}-\dot{\theta}_{t1})=-M_{t1} \qquad (6)$$

$$j_t\ddot{\theta}_{t2}-k_t(\theta_{r2}-\theta_{t2})-c_t(\dot{\theta}_{r2}-\dot{\theta}_{t2})=-M_{t2} \qquad (7)$$

Furthermore:

$$M_{bi}=F_i^{SL}r_b \quad (\dot{\delta}_i\neq 0)$$

$$M_b=Min((N_0-a_1\dot{\delta}_i)\mu_s r_b,\left|F_i^{ST}r_b\right|)\operatorname{sgn}(F_i^{ST}) \quad (\dot{\delta}_i=0)$$

$$F_i^{SL}=N\mu(\dot{\delta}_i)=(N_0-a_1t)(\mu_d+\Delta\mu e^{-a|r_b\dot{\delta}_i|})\operatorname{sgn}(\dot{\delta}_i)$$

$$F_i^{ST}=k_b\theta_{bi}+c_b\dot{\theta}_{bi}-k_d(\theta_{ri}-\theta_{di})-c_d(\dot{\theta}_{ri}-\dot{\theta}_{di})-k_t(\theta_{ri}-\theta_{ti})-c_t(\dot{\theta}_{ri}-\dot{\theta}_{ti})$$

Where,

F_i^{SL} -friction force at the interface when relative sliding is generated between brake disc and pad.

F_i^{ST} – friction force at the interface when brake disc and pad in a sticky state .

Let $\dot{\delta}_i=\dot{\theta}_{ri}-\theta_{bi}$, $\mu(\dot{\delta}_i)=\mu_d+\Delta\mu e^{-a|r_b\dot{\delta}_i|}$,

$\mu(\dot{\delta}_i)$ ––friction coefficient related to the relative velocity between brake disc and pad

$\Delta\mu$ –– difference value between static friction coefficient μ_s and kinetic friction coefficient μ_d , $\Delta\mu=\mu_s-\mu_d$

a—shape parameter

N—normal direction force imposed at interface

r_b --effective radius of friction

N_0 --normal direction force between brake disc and pad at the begain,based on the premise that brake force is released equably

a_1 ––slope of normal direction force as relative angular velocity decreases.

The final dynamics equations of disk brake can be expreesed as follow:

$$J\ddot{\theta}+C\dot{\theta}+K\theta=M(\dot{\theta},t) \qquad (8)$$

Where,

$$\theta=(\theta_d \quad \theta_{r1} \quad \theta_{r2} \quad \theta_{b1} \quad \theta_{b2} \quad \theta_{t1} \quad \theta_{t2})^T$$

$$J = (J_d \quad J_r \quad J_r \quad J_b \quad J_b \quad J_t \quad J_t)^T$$

$$K = \begin{bmatrix} 2k_d & -k_d & -k_d & 0 & 0 & 0 & 0 \\ -k_d & k_d + k_t & 0 & 0 & 0 & -k_t & 0 \\ -k_d & 0 & k_d + k_t & 0 & 0 & 0 & -k_t \\ 0 & 0 & 0 & k_b & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & k_b & 0 & 0 \\ 0 & -k_t & 0 & 0 & 0 & k_t & 0 \\ 0 & 0 & -k_t & 0 & 0 & 0 & k_t \end{bmatrix}$$

$$C = \begin{bmatrix} 2c_d & -c_d & -c_d & 0 & 0 & 0 & 0 \\ -c_d & c_d + c_t & 0 & 0 & 0 & -c_t & 0 \\ -c_d & 0 & c_d + c_t & 0 & 0 & 0 & -c_t \\ 0 & 0 & 0 & c_b & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & c_b & 0 & 0 \\ 0 & -c_t & 0 & 0 & 0 & c_t & 0 \\ 0 & 0 & -c_t & 0 & 0 & 0 & c_t \end{bmatrix}$$

$$M = (M_d \quad -M_{b1} \quad -M_{b2} \quad M_{b1} \quad M_{b2} \quad -M_{t1} \quad -M_{t2})^T$$

III. STATE EQUATION

The state equation has to be established as to make numerical analysis, so the following are defined:

$$x_1 = \theta_d,\ x_2 = \dot{\theta}_d,$$
$$x_3 = \theta_{r1},\ x_4 = \dot{\theta}_{r1},$$
$$x_5 = \theta_{r2},\ x_6 = \dot{\theta}_{r2},$$
$$x_7 = \theta_{b1},\ x_8 = \dot{\theta}_{b1}$$
$$x_9 = \theta_{b2},\ x_{10} = \dot{\theta}_{b2}$$
$$x_{11} = \theta_{t1},\ x_{12} = \dot{\theta}_{t1},$$
$$x_{13} = \theta_{t2},\ x_{14} = \dot{\theta}_{t2},$$

And the state vector and state equation are:

$$X = [x_1 \ x_2 \ x_3 \ x_4 \ x_5 \ x_6 \ x_7 \ x_8 \ x_9 \ x_{10} \ x_{11} \ x_{12} \ x_{13} \ x_{14}]^T$$

$$\dot{X} = [\dot{x}_1 \ \dot{x}_2 \ \dot{x}_3 \ \dot{x}_4 \ \dot{x}_5 \ \dot{x}_6 \ \dot{x}_7 \ \dot{x}_8 \ \dot{x}_9 \ \dot{x}_{10} \ \dot{x}_{11} \ \dot{x}_{12} \ \dot{x}_{13} \ \dot{x}_{14}]^T$$

$$\dot{X} = \begin{bmatrix} x_2 \\ \frac{1}{J_d}(M_d - 2k_d x_1 - 2c_d x_2 + k_d(x_3 + x_5) + c_d(x_4 + x_6)) \\ x_4 \\ \frac{1}{J_r}(-M_{b1} + k_d x_1 + c_d x_2 - (k_d + k_t)x_3 - (c_d + c_t)x_4 + k_t x_{11} + c_t x_{12}) \\ x_6 \\ \frac{1}{J_r}(-M_{b2} + k_d x_1 + c_d x_2 - (k_d + k_t)x_5 - (c_d + c_t)x_6 + k_t x_{13} + c_t x_{14}) \\ x_8 \\ \frac{1}{J_b}(M_{b1} - k_b x_7 - c_b x_8) \\ x_{10} \\ \frac{1}{J_b}(M_{b2} - k_b x_9 - c_b x_{10}) \\ x_{12} \\ \frac{1}{J_t}(-M_{t1} + k_t x_3 + c_t x_4 - k_t x_{11} - c_t x_{12}) \\ x_{14} \\ \frac{1}{J_t}(-M_{t2} + k_t x_5 + c_t x_6 - k_t x_{13} - c_t x_{14}) \end{bmatrix}$$

IV. NUMERICAL CALCULATION AND ANALYSIS

As for a car which engine's output torque is 20Nm，through conversion, $M_d = 76.8Nm$ .Under the situation of brake groan,velocity is very low,pavement is slippy and rolling resistance moment can be omitted,so $M_t = 0Nm$ .Some parameters are listed in Table 1:

Table 1 simulation parameters

Inertia （kg.m²）		Torsion stiffness (Nm/rad)		Torsion damping (Nms/rad)	
sign	value	sign	value	sign	value
J_d	9.54	k_d	8788	c_d	25.05

The influence of normal direction force on brake groan was studied under a constant shape parameter at a=5.As the symmetry of the The influence of normal direction force on brake groan was studied under a constant shape parameter at a=5.As the symmetry of the structure and the consistency of the assumption of parameter,from the simulation results we can see the variational disciplinarian of θ_{r1} is the same with θ_{r2} .So are $\dot{\theta}_{r1}$ and $\dot{\theta}_{r2}$, θ_{b1} and θ_{b2}, $\dot{\theta}_{b1}$ and $\dot{\theta}_{b2}$, θ_{t1} and θ_{t2}, $\dot{\theta}_{t1}$ and $\dot{\theta}_{t2}$ .To simplify space, only the numerical results subscripted 1 of some variables appear in this paper.

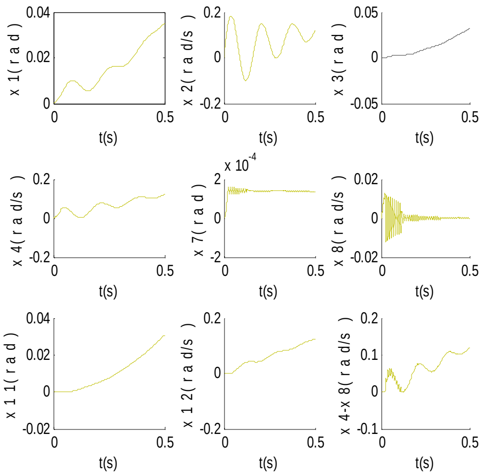

Figure3 a=5，n=100 the time history of angular displacement and angular velocity

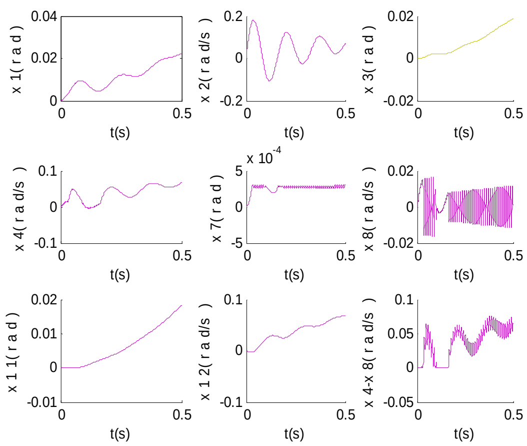

Figure4 a=5，n=200 the time history of angular displacement and angular velocity

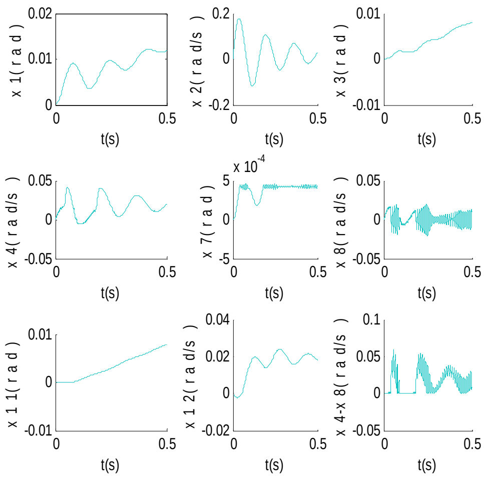

Figure5 a=5，n=300 the time history of angular displacement and angular velocity

As shown from figure3 to figure5,under constant shape parameter a=5 ,and small normal force (N=100),the vibrations of angular velocity and angular displacement were weak,the vibration angular velocity $\dot{\theta}_{b1}$ of pad was damped rapidly and the vibration condition of sticking($\dot{\theta}_{r1}-\dot{\theta}_{b1}=0$)and sliding ($\dot{\theta}_{r1}-\dot{\theta}_{b1}\neq 0$)altering was not generated..In this situation ,brake groan couldn't generate.However,under large normal force N=200 or N=300, although most vibrations of the angular displacement and angular velocity were weak, the vibration angular velocity $\dot{\theta}_{b1}$ of pad was increased gradually and there has been the vibration condition of sticking ($\dot{\theta}_{r1}-\dot{\theta}_{b1}=0$)and sliding ($\dot{\theta}_{r1}-\dot{\theta}_{b1}\neq 0$) alternating, such a situation prone to brake groan.

V. CONCLUSIONS

Through the above calculation and analysis,the paper completed the following works: A seven degrees of freedom nonlinear dynamics modal is established which includes powertrain system,brake system and tire system. The mechanism of groan generation is the stick-slip phenomenon between brake disk and friction pads.Brake groan intensifies when normal direction force increases.

ACKNOWLEDGMENT

This study is Supported by Hi-Tech Research and Development Program (863 Program) of China.

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