Moving loads and car disc brake squeal

Huajiang Ouyang
Department of Engineering, University of Liverpool, Liverpool L69 3GH, England

Disc brake vibration and squeal is a source of irritation and distraction. A squealing brake gives customers the impression of underlying quality problems of the vehicle. The warranty cost due to disc brake noise is very high and has led to much research and investment in tackling disc brake vibration and noise. Brake squeal is friction-induced vibration of immense complexity. Research in recent years has resulted in quieter brakes. However disc brake squeal still occurs frequently. Modelling disc brake systems and simulating their dynamic behaviour is an efficient and cost-effective way of correcting squealing brakes and creating new designs for improved noise performance. This paper describes the methodology for simulating disc brake vibration and predicting disc brake squeal established by the author's research group and presents numerical results against experimental results. This approach is centred on the moving-load concept. It offers a new angle in tackling the difficult disc brake squeal problem and has shown much promise.

1. INTRODUCTION

Disc brake squeal is a high-frequency noise due to friction-induced vibration. There has not been a universally accepted definition of squeal [1]. It is generally agreed that squeal is a sustained noise above about 1 kHz and it involves flexible modes of the disc. During a squeal event, there is no apparent sticking between the disc and pads. Usually a single tone dominates.

There is a wealth of literature on disc brake squeal. One cannot cover all of it within the space of this paper. Kinkaid et al. [1] recently conducted a comprehensive review on disc brake squeal. That paper is a valuable source of information for both new and established researchers in this field. Akay recently published another review paper covering a wider area of friction acoustics [2].

In order to understand how a disc brake operates, a typical disc brake is displayed below. The disc is bolted to the front axle and rotates at the same speed as the front wheel. The pads are enclosed by the carrier bracket in the horizontal plane (parallel to the disc plane) and housed in the calliper. The calliper, in a floating calliper design which is the subject of this paper, can slide fairly freely in the axial direction (normal to the disc plane) through the guide pins along the bores in the carrier bracket. When the brake is applied, the hydraulic pressure moves the piston forward which in turn pushes the inboard pad against one side of the disc. At the same time, the reaction force from the disc brings in the outboard pad, through the calliper (fingers), against the other side of the disc. This squeezing action produces a frictional torque that slows down and stops the wheel.

In an opposed-piston design, there is a piston (or pistons) housed in either side of the calliper. When the brake is applied, the hydraulic pressure brings the pistons forward and which in turn push the pads against both sides of the disc.

Friction is essential in the functioning of a disc brake. In addition, friction is also capable of generating a wide spectrum of noise, including the annoying squeal noise of 1 kHz – 20 kHz.

Why is it very difficult to study disc brake squeal and remove it? There are various difficulties. (1) The physics of friction under dynamic loading is not fully understood. It is a nonlinear phenomenon. It depends on the local normal force, the relative velocity between two mating surfaces, temperature and humidity, surface roughness which is a random variable, wear which is itself very difficult to measure and quantify, and other factors. It also depends on loading and thermal history. Of course, no models to date can

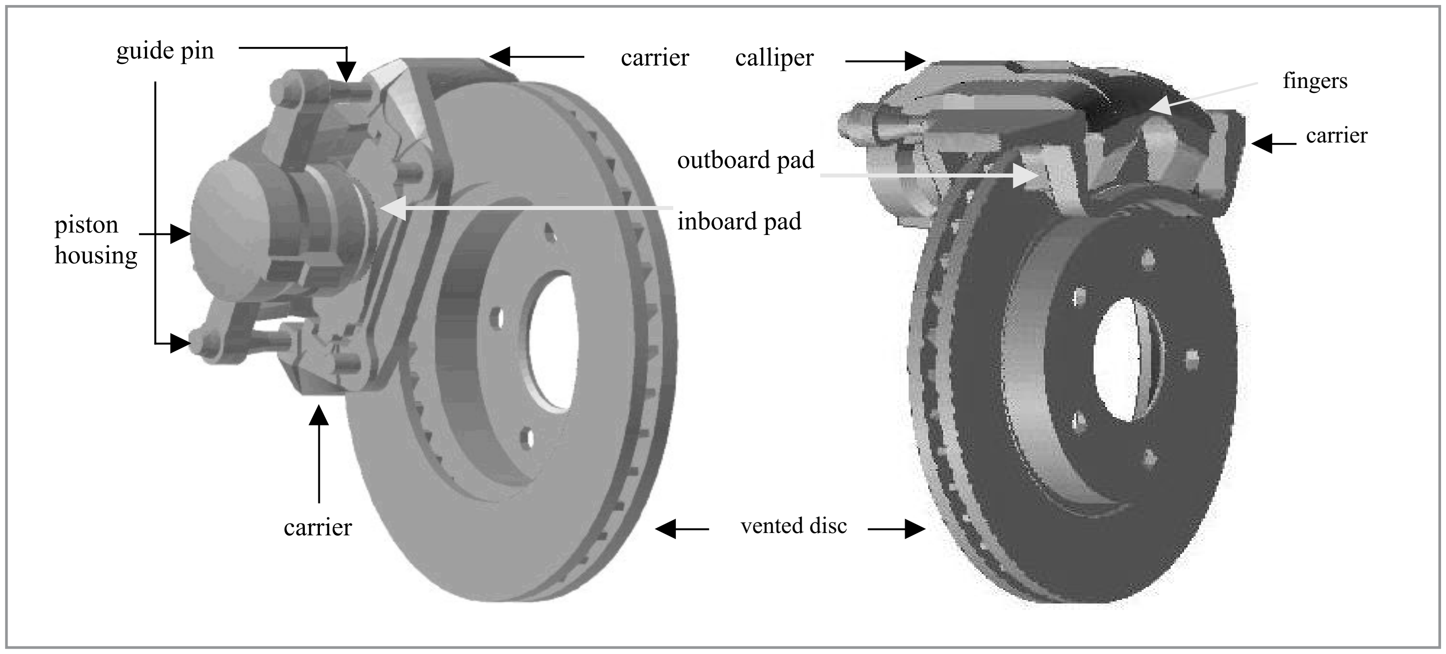

Figure 1 *Two views of a typical floating-calliper disc brake and components*

take into all the above factors. (2) The contact between the disc and the pads is dynamic and the contact interface moves temporally and spatially. The contact is closely related to friction and is equally difficult to study. (3) There are a number of contact interfaces in a disc brake system, for example, between the inboard pad back plate and the piston head, between the 'ear lugs' of the pad back plate with carrier abutments. These contacts depend on local pressure that is unknown a priori. Pads are loosely held in the calliper housing and within the carrier bracket. These boundary conditions for the brake components are highly circumstantial and again are not fully known beforehand. (4) Pads are nonlinear, temperature dependent, viscoelastic material. The braking fluid also possesses unknown nonlinear material properties sensitive to temperature variations.

Despite that the knowledge of friction and dynamic contact is still not complete and that there is a considerable uncertainty in the material properties and boundary conditions, disc brakes must be designed to satisfy customer demands. Customers believe that a squealing brake indicates poor quality and reliability of the vehicle. This perception drives up warranty cost. The warranty cost owing to the NVH issues (including disc brake squeal) was recently estimated to be about US$1 billion a year to the automotive industry in North America alone [2].

Experimental study has been a major means of investigating disc brake squeal problem, but numerical modelling and computer simulation of disc brake squeal is coming to be preferred. Compared with the experimental study, numerical modelling and simulation has the following advantages: (1) it is much quicker and cheaper to build numerical models and run simulations, (2) conceptual ideas can be numerically tested to evaluate the merits and pitfalls before a physical model is built, (3) there are virtually unlimited kinds of structural modifications to be studied. Consequently an optimised design may be found. Ouyang et al. [3] recently reviewed these theoretical methods.

As mentioned before, squeal is a result of friction-induced vibration. The friction, acting in the tangential plane of two contacting surfaces, has been observed to excite large-amplitude transverse (normal to the contact interface) vibration. The hypotheses of the ways whereby the tangential friction excites transverse (and trivially the tangential) vibration are called squeal/friction mechanisms. There are a number of squeal mechanisms. They were reviewed in [1] and are not covered in this paper. A particular squeal mechanism is presented in this paper.

2. FIVE MODELLING ISSUES

2.1. THE MOVING LOAD CONCEPT

The disc rotates past the non-rotating pads. As such, the pads are mating with different areas of the disc at different times. Because squeal tends to occur at low disc speeds, it is natural for people to omit this moving contact. It is shown in this paper that this relative motion between the disc and the pads may be important and its effects on the noise behaviour should be known. The author and his colleagues recently put forward a moving load model for disc brakes and the numerical results indicated that the dynamic stability (or instability) was indeed affected by the relative gross velocity between the disc and the pads [4, 5]. The moving-load concept will be explained in the next section.

2.2. CONTACT ANALYSIS

Due to the friction forces acting at the disc and pads interface, the interface pressure distribution is highly uneven and biased towards the leading edge of the pad back plate [6]. Furthermore, part of the disc and pads interface even loses contact [6]. It has long been speculated that the interface pressure distribution affects the squeal propensity. The current industrial approach contains a two-step analysis procedure: a nonlinear, static contact analysis to determine the interface pressure distribution followed by a complex mode analysis to determine the dynamic instability represented by the positive real parts of the complex eigenvalues [5, 7-10]. This strategy is also kept in this paper.

2.3. CONTACT INTERFACE MODEL

Under pressure and friction, there are a large number of small, localized areas undergoing plastic deformation and high temperature variation at disc and pads interface. The disc is energised at this interface. Thus it is very important

noise notes

HYDERABAD NIGHTFLIGHTS

Incidence of sensory neural deafness is on a rise among residents in areas surrounding Begumpet airport, Hyderabad, thanks to the noise pollution from the flights even during the night time. There has definitely been a rise in cases of sensory neural deafness because of the noise pollution, senior consultant, ENT surgeon, claims Dr Venu Gopal Reddy who works at Krishna Institute of Medical Sciences. People exposed to more than 90 decibels of sound for two or three hours every day suffer damages in the hair cells in cochlea (organ for hearing). The damage of these cells is permanent and causes deafness, he said. In areas surrounding the airport about two to five kilometres, the level of sound is about 90 decibels for about eight hours per day. Moreover, most international flights operate only in the night and the sound is unbearable during landing and take off, president of Save Secunderabad Environment Committee, captain S Vijay Kumar said. The permissible standards are 55 db for normal working place and 90 db for eight hours in factories.

to model it. In Liles and Nack's method, a number of springs are installed at the disc and pads interface, representing the contact stiffness [7-10]. Yuan [11] put forward a general formulation for any type of finite element for the contact interface. Ouyang et al. [5] used a thin layer of solid elements with pressure-dependent Young's moduli.

2.4. FRICTION LAWS

Many people may assume that sophisticated friction laws should be used in analysing disc brake squeal. Kinkaid et al. [1] found that sophisticated friction laws had not been accepted by the brake community. The reason is simple. Faced with such a complicated system as disc brakes, sophisticated friction laws may not enhance accuracy. On the contrary, the extra complexity and accumulated computing errors may overwhelm the limited benefit. It was also recognised that a constant friction coefficient in Coulomb's friction law was sufficient to bring about dynamic instability [12]. Therefore, the simple Coulomb's friction law is used in this paper.

2.5. SQUEAL MECHANISMS

It is crucial to use a right squeal mechanism in the dynamic model of a disc brake system. However, a consensus on the right squeal mechanism has not been reached yet. An extended version of North's idea of friction couple [12] is used in this paper.

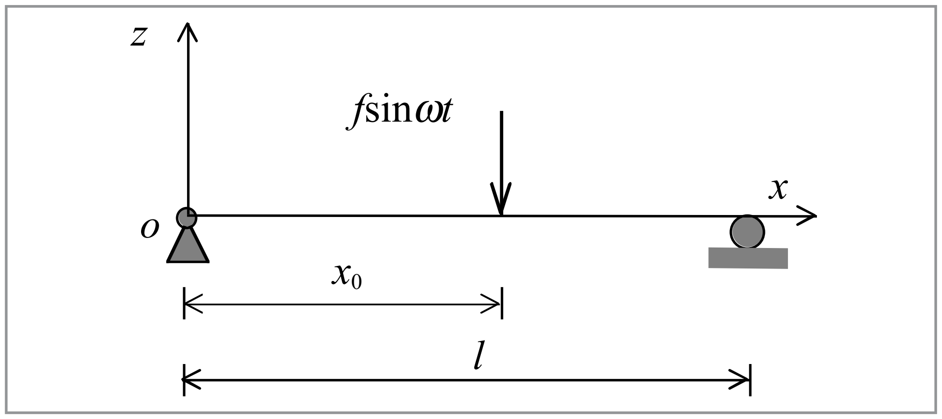

Figure 2. A simply-supported beam under a harmonic load

3. VIBRATIONS DUE TO MOVING LOADS

This section consists of three parts. In the first part, the solution of the vibration of a beam under an external harmonic load is described. The second part introduces the moving-load concept by means of a beam excited by a moving mass. Finally, the vibration of a circular plate subjected to a moving mass is discussed.

3.1. FORCED VIBRATION OF A SIMPLY-SUPPORTED BEAM

The equation of undamped, forced vibration of an Euler beam of uniform material and cross-section, as shown in Figure 2, is

$$\rho A\frac{\partial^2 w}{\partial t^2}+EI\frac{\partial^4 w}{\partial x^4}=-f\sin(\omega t)\delta(x-x_0) \tag{1}$$

where ρ is the density, A the area of the beam cross-section, I the second moment of this area, E the Young's modulus and x_0 is the spatial coordinate where the harmonic load is acting, whose amplitude is f and excitation frequency ω, δ is the Dirac delta function. It represents a concentrated force located at $x = x_0$ and has the following property

$$\int_0^l y(x)\delta(x-x')\mathrm{d}x=y(x') \quad (0<x'<l) \tag{2}$$

where y is an arbitrary continuous function of x.

The natural frequencies of this beam are

$$\omega_j=\sqrt{\frac{EI}{\rho A}}\left(\frac{j\pi}{l}\right)^2 \quad (j=1,2,\ldots) \tag{3}$$

The transverse vibration w of the beam can be expressed as a sum of its modes as

$$w(x,t)=\sum_{j=1}^{\infty}\psi_j(x)q_j(t)$$

(4)

where $q_j(t)$ is the modal coordinate for the j-th mass-normalized mode $\psi_j(x)$ of the beam, which is

$$\psi_j(x)=\sqrt{\frac{2}{\rho Al}}\sin(\frac{j\pi x}{l}) \quad (j=1, 2, \ldots)$$

(5)

Substituting equation (4) into (1), multiplying it by $\psi_j(x)$ $(i = 1,2,...)$ and then integrating the resultant equation yields

$$\frac{d^2 q_i}{dt^2}+\omega_i^2 q_i=-\sqrt{\frac{2}{\rho Al}}\sin(\frac{i\pi x_0}{l})f\sin\omega t$$

$(i=1,2,...)$

(6)

According to linear vibration theory, whenever the excitation frequency is very close to any one of the beam's natural frequencies, that is, $\omega \approx \omega_i$, then the amplitude of $w(x, t)$ becomes very large and resonance happens.

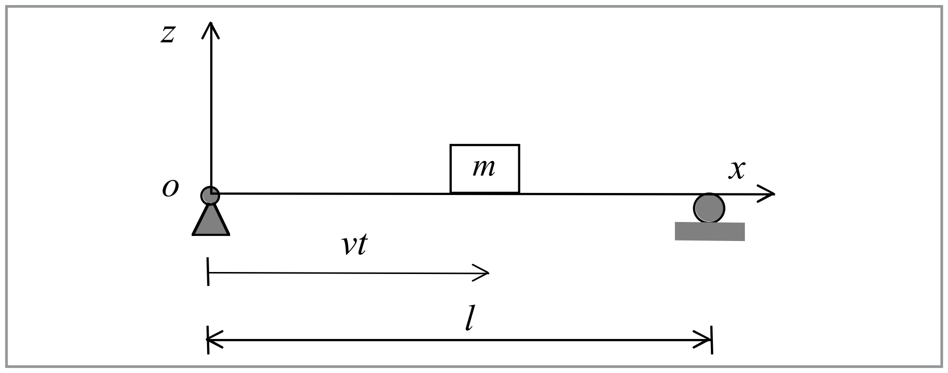

Figure 3. A simply-supported beam subjected to a moving mass

3.2. SELF-EXCITED VIBRATION UNDER A MOVING LOAD

Next, the vibration of the same beam excited by a moving (frictionless) mass of m is considered. This structure is shown in Figure 3. The mass starts to move along the beam from $x=0$ at $t=0$ at constant speed v. It reaches $x = vt$ at an arbitrary time t.

The mass m exerts a transverse inertial force of

$$m\frac{d^2 u}{dt^2},$$

where $u(t) = w(vt, t)$ is the instantaneous deflection of the beam in contact with m, and a constant weight W. It is implied that there is no separation between the mass and the

noise notes

ENFORCING THE NOISE ABATEMENT ACT

From the "Jamaica Gleaner". The Christiana police must be commended for firmly enforcing the Noise Abatement Act on New Year's Eve. At the receiving end of the law this time were members of a church who attended a Watchnight service and gospel concert. The gathering filled the town's taxi stand and spilled over into the main road, creating yet another problem for those who may not have wished to be unduly disturbed or hindered by the gathering and joyful noise of the faithful. The event was hosted by the Spaldings Ministers' Fraternal and so came with the blessings of the church organisation. Organisers wisely advised patrons that they would be conforming with the instructions from the authorities as the Noise Abatement Act was a law of the land and should be observed by all, including the church. One minister present noted that the church should set the example for others to follow. It is just a pity that the example was not set before the intervention of the police to instruct that the concert be wound down by 2:00 a.m., well beyond the specifications of the law.

beam even when the transverse vibration of the beam becomes very large. It is easy to see that

$$\frac{du}{dt} = \frac{\partial w}{\partial t} + v\frac{\partial w}{\partial x}.$$

By analogy to equation (1), the equation of transverse vibration of this beam is

$$\rho A\frac{\partial^2 w}{\partial t^2} + EI\frac{\partial^4 w}{\partial x^4} = -(W + m\frac{d^2 u}{dt^2})\delta(x - vt)$$
$$= -[W + m(\frac{\partial}{\partial t} + v\frac{\partial}{\partial x})^2 w]\delta(x - vt)$$

(7)

Following the same steps described in Section 3.1, equation (7) can be converted to

$$\frac{d^2 q_i}{dt^2} + \omega_i^2 q_i = -\sqrt{\frac{2}{\rho Al}}W\sin(\frac{i\pi vt}{l}) -$$
$$\frac{2m}{\rho Al}\sum_{j=1}^{\infty}\{[\frac{d^2}{dt^2} - (\frac{j\pi v}{l})^2]q_j\sin(\frac{j\pi vt}{l}) +$$
$$\frac{2j\pi v}{l}\frac{dq_j}{dt}\cos(\frac{j\pi vt}{l})\}\sin(\frac{i\pi vt}{l})$$

(8)

By examining the first term on the right-hand side of equation (8), one can conclude (comparing it with equation (6) if one is unsure) that when

$$\frac{i\pi v}{l} = \omega_i \text{ or } v = \frac{l\omega_i}{i\pi} = \frac{i\pi}{l}\sqrt{\frac{EI}{\rho A}}$$
$$(i = 1, 2, ...)$$

(9)

resonance will happen. This is a distinct result from the (conventional) forced vibration represented by equation (1). It means that a constant moving load may cause resonance when the moving speed of this constant load is *right*. While under a non-moving load, resonance can only occur when the external load is oscillatory in time and at the same time $\omega \approx \omega_i$. Resonance of the type dictated by equation (9) is referred to as single-mode resonances.

What is more striking, though less apparent, is that resonance can also occur when any one of the four following expressions holds,

$$\frac{(i \pm j)\pi v}{l} \approx \omega_i \pm \omega_j \quad (i, j = 1, 2, ...; i \neq j)$$

(10)

The above formula can be derived when considering the following trigonometric identities

$$\cos(\frac{j\pi vt}{l})\sin(\frac{i\pi vt}{l}) =$$
$$\frac{1}{2}[\sin\frac{(i + j)\pi vt}{l} + \sin\frac{(i - j)\pi vt}{l}],$$
$$\sin(\frac{j\pi vt}{l})\sin(\frac{i\pi vt}{l}) =$$
$$\frac{1}{2}[\cos\frac{(i - j)\pi vt}{l} - \cos\frac{(i + j)\pi vt}{l}]$$

and going through a mathematically-involved procedure. For details, one can check [13]. This indicates that a moving mass may excite resonances involving two modes (thus called combination resonances). This phenomenon will not happen for the steady vibration excited by a non-moving load with a single excitation frequency.

There can be other resonance conditions. Since they have to be found using rather tedious numerical methods, they are not shown here.

One can see that moving loads can bring about interesting phenomena not found in conventional vibrations due to non-moving loads. Another way of looking at a moving load problem is to

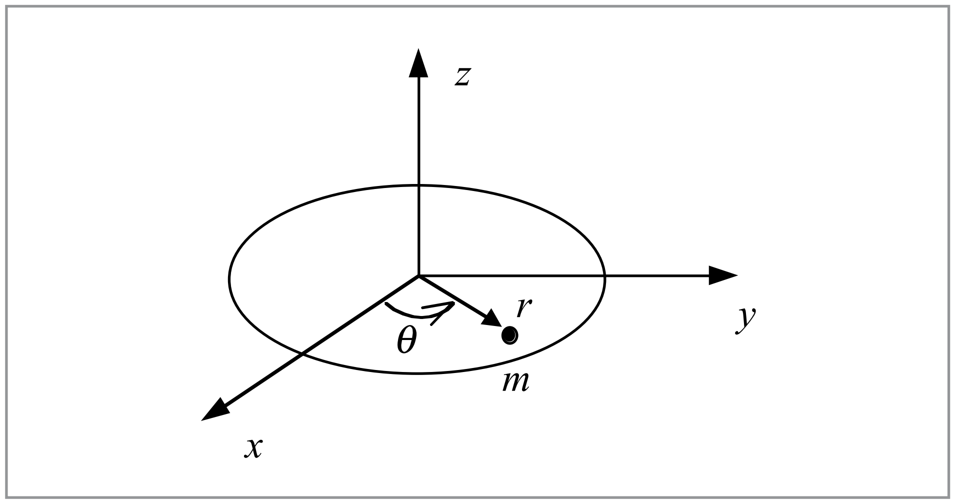

Figure 4. Circulate plate and its coordinate system

move some terms on the right-hand side of equation (8) to the left. In so doing, one can get three sets of terms on the left-hand side as follows

$$[1-\frac{2m}{\rho Al}\sin^2(\frac{i\pi vt}{l})]\frac{\mathrm{d}^2 q_i}{\mathrm{d}t^2},$$
$$-\frac{2m}{\rho Al}\frac{2j\pi v}{l}\sin(\frac{i\pi vt}{l})\cos(\frac{i\pi vt}{l})\frac{\mathrm{d}q_j}{\mathrm{d}t},$$
$$[1+\frac{2m}{\rho Al}(\frac{j\pi v}{l})^2]\sin^2(\frac{i\pi vt}{l})\omega_i^2 q_i \quad .$$

(11)

Apparently the moving mass adds a time-periodic mass (in the first set of terms in equation (11)), damping (the second set) and stiffness (the third set) to the beam. In particular, damping becomes negative as often as become positive. It is no wonder that the vibration of the beam can be destabilised by a moving mass.

Moving loads are very common in engineering systems. Examples include vehicle-bridge interaction, computer disc and reader vibration, vibration and chatter in machining, wood saws, train wheel/rail squeal. Fryba's monograph [14] detailed analytical solutions of many simple moving load problems.

3.3. VIBRATION OF A CIRCULAR PLATE SUBJECTED TO A ROTATING MASS

The equation of the transverse motion of a thin, circular plate (in a cylindrical coordinate system) subjected to a frictionless mass m rotating around the disc surface at a constant speed Ω is

$$\rho h\frac{\partial^2 w}{\partial t^2}+D\nabla^4 w=-\frac{1}{r}\delta(r-r_0)\delta(\theta-\Omega t)\times$$
$$[W+m(\frac{\partial}{\partial t}+\Omega\frac{\partial}{\partial\theta})^2 w]$$

(12)

where m is located at $(r_0, 0)$ at time zero. h is the thickness and D the flexural rigidity of the plate. The partial differential operator is

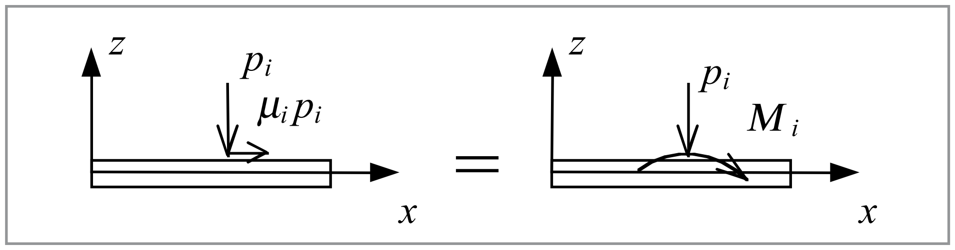

Figure 5. Graphical explanation of the idea of North's friction couple

$$\nabla^4=(\frac{\partial^2}{\partial r^2}+\frac{1}{r}\frac{\partial}{\partial r}+\frac{1}{r^2}\frac{\partial^2}{\partial\theta^2})^2$$

Following the similar but more complicated procedure to that of Sections 3.1 and 3.2, the same qualitative conclusions [13] can be drawn on Ω about single-mode and combination resonances as those in Section 3.2.

4. MODEL OF A DISC BRAKE SYSTEM

4.1. SEPARATE MODELS FOR THE DISC AND THE STATIONARY COMPONENTS

A brake disc has a large amount of cyclic symmetry and is approximated as an annular, thin plate. The pads, calliper, carrier bracket and guide pins (together known as the stationary components) possess complicated geometry and must be represented by a large number of finite elements. Lumped-parameter models of small numbers of degrees-of-freedom will not predict squeal frequencies well enough or give unstable modes. This separate treatment of the rotating disc (analytical model) and the stationary components (finite element model) was put forward in [15] and can greatly facilitate the implementation of the moving-load concept in a disc brake model [4]. The contact pressure at the disc and pads interface is now represented by vector **p** of nodal forces p_i (i=1, 2, ..., number of pads' nodes in contact with the disc), which are the normal dynamic forces acting onto both surfaces (the disc and pads interface) of the disc from the pads.

4.2. FRICTION LAW AND SQUEAL MECHANISM

Each normal force p_i acting on the surface of the disc from the pads produces a tangential friction force $\mu_i p_i$ from Coulomb's friction law, where μ_i is the dry friction coefficient at node i on the disc and pads interface. Each friction force acting on the disc surface in the tangential direction can be considered to produce a bending couple about the central plane of the plate (where the equilibrium is established), shown in Figure 5, as

$$M_i = \mu_i p_i h/2 \tag{13}$$

This particular way of incorporating friction as a non-conservative force is based on North's idea of the follower force [12].

4.3. VIBRATION OF THE DISC

The equation of the transverse vibration of the disc, approximated as an annular, thin plate, is [5]

$$\rho h\frac{\partial^2 w}{\partial t^2} + c\frac{\partial w}{\partial t} + D\nabla^4 w = -\frac{1}{r}\sum_i \{p_i(t)\delta(\theta-\theta_i-\Omega t) + \frac{\partial}{r\partial\theta}[M_i(t)\delta(\theta-\theta_i-\Omega t)]\}\delta(r-r_i) \tag{14}$$

The δ functions in equation (14) indicates that p_i (and M_i) initially sits at the polar coordinate of (r_i, θ_i) and then moves to the new polar coordinate of $(r_i, \theta_i + \Omega t)$ at time t. c is viscous damping of the disc. All the p_i (and M_i) must be summed up as external forces acting onto the disc from the two pads. p_i now reflects the dynamic force (inertia, damping and elastic) of the stationary components. Equation (14) may be understood by comparing it with equations (12).

4.4. VIBRATION OF THE STATIONARY COMPONENTS

The equation of motion of the stationary components using the finite element method is

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{f}(t) \tag{15}$$

where **M**, **C** and **K** are the mass, damping and stiffness matrices, and **x** the displacement vector of the stationary components respectively. The dot over a symbol in equation (15) represents the derivative with respect to time.

For a disc brake, the excitation comes from the friction forces acting at the moving disc and pads interface. These moving friction forces and the dynamic normal forces are internal to the whole brake system. That is why squeal is self-excited vibration. There is no external force involved. One can regroup the displacement and force vectors in equation (15) into two separate sets: $\mathbf{x}_p$ for nodes of the pads in contact with the disc and $\mathbf{x}_o$ for all the other nodes. Then one gets

$$\mathbf{f}^T = [\mathbf{f}_p^T, \mathbf{f}_o^T], \quad \mathbf{x}^T = [\mathbf{x}_p^T, \mathbf{x}_o^T] \tag{16}$$

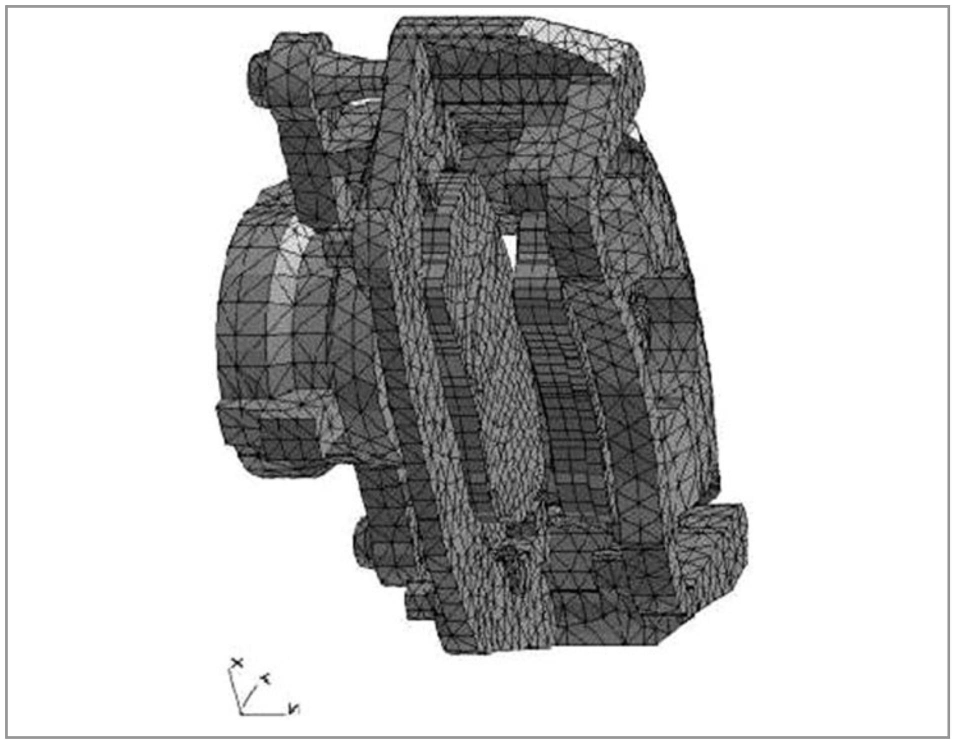

Figure 6. Finite element mesh of the stationary components

Table 1. Material data of the stationary components

Components	E(GPa)	ν	ρ(kg.m^{-3})
Calliper	187.63	0.3	7100
Carrier	170	0.3	7564
Others	210	0.3	7850

where

$$\mathbf{f}_p^T = [0, \mu_1 p_1, p_1, 0, \mu_2 p_2, p_2, ...],\ \mathbf{f}_o^T = 0 \qquad (17)$$

and superscript 'T' stands for vector (matrix) transpose. It also follows that

$$\mathbf{f}_p = \begin{bmatrix} 0 & 0 & . & . & . \\ \mu_1 & 0 & . & . & . \\ 1 & 0 & . & . & . \\ 0 & 0 & & & \\ 0 & \mu_2 & & & \\ 0 & 1 & & & \\ . & . & . & & \\ . & . & & . & \\ . & . & & & . \end{bmatrix} \mathbf{p} \qquad (18)$$

From equations (16)–(18), one can derive a solution of equation (15) $\mathbf{x}(t)$ in terms of $\mathbf{p}(t)$ in principle. Now select all the w-displacements for those nodes of the pads in contact with the disc and form a new vector $\mathbf{w}$. Then in theory one can obtain [4, 5]

$$\mathbf{w}(t) = \mathbf{A}(\lambda)\mathbf{p}(t) \qquad (19)$$

where $\mathbf{A}$ is a square matrix and λ is a complex eigenvalue of the whole brake system.

4.5. COMPLEX EIGENVALUE ANALYSIS

From equation (14), one can solve $w(r, \theta, t)$ in terms of $\mathbf{p}$. The displacement continuity at the disc and pads interface requires that the normal displacements of the pads and the instantaneous transverse deflection of the disc to be equal. Due to the relative

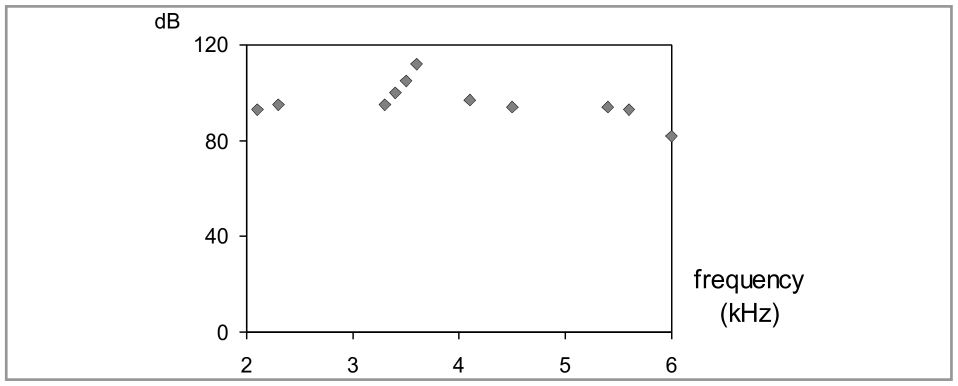

Figure 7. Experiments-established noise frequencies

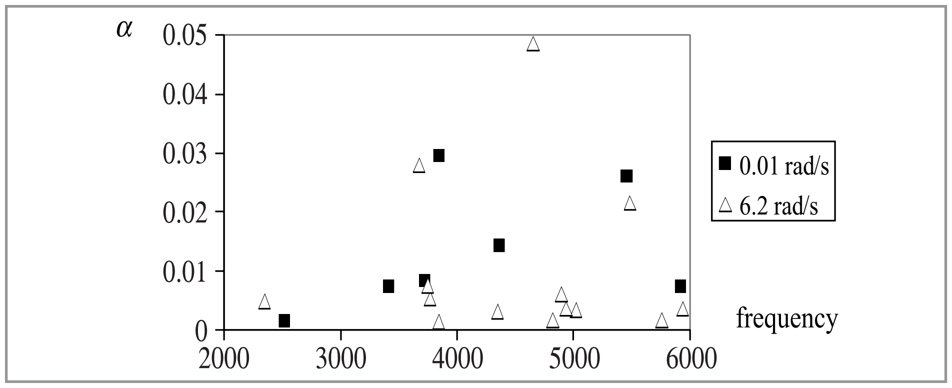

Figure 8. Predicted noise indices versus frequencies

motion between the disc and the pads, this requirement means

$$\mathbf{w} = \{w(r_1, \theta_1 + \Omega t, t), w(r_2, \theta_2 + \Omega t, t), ...\}^T = \mathbf{B}(\lambda)\mathbf{p}(t) \qquad (20))$$

where $\mathbf{B}$ is a square matrix. By combining equations (19) and (20), one gets

$$[\mathbf{A}(\lambda) - \mathbf{B}(\lambda)]\mathbf{p} = 0 \qquad (21)$$

which is a nonlinear eigenvalue formulation.

A recent paper also presented a linear complex eigenvalue formulation for the dynamic instability analysis of disc brake squeal with much larger matrices [16].

5. RESULTS AND DISCUSSION

A typical disc brake, as shown in Figure 1, is considered here. The finite element model of the stationary components has many thousand degrees-of-freedom, as shown in Figure 6.

The disc has the following dimensions and properties: its inner and outer radii are 0.045m and 0.133m, $E = 120$GPa, $\rho = 7200$kh·m^{-3}, Poisson's ratio $\nu = 0.211$, $h = 0.012$m.

The Young's modulus of the friction material depends on piston line pressure and is in the range of 5.4 – 10.8 GPa. It is treated as an orthotropic material (but $\nu = 0.3$). The material properties of other stationary components are listed in Table 1.

The noise frequencies with level above 80 dB obtained from experiments are illustrated in Figure 7. No piston line pressures or disc speeds are recorded in the experiments, which are from an external source.

Yuan [17] defined $\alpha = \frac{\sigma}{\sqrt{\sigma^2 + \omega^2}}$ as noise index, where *s* and *w* are the real and the imaginary parts of a complex eigenvalue. The numerical results at two speeds of Ω=0.01 rad/s (very slow sliding) and Ω=6.2 rad/s are shown in Figure 8.

Since no knowledge of disc speeds is available in the experiments from an external source, predicted results at two different disc speeds are given. The predicted unstable frequencies roughly cover all the noisy experimental frequencies, though the real parts do not match the noise levels so well. The correlation in the two sets of frequencies would be even better if the numerical frequencies were shifted down by about 200Hz. This should be a reasonable adjustment, considering that the disc in the model is bolted to a rigid ground while it is bolted to an elastic suspension in the experiment.

Another phenomenon to be observed is that the predicted complex eigenvalues at two different disc speeds are quite different. This supports the incorporation of the moving-load concept into the modelling of disc brake systems. To what extent the moving loads affect squeal intensity and occurrence rate in disc brakes needs further investigations.

It is recognised that the magnitude of the real parts of the complex eigenvalues indicates the growth rate of the particular motions in a linear model and does not necessarily indicate noise intensity of squeal frequencies [7], which can only be calculated from a transient analysis of the corresponding nonlinear model. However, the complex eigenvalue

noise notes

GLOVES OFF

The owner of Gold's Gym on Main Street in Vacaville is asking to open its doors one hour earlier, at 4 a.m., on weekdays. The City Council first dealt with the issue in mid-July when a resident complained that the noise from gym patrons talking and car doors slamming woke their family up in the early morning hours. After hours of discussion, the council decided the owner, Rick Martindale, could apply to modify its hours with the planning commission. Again facing the commission, both the owner of the adjoining building and local residents at Main and Cernon streets will oppose the proposed gym hours. They say the foot traffic, patrons taking up street parking and excessive noise are problems for many neighbours in the area. But Martindale says the hours are necessary for those who commute to the Bay Area. The adjoining neighbour to the gym says he will not support the proposal unless a six-foot high masonry wall is constructed along the Cernon Street side of the parking lot. Martindale, who contends that noise problems won't be heightened with the new hours, has proposed his own modifications to the parking lot to reduce noise levels. Various analyses of noise in the vicinity of the gym's parking lot show conflicting results. Martindale requested a report by Bollard & Brennan, Inc., a consultant in acoustic and noise control, that shows the gym's parking lot has no impact on noise measurements at the nearest residence. A conflicting report commissioned by neighbour Jo-Anna Camilleri-Olin shows that "some activity is causing a rise in the ambient noise in the early morning hours" at the gym. This report was done by Environmental Safety Associates, based in Florida.

analysis provides a conservative approach for assessing the system's dynamic instability. If no positive real part is present in all the complex eigenvalues, then the vibration will not grow into limit-cycle oscillation and squeal can be avoided [11].

CONCLUSIONS

This paper provides a detailed account of the disc brake squeal problem and the moving-load concept. It gives a brief description on how moving loads affect the dynamic instability of a simple beam and a real disc brake system. Numerical results of unstable frequencies agree reasonably well with experimental squeal frequencies. Numerical complex eigenvalues at two different disc speeds display significant differences in both the real parts and the imaginary parts. They indicate that consideration of moving loads really make a big difference in the predicted dynamic behaviour of a disc brake system

ACKNOWLEDGEMENT

The work presented in this paper is a result of collective effort of the author's research group at the University of Liverpool. Special mention is given to Professor J E Mottershead who led the disc brake squeal research for a number of years, Dr Q Cao who made Figures 1 and 6 and obtained some numerical results and Dr S James who provided modal testing results. Financial supports from the Engineering and Physical Sciences Research Council (grant number GR/L91061) and TRW Automotive are gratefully acknowledged. Thanks also go to the Visiting Scholarship from the State Key Laboratory of Structural Analysis of Industrial Equipment of China.

REFERENCES

1. Kinkaid, N. M., O'Reilly, O M and Papadopoulos, P. (2003) Automotive disc brake squeal: a review. *J. Sound Vib.*, Vol. 267, No. 1, pp. 105–166.

2. Akay, A. Acoustics of friction. (2002) *J. Acoust. Soc. Am.*, Vol. 111, No. 4, pp. 1525–1548.

DESTINED FOR DEAFNESS

Hearing loss in adulthood may be programmed at birth, and short people may be particularly prone to hearing problems, according to new research. Researchers in Sweden base these conclusions on the thrifty phenotype hypothesis. According to this theory, events during fetal life, such as malnutrition or exposure to alcohol or nicotine, may cause disease in adulthood. The scientists say a malnourished fetus will make metabolic adaptations which may become permanently programmed and persist throughout life. In this study, the researchers looked for a solution to sensorineural hearing loss, or SNHL – the most common type of hearing loss. It is usually permanent and not treatable by either medications or surgery. For their study, the Swedish researchers assessed the hearing of 479 men aged 20 to 64, who were exposed to noise in their jobs, and 500 randomly selected males born in 1974. Researchers took into account factors like weight, height, exposure to noise, heredity for hearing loss and other medical disorders including drug use. Among the randomly selected men, shortness was found twice as often in those with SNHL as in men with normal hearing. SNHL was also associated with a positive heredity for hearing loss but not with noise exposure. Among the workers, short workers had worse hearing than expected by age – three-times more often than taller workers. Short workers were also 12-times more often taking drugs. The researchers also found that older short men with high blood pressure has significantly worse hearing, but among tall men, blood pressure had no effect on hearing and the influence of age was less pronounced. The authors say the thrifty phenotype hypothesis is applicable to SNHL. They suggest a low level of growth hormone before birth may lead to a reduced number of cells at birth. This could result in short stature and earlier onset of age-related diseases. The authors say before their study, high blood pressure and high cholesterol were commonly thought to cause SNHL. But they say they now consider those "superficial markers."

3. Ouyang, H., Nack, W., Yuan, Y. and Chen F. (2003) *On automotive disc brake squeal. Part II: Simulation and analysis*. SAE Paper 2003-01-0684.

4. Ouyang, H., Mottershead, J. E. and Li, W. A moving-load model for disc-brake stability analysis. (2003) *ASME Journal of Vibration and Acoustics*, Vol. 125, No. 1, pp. 53–58.

5. Ouyang, H., Cao, Q., Mottershead, J. E., Treyde, T. and Cartmell, M. P. (2002) Modelling and simulation of the vibration and squeal of a disc brake. *Proc. of International Conference on Vehicle Brakes* (Braking 2002, Ed. D Barton and B Shilton, Professional Engineering Publishing Ltd., Bury St Edmund, UK, pp. 43–52.

6. Tirovic, M. and Day, A. J. (1991) Disc brake interface pressure distribution". Proc. IMechE, *J. Auto. Eng.*, Vol. 205, pp. 137–146.

7. Liles, G. D. (1989) *Analysis of disc brake squeal using finite element methods*. SAE Paper 891150.

8. Nack, W. V. (2000) Brake squeal analysis by the finite element method. *Int. J. Vehicle Des.*, Vol. 23, No. 3/4, pp. 263–275.

9. Lee, Y. S., Brooks, P. C., Barton, D. C. and Crolla, D. A. (2003) A predictive tool to evaluate disc brake squeal propensity Part 1: The model philosophy and the contact problem. *Int. J. Vehicle Des.*, Vol. 31, No. 3, pp. 289–308.

10. Blaschke, P., Tan, M. and Wang, A. (2000) *On the analysis of brake squeal propensity using finite element method*. SAE Paper 2000-01-2765.

11. Yuan, Y. (1996) An eigenvalue analysis approach to brake squeal problem. *Proc. 29th ISATA Conf. Automotive Braking Systems*, Florence, Italy.

12. North, N. R. (1976) *Disc brake squeal*. Proc. IMechE, C38/76, pp.169–176.

13. Ouyang, H. and Mottershead, J. E. (2001) Unstable travelling waves in the friction-induced vibration of discs. *Journal of Sound and Vibration*, Vol. 248, No. 4, pp. 768–779.

14. Fryba L. *Vibration of Solids and Structures under Moving Loads*, Noordhoff: Groningen, 1972.

15. Ouyang, H., Mottershead, J. E., Brookfield, D. J., James, S. and Cartmell M. P. (2000) A methodology for the determination of dynamic instabilities in a car disc brake. *Int. J. Vehicle Des.*, Vol. 23, No. 3/4, pp. 241–262.

16. Cao, Q., Friswell, M. I., Ouyang, H., Mottershead, J. E. and James, S. (2003) Car disc brake squeal: theoretical and experimental study. *Proc. 5th Int Conf on Modern Practice in Stress and Vibration Analysis*, Ed. M P Cartmell. Trans Tech Publications, pp. 269–276.

17. Yuan Y. (1995) A study of the effects of negative friction-speed slope on brake squeal. *Proc. 1995 ASME Des. Eng. Conf.*, Boston, Vol. 3, Part A,

noise notes

WIND FARM NOISE

Of those questioned, living in the vicinity of Bear Downs wind farm, Padstow, Cornwall, 93% said their lives had been adversely affected by the noise, with 70% reporting sleep problems and anxiety.