Squeal noise of friction material with groove-textured surface: an experimental and numerical analysis

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## Abstract

In this work, an experimental and numerical study is performed to understand squeal generation and suppression of a pad-on-disc friction system. Several friction material specimens having various orientation degrees of grooves cut on their surfaces are tested. Numerical studies using the methods of complex eigenvalue analysis and dynamic transient analysis are conducted to simulate the experimental process with the finite element software ABAQUS. Both experimental and numerical results show that surface modifications of friction material specimens have a significant influence on the squeal instability: cutting a 45° or 90° groove on the material surface can significantly reduce squeal noise, cutting a 135º groove just reduces squeal noise moderately and cutting a 0º groove cannot reduce squeal noise. Moreover, the contact pressure distributions for the original surface and modified surfaces are studied to provide a physical explanation of the noise phenomenon. The major finding that friction-induced noise can be reduced by means of suitable structural modifications of the contact interface is expected to have important and much wider applications.

*Key words:* Friction-induced squeal; Surface modification; Experiment; Numerical analysis.

1. **Introduction**

Automotive brake squeal is still an unsolved noise problem caused by friction-induced vibration in spite of many years of investigation [1]. Brake squeal, defined by its high frequency (usually above 1 kHz) and great squeal intensity (above 78 dB (A)), is a nuisance to the customers and contributes to noise pollution [2]. Engineers and researchers have made many attempts to prevent or reduce brake squeal in brake design [3].

In the past years, extensive research effort has been made to understand and eliminate the friction-induced squeal noise [3-9]. Ibrahim [3] gave a review of friction-induced vibration and noise. A rich number of friction-induced noise problems were reviewed by Akay [4]. Kinkaid et al. [5] gave a comprehensive review and bibliography of works on disc brake squeal. Ouyang et al. [6] reviewed the numerical methods used in the studies of disc brake squeal. Papinniemi et al. [7] provided a review of the analytical, experimental and numerical methods used for the investigation of brake squeal and outlined some of the challenges facing brake squeal research. As pointed out by Oberst and Lai [8], because the nature of brake squeal is transient, fugitive, non-repeatable and sensitive to many factors, the problem of predicting, reducing and eliminating brake squeal propensity remains as challenging as ever. As a consequence, although some mechanisms can successfully explain some squeal phenomenon, the cause of squeal is still not known completely.

In general, many kinds of techniques and methods are performed to solve the brake squeal problem, changing damping, stiffness, or friction parameters of a brake, for example, can reduce brake squeal at some level [9-18]. Hochlenert [9] described a nonlinear stability analysis of a 12-degree-of-freedom realistic disc brake model and studied the effect of the stiffness and damping of the brake pad lining on squeal. Butlin and Woodhouse [10, 11] examined the sensitivity to variations in contact parameters of the friction system though different models to find the best one for friction-induced squeal. Sinou et al. [12] investigated brake squeal through the experimental approach and found that the damping and the brake friction coefficient had a significant effect on the brake system stability. Coudeyras et al. [13] found that the higher friction coefficient, higher piston pressure, higher contact stiffness lead higher vibration amplitudes for the brake system. Cantone and Massi [14] found that while the system would be stabilized with a homogenous distribution of damping, while the squeal propensity would increase with a non-uniform repartition of damping. Ouyang et al. [15] studied dry-friction-induced instability caused by non-smooth stick-slip motion of the slider over the surface of a disc, and they found that the normal pressure, rotating speed, rigidity, damping and stiffness all had a big influence on the stability of the system. Duffour and Woodhouse [16-18] studied the instability of systems with a frictional point contact by experimental method and simulated the behavior of the generic systems through the stability criterion in terms of transfer functions. They found that the damping, the contact compliance, the coefficient of friction had significant effect on the stability of friction system. Among the methods for squeal suppression, the method of structural modification is considered to be one of the most effective ways [13-28]. Eriksson et al. [19, 20] studied the effect of the brake pad surface topography on the generation of squeal noise. They found that the size of brake pad surface plateaus had a significant influence on the generation of squeal noise, and relatively small contact plateaus on the brake pad surface tended to generate strong squeal noise than pads with large plateaus. Eadie et al. [21] discovered that changing friction characteristics by using a ‘third body’ could avoid squeal noise. Sherif et al. [22] studied the relationship between surface topography and squeal noise generation and put forward the concept of ‘squeal index’ which can be used to describe the establishment and disappearance of squeal. Vayssière et al. [23] concluded that there was a really strong correlation between the disc geometry and squeal initiation. Hammerström et al. [24] studied the effect of a spiral-shaped modification of the brake disc surface topography on brake squeal and they found that squeal instability could be significant reduced. Massi et al. [26, 27] investigated brake squeal through experimental and numerical method, they found that if squeal events occurred, there were several cracks and material exfoliations on the surface layer of the pad material, however, the contact surfaces without squeal after braking were smooth and compact. They also reported that making structural modifications to the disc could avoid the growth of vibration of the brake system and hence reduce brake squeal. Nouby et al. [28] investigated the influences of several parameters, namely the back-plate thickness, the chamfer, the distance between two slots, the slot width, and the angle of the slot on the pads. They found that brake squeal could be reduced by cutting the chamfer on both sides of the friction material and by using different slot configurations. Oberst and Lai [29, 30] introduced different slots on the surface of pads, and studied the influence of the geometrical designs of brake pads on brake squeal, and also found that the squeal instability could be significantly changed. Wagner et al [31] presented an efficient way to separate the automotive brake disc eigenfrequencies by a structural optimization, which had been considered as a good measure against brake squeal. Abd Rahman [32] attempted to prevent the drum brake squeal by means of modifications on pads. The above studies showed that structure modifications of friction system have a significant effect on the squeal instability.

In recent years, numerical studies relying on the finite element method have been widely used in investigating friction-induced squeal noise. This method can be used to explain test results, prepare for the design of experiment, simulate structural modifications and investigate innovative ideas. It plays an important role in understanding squeal noise [33-36]. Kung [33] studied the effect of the stiffness and friction coefficient on brake squeal through the numerical method. Bajer [34] studied the brake squeal through the complex eigenvalue analysis by using the finite element software ABAQUS. Kang et al. [35, 36] studied both mode-coupling and negative slope squeal mechanisms in a disc brake system through the numerical method. The popularity of numerical methods is owing to the fact that they offer much faster and more cost-effective solutions than experimental methods, and they can predict squeal noise performance at early stage of design development. Based on the experimental study of squeal, the numerical study was performed [37-39]. Meziane et al. [37] investigated friction-induced vibration by using a beam-on-beam friction system through experimental and numerical methods. Weiss et al. [38] made an experimental and numerical study of squeak in hip endoprosthesis systems. Wang et al. [39] studied squeal noise of a ball-on-flat system numerically based on friction tests and found that cutting T-500-250 grooves on the flat surface could suppress the squeal efficiently.

There are two big categories of the numerical methods of friction-induced vibration which are complex eigenvalue analysis in the frequency domain and transient dynamic analysis in the time domain [40]. The complex eigenvalue approach linearizes the friction system at its static steady sliding states. By analysing the eigenvalues of the friction system around a steady sliding state, the complex eigenvalue approach permits detection of the stability limit of the system. If the results predicted through complex eigenvalue analysis agree with the experimental results well, it proves that the region of friction-induced instability can be detected with a linear hypothesis in the contact zone [41, 42]. On the other hand, the non-linear effects of the contact may not be neglected when investigating the instability evolution, once squeal is developed. The transient dynamic analysis can take into account the non-linear aspects of the friction system. This methodology allows determination of displacements, velocities, accelerations and forces during vibration [43]. More often than not, the two numerical methods are used separately. Recently, several numerical works have performed both types of analyses which showed that they were complementary [44-46]. AbuBakar et al. [44, 45] and Massi et al. [46] predicted brake squeal through complex eigenvalue analysis and transient dynamic analysis. It was found that the complex eigenvalue analysis could detect the stability limit of the system and predict the squeal frequencies of the friction system, and the dynamic transient analysis could determine the time histories of acceleration, velocities and displacements. However, the dynamic transient analysis takes much longer time to run. Therefore, for studying the generation of squeal of the friction system, the complex eigenvalue analysis and the dynamic transient analysis are complementary.

In this paper, the effect of surface modifications on the characteristics of squeal noise is studied by using groove-modified friction material (pad) surfaces. An experimental and numerical study is conducted to study squeal through a pad-on-disc model. This device has the advantage of reduced complexity of the real brake system, and at the same time possesses the major features of a real disc brake so that its fundamental dynamic behaviour can be compared with that of a real disc brake. Five different pad specimens whose surfaces are respectively smooth (refers to as untreated original pad surface), and cut with grooves at 0°, 45°, 90°, and 135° orientations, are tested. Subsequently, a numerical study is conducted to simulate the experimental process and study the effect on squeal of the five kinds of specimens being tested. In the numerical study the complex eigenvalue analysis is used to validate the model created by the finite element software (ABAQUS), while the dynamic transient analysis is used to investigate the evolution of squeal in time domain. The numerical results are then compared with the squeal events observed in the experiments, showing fairly good agreement. Finally, a possible mechanism is proposed to explain squeal reduction by grooves shown in experimental and numerical studies.

In this investigation, a Coulomb friction model is adopted. Although it is simplistic, the numerical results are shown to agree with the experimental results in a qualitative manner; in particular, supporting the experimental finding of which groove feature demotes friction-induced noise. For more sophisticated friction models, please refer to, for example [47]

1. **Experimental procedure and numerical modelling**
2. *Experimental procedure*

A simplified physical system is used to investigate squeal: the pad-on-disc setup. This set up has the advantage of reduced complexity of a real brake system in the form of a well-defined contact area and a simple [friction](app:ds:friction) [pair](app:ds:pair). Nevertheless, the dynamics of the pad-on-disc setup has characteristic features of more complex systems. Therefore, this set up is well-suited for the development of numerical representations of contact and understanding of squeal generation and suppression.

The schematic view of the experimental setup is illustrated in Fig.1. It consists of a friction material specimen (pad) in sliding contact with a rotating flat disc, housed in a tribological testing system and attached to a signal acquisition and analysis system. The disc is fixed to the base which is mounted on the rotational motion device. The friction material specimen is fixed to the upper holder which is held by a chuck with a CETR DFH-50 2-D force sensor attached (sensitivity: 0.025 N; measuring range: 5~500 N). At the start of the test, the moving stage moves down slowly to allow the upper holder to bring the pad into contact with the disc with a constant normal load. Then, the disc is driven into rotational motion against the specimen. A KISTLER 8688A50 3-D acceleration sensor (sensitivity: 100 mV/g; measuring range: ± 50 g; frequency response: 0.5 Hz~5 kHz) is mounted on the upper holder to measure the vibration of the friction system, and a MTG MK250 microphone (sensitivity: 50mV/Pa; measuring range: 15~146 dB; frequency response: 3.5~20000 Hz) is located near the friction interface to measure the noise signal. The friction force, vibration acceleration and noise signals are synchronously measured and analysed during the tribological test.

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Fig. 1. Schematic view of the experimental setup.

The tribological testing parameters are as follows: normal load of 100 N, disc speed of 6.28 rad/s, and testing time of 1800 s. Before testing, the surfaces of each pad and disc specimens are cleaned with acetone.

Compacted graphite iron (~3.5 wt% C, ~2.5 wt% Si and~1.5 wt% Mn) with microhardness of HV0.05 240 kg/mm2 and elastic modulus (*E*) of 158 GPa is used as the disc material. The disc is cut from a real brake disc to the diameter of 25 mm and the thickness of 3 mm. The pad specimens are cut from a real disc brake pad friction material (density: 1±0.5 g/cm3; elastic modulus: *E*≤1×103 MPa; hardness: HR 50~90), with size of 10×10×15 mm. In the surface-modified pad specimens, four kinds of straight grooves, i.e. a 0° groove (perpendicular to the local relative tangential velocity of the disc), a 45° groove (measured from the leading edge of the pad in the clock wise direction), a 90° groove-textured (parallel to the disc's tangential velocity) and 135° groove-textured (inclined to the trailing edge) are cut on the pad specimen surfaces, as shown in Fig. 2. The optical images of the groove-textured pad surfaces are shown to illustrate the detailed external appearance, as revealed in Fig. 3. The width and depth of the groove are 1 mm and 0.3 mm, respectively. The microphone is placed 10 mm away from the disc centre in the horizontal direction. Considering the random nature of squeal generation, five samples are prepared for each kind of surface, and each sample is tested at least three times to ensure good repeatability in this study. To ensure the reliability of the experimental results, all the tests are conducted under the same atmospheric conditions with strictly controlled relative humidity of 60%±10% and ambient temperature of around 25°C.

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Fig. 2. The definition of the groove orientation on pad surface.

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Fig. 3. Optical images of the modified pad surfaces.

1. *Numerical model*

Fig. 4(a) shows the simplified 3-dimensional finite element model of the experimental system presented in Fig. 1. The numerical model is created according to the geometry of the experimental setup, which includes a base, a disc (active device), a pad specimen (passive specimen), an upper holder and a connection part. All the material parameters of the parts used in the numerical calculation reflect those of the real experimental setup and their values (Mass density: *ρ*; Young’s modulus: *E*; Poisson ratio: *υ* are given in Table 1. There is always frictional contact between the pad and the disc. Considering the rotational speed is very low and normal load is small during the tests and the pad specimens are small too, the friction-generated heat has little effect on the squeal, so it is ignored in the numerical study. The load and boundary conditions of the finite element model are shown in Fig. 4(b): the bottom of the disc is rigidly constrained except in the circumferential direction, which is used to apply the velocity boundary condition. For the pad specimen, the normal load is applied on the top of the connection part in the y-c. Those places marked in blue on the top of the connection part are constrained in the x and z directions but are free in the y direction. All the constraint conditions are consistent with the real experimental setup except the part of threaded connections (as marked in three places in Fig. 4a), which are simulated by a tie constraint.

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Fig. 4. Finite element model of the experimental system: (a) finite element mesh; (b) load and boundary conditions.

Tab 1 Material parameters.

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Parameter | Connection part | Upper holder | Pad | Disc | Rotation motion device |
| *ρ*kg/m3 | 7800 | 7200 | 2100 | 7200 | 7800 |
| *E* GPa | 172 | 158 | 3 | 158 | 172 |
| *υ* | 0.3 | 0.27 | 0.25 | 0.27 | 0.3 |

Based on the experimental study, finite element models for pad specimens of the smooth surface, 0°, 45°, 90° and 135° groove-textured surfaces are created. These modifications can change the friction behaviour of the pads to affect the decoupling of the mode shapes and to shift the squeal frequencies. The groove patterns are illustrated in Fig. 5. For the five models, all the element type used is C3D8R (8-node linear brick with reduced integration) with advancing front meshing algorithm which can efficiently improve the quality of the mesh. Compared with other element types, C3D8R can efficiently reduce computational time and ensure higher accuracy. Table 2 lists the characteristics of the mesh, in which the FE mesh is found to have converged.

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Fig.5. Five kinds of the pad surfaces.

Tab 2 Mesh features of the finite element model.

|  |  |  |
| --- | --- | --- |
| Features of the friction system | No. elements | No. nodes |
| Smooth surface | 13799 | 17727 |
| 0º groove-textured surface | 17562 | 21917 |
| 45º groove-textured surface | 18768 | 23185 |
| 90º groove-textured surface | 17562 | 21917 |
| 135º groove-textured surface | 18768 | 23185 |

*2.3 Modal Analysis of the experimental setup*

For the modal parameters of the experimental setup, the single components are too small to be tested with an accelerometer, especially for the disc and the pad specimen. So instead of testing these small components separately, a modal test on the whole testing machine is carried out. The upper holder is hit by an instrumented hammer in the X direction and the accelerometer on the upper holder records the vibration. The measured frequency is 1194 Hz. The boundary conditions applied to the FE model are shown in Fig.4 (b). The main frequency for the FE model is 1298 Hz.

The 8.8% difference between the tested and predicted main frequencies is probably due to the required simplifications of the FE model. Especially when the glued connection between the disc and the base is set as a tie constraint, a relative higher value of frequency results from neglecting the elasticity of the disc. The threaded connections above the upper holder and pad specimen (please see Fig. 4(a)) are also taken as tie constraints which will also increase the value of the predicted frequency.

1. **Experimental results and discussion**

As shown in Fig. 6, equivalent continuous A-weighted sound pressure level during each 100-s test duration is evaluated for the five kinds of pad surfaces. Any noise signal that has a dominant frequency of over 1 kHz and sound pressure level of above 78 dB is taken as a squeal event. Throughout the whole test, the sound pressure level of the 90° groove-textured surface stays at a significantly lower level compared with the other surfaces, below 78 dB, suggesting no squeal being generated from this surface. This indicates that the 90° groove-textured surface modification of the pad has a great potential in squeal suppression. In the initial stage, the sound pressure levels generated by both the 45° and 135° groove-textured surfaces are found to be relatively lower than those of the smooth surface and 0° groove-textured surface, at only about 83 dB. With the increase of testing time, the squeal level of those surfaces increases. It goes up to nearly 105 dB for the smooth surface and a 0° groove-textured surface and about 99 dB for the 135° groove-textured surface. However, for the specimen of 45° groove-textured surface throughout the whole test, the sound pressure level stays at a relatively lower level, which indicates that the 45° groove-textured surface modification of the pad also has a good potential in squeal suppression.

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Fig. 6. Equivalent sound pressure level for the five surfaces.

Fig. 7 shows the power spectrum density (PSD) of the sound pressure throughout the test for the three surfaces. Four dominant frequencies of the squeal can be found for the smooth surface: they are 1120 Hz, 2225 Hz, 2750 Hz and 3337 Hz. For the 0° and 135° groove-textured surfaces, three dominant frequencies can be found: they are 1150 Hz, 2287 Hz and 3425 Hz and 1250 Hz, 2525 Hz and 3787 Hz, respectively, which indicates that this modification of pad surface can suppress one of the squeal frequency in the range of 2300 Hz to 3000 Hz. However, for the 45° and 90° groove-textured surfaces, only one dominant frequency (1112 Hz and 1262 Hz, respectively) is detected during the whole test, which indicates that this modification can suppress squeal, particularly above 2000 Hz. Thus, changing the contact surface by cutting a 45° or 90° groove on the pad surface has a significant effect on the stability of the friction system and on suppressing some high-frequency squeal.

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Fig. 7. PSD of sound pressure during the whole test.

To further investigate the effect of surface texture on squeal noise, this paper analyses the vibration acceleration in the friction direction (tangential direction) of the five kinds of surfaces during the steady stage of 17001701 s, as shown in Fig. 8. For the smooth surface and 0° groove-textured surface, the vibration acceleration signals show an obvious limit cycle, and the amplitudes of vibration acceleration are very large, nearly at 200 m/s2. However, for the 45° and 135° groove-textured surfaces, although the vibration acceleration signals also show an obvious limit cycle, the amplitudes of vibration acceleration are smaller, especially only less than 100 m/s2 for the 45°groove-textured surface. In contrast, almost no obvious limit cycle can be observed for the vibration acceleration signals of the 90° groove-textured surface. Therefore, surface modification by cutting a 45° or 90° groove on the pad surface plays a crucial role in suppressing the generation of high-amplitude vibration, and consequently the generation of squeal.

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Fig. 8. Tangential vibration acceleration in the steady stage for the five pad surfaces under testing.

1. **Complex eigenvalue analysis and discussion**

*4.1 Complex eigenvalue analysis*

Nowadays, the complex eigenvalue analysis is proved to be a useful tool for studying the instability of braking systems and revealing the mechanism of friction-induced vibration. This numerical method computes the system’s complex eigenvalues in which friction leads to asymmetric terms in the stiffness matrix. The real and imaginary parts of the complex eigenvalues are the decay rates and frequencies of the system, respectively [6]. Here, in order to perform the complex eigenvalue analysis using ABAQUS 6.10, the methodology of the complex eigenvalue analysis is described briefly. A complex eigenvalue problem is solved using the subspace projection method in ABAQUS, thus a natural frequency extraction analysis must be performed first in order to determine the projection bases.In the complex eigenvalue analysis, the system is firstly linearized and approximated to have linear stiffness and equivalent linear viscous damping. Then the equations of motion can be written as follows [6]:

(1)

where {} is the acceleration vector, {} is the velocity vector, {} is the displacement vector, is the mass matrix, which is symmetric and usually positive definite. is the damping matrix, which includes friction-induced contributions. is the stiffness matrix, which is asymmetric due to friction. The corresponding characteristic equations of Eq. (1) can be written as follows:

(2)

where is an eigenvalue and is the corresponding eigenvector. Both eigenvalues and eigenvectors may be complex. This system is symmetrized by dropping the damping matrix and asymmetric contributions to the stiffness matrix . In this case becomes a pure imaginary number, , and the eigenvalue problem can be written as follows:

(3)

where is the symmetric part of the stiffness matrix, is a frequency of the system. This symmetric eigenvalue problem is solved using the subspace iteration eigen-solver. The next step is that the original matrices are projected in the subspace of real eigenvectors and given as follows:

, , , (4)

where , Now the reduced eigenvalue problem is expressed in the following form:

(5)

Finally, the complex eigenvectors of the baseline system can be obtained by：

(6)

An eigenvalue of Eq. (5) is in complex form of , where is the real part of , denoted as *Re*(), which indicates the stability of the system and is the imaginary part of , denoted as *Im*(), which indicates the modal frequency. When becomes positive, the system is unstable and squeal noise may occur. The effective damping ratio is deﬁned as:

(7)

If the effective damping ratio is negative, the system is unstable and has a tendency to radiate squeal. The complex eigenvalue analysis is the preferred method in the vibration and noise research community, since the complex eigenvalue analysis can provide rapid solutions of unstable vibration.

In the present paper, ABAQUS’s complex eigenvalue analysis capability is applied to validate the finite element model, by comparing the dominant unstable frequency from the numerical model and the squeal frequency from the experimental test. The effect of these five pad surface modifications on the unstable frequencies of friction systems can be observed by using complex eigenvalue analysis. Four main steps should be followed when making a complex eigenvalue analysis [43, 44].

* 1. *Complex eigenvalue analysis results and discussion*

Complex eigenvalue analysis is undertaken to study the instability of the friction system at different friction coefficients since friction is the main cause of instability, which causes the stiffness matrix to become asymmetric in Eq. (2) [6]. In the complex eigenvalue analysis, the contact formulation is small sliding with a penalty method, and the friction formulation is the simple Coulomb's law, but material and structural damping is ignored. The result shows that for the five friction systems, the friction coefficient of 0.1 is thus the critical value for the generation of unstable vibration at the dominant frequency of about 1346 Hz.

The complex eigenvalue analysis is conducted to predict which kind of surface feature would demote squeal noise. Although the real part of a complex eigenvalue indicates the propensity of squeal but does not necessarily indicates noise level, it is the most widely used method for evaluating squeal propensity and it would be good to know how it fares in studying this particular friction-induced noise problem. In this particular study, the smallest real part of a complex eigenvalue turns out to match the lowest level of measured noise. However, the magnitudes of the real part do not always correlate with the noise levels, as can be seen from Fig. 9

Fig. 9 shows the positive real parts of the complex eigenvalues versus the friction coefficient () and the unstable mode of these five different kinds of pad surfaces against the rotating disc. With the increase of the friction coefficient, the real parts of the complex eigenvalues become positive, which indicates that the system is unstable. As the friction coefficient increases continuously, the value of the positive real parts increases, which indicates the increase of the instability level of the systems. For the five friction systems, the positive real part of the 0º groove-textured surface is found to be greater than those of the other four friction systems, which indicates it is more unstable and more likely to generate squeal noise. While the positive real part of the 90º groove-textured surface is found to be smallest and thus it is most stable compared with the other four systems. Comparison between the values of the positive real part of 45º groove-textured surface and of 135º groove-textured surface with that of the smooth surface does not show a significant difference. The unstable modes of the five systems are found to mainly take place on the upper holder in its flexural mode. This mode is associated with the flexural motion of the upper holder carrying the pad sliding along the x direction. To further study the influence of the grooves cut on the pad surfaces on the squeal noise, the dynamic transient analysis is carried out in the next section.

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Fig. 9. Positive real parts of complex eigenvalues and unstable mode of five friction systems versus the friction coefficient ().

Comparing the complex eigenvalue analysis results of the smooth surface, 0º groove-textured surface and 45º groove-textured surface with the squeal test results, it is observed that one squeal frequency (1120 Hz) measured in the experiments is not very far from the predicted frequency (1346 Hz). The discrepancy between the numerical and experimental results is probably due to the required simpliﬁcations and approximations necessary for the construction of the numerical model, especially when the threaded connections, a number of node on the top surface of the connection part and all nodes on the bottom surface of the rotational motion device (compare Fig. 1 and Fig. 4) are set to a tie constraint, a higher value of frequency results from neglecting the flexibility at these locations.

It is noted that the complex eigenvalue analysis finds no frequencies above 2000 Hz, which may also be due to the required simpliﬁcations and approximations necessary for the model construction which increases the stiffness of the friction system. Therefore, a numerical solution of the complete nonlinear system has to be performed in addition to the stability analysis to estimate the non-linear behaviour of the solution. The dynamic transient analysis is performed in the next section.

Overall, the finite element model created in this work seems to reflect the dynamic behaviour of the real experimental system, and thus can be used in the dynamic transient analysis, to reveal the effect of pad surface texture on the friction-induced vibration and noise.

1. **Dynamic transient analysis results and discussion**
2. *Dynamic transient analysis*

Some researchers determined the squeal instability of friction system in time domain by using transient dynamic analysis, and concluded that transient dynamic analysis can be used to understand the physics behind the friction-induced unstable vibration [6]. The methodology of the transient dynamic analysis is described briefly. In the process of dynamic transient analysis, the equation of motion is as follows [6]:

(8)

At the beginning of a time increment, accelerations are computed as follows:

(9)

where is the internal force vector and is the applied external force vector including the calculated contact forces. The superscript refers to an arbitrary time instant.

The velocity and the displacement of the system are expressed in the following equations:

(10)

(11)

where superscripts and refer to the mid-increment values before and after *t*. Since the above central difference operator is not self-starting because of the mid-increment of velocity, the initial values at time =0 for velocity and acceleration need to be defined. In this case, both values are set to zero as the disc is stationary at time =0.

As opposed to the implicit dynamic integration, the explicit dynamic integration does not need a convergent solution before attempting the next increment step. Each increment step is so small that its stability limits for is bounded in terms of the highest frequency () involved:

(12)

It can be found that is very small. That is why the explicit dynamic transient analysis takes a long computing time.

1. *Results and discussion*

For the dynamic transient analysis, the time history of the normal load and the disc velocity are applied to reflect the real operating conditions, as shown in Fig. 10. At first the normal load is applied and gradually increased until time when it reaches the maximum of 100 N and is kept at this constant level. The disc starts to rotate at and the speed gradually increases up to time when the rotational speed reaches the maximum of 6.28 rad/s and becomes constant too. In this investigation, a constant friction coefficient, =0.35 is used, which was obtained from an experimental study. The contact formulation is finite sliding with a penalty method, and the friction formulation is the simple Coulomb's law, while material and structural damping is ignored, just as has been done in the complex eigenvalue analysis. The observation point (as marked in Fig.4) of the vibration acceleration is located on the surface of the upper holder, which is where the acceleration signals are recorded in the experimental study.

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Fig. 10. Time history of the normal load and the rotational speed of the disc in the dynamic transient analysis.

For the smooth surface and 0º groove-textured surface, the simulated time in dynamic transient analysis is first set to 2 s and the corresponding computing time is about 80 hours. Fig. 11 shows the simulation results of vibration acceleration in tangential direction in time domain. It can be seen that for the two kinds of friction systems, visible limit cycles appear with the highest acceleration magnitude occurring at =0.04 s and this kind of behavior lasts until 2 s. This type of dynamic behaviour is consistent with that observed in experimental tests, as shown in Fig. 8, which indicates that cutting a 0º groove on the pad surface could not prevent the generation of squeal. Furthermore, for the smooth surface and 0º groove-textured surface friction systems, the amplitude of the vibration acceleration do not change significantly during the 0.04 to 2 s period.

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Fig.11. Simulation results of vibration acceleration for smooth surface and 0º groove-textured surface friction systems in the tangential direction in time domain.

Considering the fact that the amplitude of the vibration acceleration during 0.04 to 2 s period is almost the same, the simulated time would be set to 0.1 s to reduce computing time. Before this, the vibration acceleration response in time domain is converted into the frequency domain for both the 2 s and 0.1 s durations, as shown in Fig.12. When the simulated time is set to 2 s, the main unstable frequencies of the smooth surface friction system are 1321.9 Hz and 3914.3 Hz, which are not far from the measured squeal frequencies of 1120 Hz and 3337 Hz, respectively. And the unstable frequency of the 0º groove-textured surface friction system is predicted to be 3286.5 Hz, which is also not far from the measured squeal frequency of 3425 Hz. For the case of simulated time of 0.1 s, the unstable frequencies of the smooth surface friction system are predicted to be 1299.7 Hz and 3919 Hz, which are almost the same as the ones obtained in the case of simulated time of 2 s. Moreover, the main unstable frequency of the 0º groove-textured surface is 3439.1 Hz, which is also almost the same as the unstable frequency obtained from the simulated time of 2 s. Therefore, the computing time can be set to 0.1 s to reduce computing time in this study.

C:\Users\wangxiaocui\Desktop\Submit to Int J Mech Sci-0214\Figures\Fig. 13(a).tif C:\Users\wangxiaocui\Desktop\Submit to Int J Mech Sci-0214\Figures\Fig. 13(b).tif

Fig. 12. Predicted unstable frequencies in the periods of 0 to 2 s (a) and 0 to 0.1 s (b) for smooth surface and 0º groove-textured surface friction systems.

Fig. 13 shows the simulation results of vibration acceleration in tangential direction in time domain. It can be seen that for the smooth surface and 0º groove-textured surface friction systems, visible limit cycles appear with the highest acceleration magnitude occurring at =0.04 s. This type of dynamic behaviour is consistent with that observed in experimental tests, as shown in Fig. 8, which indicates that cutting a 0º groove on the pad surface could not prevent the generation of squeal.

However, for the 45º groove-textured surface, the magnitude of predicted vibration acceleration is significantly reduced, which agrees with the experimental results (see Fig. 8) well. The friction system does not generate unstable vibration in this condition. Therefore, cutting a 45º groove on the pad surface is a good modification to suppress squeal instability.

The numerical results of the 90º groove-textured surface show that the magnitude of predicted vibration acceleration is even smaller than that of the 45º groove-textured surface, which is very consistent with the experimental results (see Fig. 8). Consequently, cutting a 90º groove on the pad surface can significantly suppress squeal instability.

For the 135º groove-textured surface, limit cycles can be seen but not as obvious as those of the smooth surface and groove-textured surface, which also agrees with the experimental results well, as shown in Fig. 8. Corresponding, the magnitude of the vibration acceleration decreases to some extent but not as significantly as that of the 45º groove-textured surface and 90º groove-textured surface. Therefore, cutting a 135º groove on the pad surface can suppress squeal instability moderately but not as effectively as cutting a 45º groove. Overall, cutting a 90º groove seems most effective, according to both the experimental and numerical analyses. To further investigate what exactly limits the amplitudes of five surfaces, the limit cycles of the velocity against displacement for the smooth and 90º groove-textured surfaces are presented in Fig.14. Visibly, the limit cycle of the smooth surface is much bigger than that of the 90º groove-textured surface.

C:\Users\Administrator\Desktop\Fig.14.tif

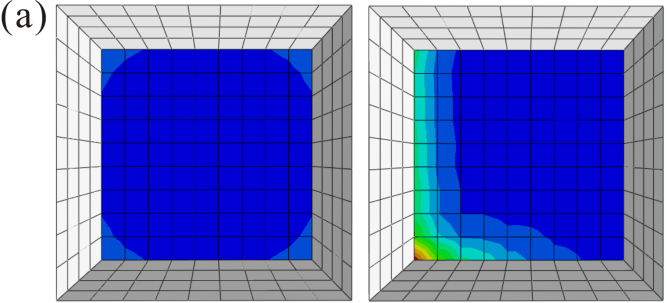
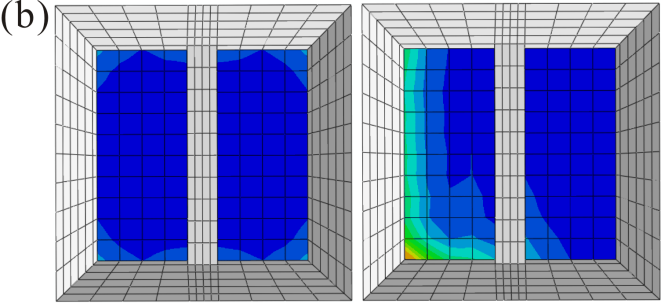
Fig. 13. Simulation results of vibration acceleration for the five different friction systems in tangential direction in time domain.

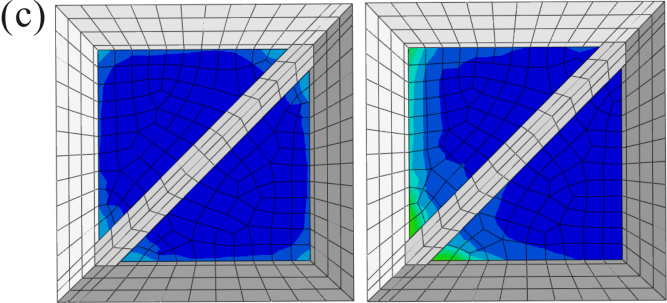
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Fig. 14. Velocity against displacement for the smooth and 90º groove-textured surface.

To further investigate the phenomenon presented above, and propose a possible physical explanation on why the pads having 45º groove and 90º groove respectively show capability of reducing vibration acceleration amplitudes, the contact pressure distributions for the smooth, 0º groove-textured, 45º groove-textured, 90º groove-textured and 135º groove-textured surfaces at the speed of 0 rad/s (left) and 6.28 rad/s (right) are provided, respectively, as shown in Fig. 15. For the study of contact pressure distributions, the criterion of a good modification which could reduce squeal noise efficiently is higher uniformity of pressure distributions by seeking a greater contact area but lower pressures [48, 49]. From Fig. 15, it can be found the contact pressure distributions are symmetrical for all the five kinds of surfaces when the disc is at rest i.e. the speed is 0 rad/s. However, the differences are the contact pressure is distributed symmetrically along the groove of the groove-textured surfaces, while the pressure is concentrated at the four points of the smooth surface.

When the disc speed is 6.28 rad/s, the contact pressure distributions of the five kinds of surfaces are no longer symmetric and the highest pressure occurs at the leading edge of the surfaces. For the smooth surface, the contact pressure is concentrated at the leading edge and the maximum contact pressure (in red colour) is located mostly at the leading point, the contact pressure at the trailing edge of the surface is almost zero. For the 0º groove-textured surface, the existence of the groove does not interrupt the contact pressure concentration of the leading edge, and thus the contact pressure distribution does not show a significant difference compared with the situation of the smooth surface, with just a very slight decrease in the maximum contact pressure. However, the groove on the surface can significantly interrupt the contact pressure concentration in the leading edge for the 45º groove-textured surface. The existence of the groove can further interrupt the contact pressure concentration at the leading edge for the case of 90º groove-textured surface, and the maximum contact pressure is further reduced. It shows a significantly higher uniformity of pressure distribution compared with the situation of the smooth surface. For the case of 135º groove-textured surface, the groove can also interrupt the contact pressure concentration at the leading edge. However, the position of the interruption is in the opposite direction of the leading point, which is different from the case of the 45º groove-textured surface. This will lead to the relatively higher maximum contact pressure and also the less uniform pressure distributions of the 135º groove-textured surface compared with the 45º groove-textured surface. Apparently, the more uniform the contact pressure distribution, the less likely for the pad specimen to generate squeal. Therefore, the contact pressure analysis of these five frictional systems can well explain the results obtained from both the experimental and numerical analyses.

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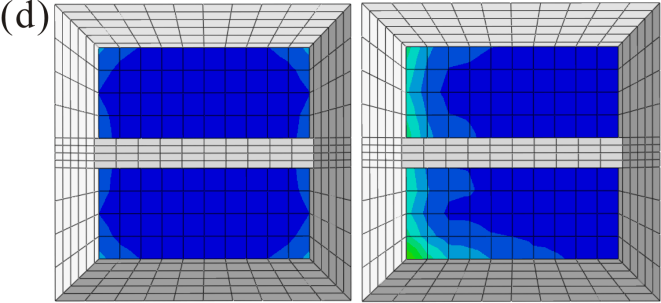
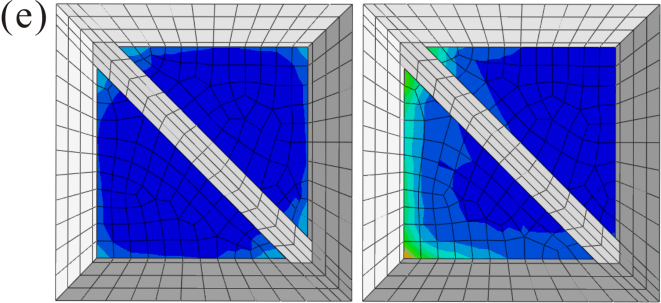
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Fig. 15. Contact pressure distribution for the smooth surface (a), 0° (b), 45° (c), 90° (d) and 135° (e) groove-textured surfaces at speed of 0 rad/s (left) and 6.28 rad/s (right).

1. **Conclusions**

This paper presents a combined experimental and numerical study of squeal noise using a pad-on-disc experimental setup. In the experimental study, four kinds of modifications of surfaces (in the form of cut grooves) of the friction material from a brake pad are made, and the effect of groove orientations on squeal instability is investigated. The numerical study is consequently conducted to simulate the experimental process, using complex eigenvalue analysis and dynamic transient analysis. From both the experimental and numerical results, conclusions could be made as follows:

(1) Experimental results show that cutting a 0º groove on the pad surface has no effect in reducing the unstable vibration of the friction system while cutting a 135º groove can reduce unstable vibration to some degree; in contrast, cutting a 45º or 90º groove on the pad surfaces shows a great potential in squeal reduction.

(2) The numerical model created in this work can be efficiently used to reveal the effect of 0º groove, 45º groove, 90º groove and 135º groove on pad surfaces respectively on the squeal noise and the numerical results agree fairly well with the experimental results.

(3) The maximum contact pressure is the lowest for the 90º groove-textured surface, followed by the 45º groove-textured surface. This may be the reason that results in the visible reduction of vibration acceleration amplitude of the 45º groove surface and 90º groove surface. And the maximum contact pressure for the 135º groove-textured is lower than that of the smooth surface and 0º groove-textured surface.

(4) The findings made in this paper are particularly relevant to brake squeal and are also applicable in other applications involving dry-friction-induced high-frequency noise.

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