Debris trapping and space-varying contact via surface texturing for enhanced noise performance

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**Abstract:**

In this work, groove-textured surfaces with a certain widths and pitches are manufactured on flat cast iron samples. Whereafter, another material (aluminium and steel, respectively) is filled into the grooves to form a special surface which possesses space-varying properties. These surface-treated specimens are tested in a ball-on-flat system and the ensuring friction-induced vibration and noise are measured. The ball sample used in the test is chromium bearing steel ball (AISI 52100). It is found that adding aluminium material into the grooves will increase the friction-induced vibration and noise level of friction system. Conversely, adding steel material into the grooves can stabilize the friction system. Moreover, this space-varying contact state surface is valuable to study the relationship between the trapping wear debris behaviour and stability of texturing surface, which validates our previous speculation that wear debris accumulation on the contact surface will deteriorate the friction-induced vibration and noise behaviour of friction system, and the capability of grooves in trapping wear debris is beneficial for reduction of vibration and noise. Numerical simulation is performed to give reasonable explanations for the experimental phenomenon. This investigation suggests that friction surfaces with grooves filled with suitable materials are able to reduce friction-induced vibration and noise.

Key words: Space-varying surface property; groove-textured surface; friction-induced vibration and noise; friction; wear; wear debris.

1. **Introduction**

Friction-induced vibration and noise (FIVN) arising from mechanical applications which contain sliding frictional contact is commonplace [1-4]. The FIVN phenomenon in most cases is undesirable or disturbing, such as the squeal noise of a vehicle brake system or the squeaking of door hinges [1]. Generally speaking, vibration and noise caused by friction can be roughly divided into two categories according to its frequency range, i.e. the low frequency chatter (*f*<1000 Hz), and high frequency squeal (*f*>1000 Hz) [3]. Compared with chatter, squeal noise is regarded as one of the major concern problems, because it may address serious damage and pollution for modern industry and living.

FIVN is of major interest in many disciplines such as tribology, vibrational dynamics, fracture mechanics and earth science, Ibrahim et al. [1] and Akay [3] et al. all provided a wide panorama of FIVN problems. Over the past decades, the related issues of FIVN have been extensively studied experimentally as well as numerically. Nowadays, a consensus has been reached that mainly four possible mechanisms are responsible for causing FIVN: stick-slip, negative friction-velocity slope, sprag-slip and mode coupling of structures [5]. However, because FIVN is very sensitive to many operating factors [6], none of the above theories can completely explain all of the FIVN phenomenon.

Among these operating factors, surface topography is acknowledged to be able to significantly affect the generation and evolution of FIVN. In the literature, the relationship between surface topography and friction instability has been extensively studied experimentally [7-19] and numerically [20-29]. For the experimental study, research methods can be simply divided into two categories. In the first method, researchers study the evolution of squeal noise and link friction and wear with vibration and noise performance [7-13]. The second method involves studying the role of different surface topographies in affecting tribological and FIVN behaviour of hardware with special contact surfaces [14-19]. In numerical simulations, researchers introduced the realistic interfacial topography parameters to finite element models of friction systems, and determined the contact pressure distribution on surfaces of different interfacial topographies and found the particular surface topography which could suppress squeal instability [20-25]. In addition, statistical theory is also used to investigate how uncertainty associated with surface topography affects noise generation, and extensive laboratory measurements were carried out to evaluate the uncertainty in some modal and contact parameters, which has become a significant methodology for the numerical study of the FIVN [26-29].

The above mentioned studies have shown the significant role of surface properties in affecting FIVN. However, uncertain interface conditions, such as the random distribution of asperities and wear debris always cause difficulties in predicting FIVN performance, and the complicated micro-morphology makes it difficult to obtain repeatable tests results. Thus, the FIVN is characterized by the typical ‘capricious’ behaviour [30, 31]. As a consequence, looking for a special surface which possesses relatively regular surface properties on macroscopic level to study the FIVN related to surface topography is absolutely essential.

In the authors’ recent work, the FIVN behaviour related to the regular groove-textured surfaces has been studied experimentally and numerically [32-34]. It is demonstrated that a certain groove dimension and configuration on the contact surface is beneficial for reducing FIVN. Two reasons are proposed to explain why the grooves can reduce FIVN: firstly, the grooves on the contact surface creates a discontinuity contact condition, which is able to cause a favourable distribution of contact pressure at the contact surface and thus suppress the FIVN of the friction system; Secondly, the FIVN reduction is partly attributed to the capability of grooves in trapping wear debris and avoiding the accumulation of wear debris at the contact interface [35]. On the first reason, a systematic study has been performed to shed light on the role of grooves in modifying contact states and FIVN [33-34]. However, on the second reason, there is not a strong theoretical study that proves that the trapping of wear debris by grooves can suppress FIVN; moreover, the connection between the FIVN and surface wear is not fully clear, particularly that relationship between the wear debris accumulation on the contact interface and deterioration of FIVN behaviour. Therefore, it is essential to perform further research to investigate the relationship between wear debris trapping of grooves and FIVN, and consequently evaluate the role of grooves in suppressing the FIVN of friction system through the wear debris trapping.

Apparently, those groove-textured surfaces studied in [32-34] can not only trap wear debris, but also create time-varying contact states during the relative sliding between the two contacting surfaces, thus it is difficult to separate the double roles played by grooves in reducing FIVN-trapping wear debris and presenting space-varying contact. With this in mind, a special contact interface is designed and created, which possesses the property of time-varying contact state during relative sliding but disallows trapping of wear debris. If strong FIVN can still occur from this kind of time-varying contact states, the effect of wear debris on FIVN is clear.

In this work, groove-textured surfaces with a certain widths and pitches are manufactured on cast iron samples. To form special surfaces which possess the time-varying contact state property during sliding without trapping wear debris, another material (aluminium and steel, respectively) is filled into the grooves. These specimens are slid underneath a steel ball under pressure in tests and FIVN ensue. Moreover, a numerical study is accordingly carried out to give a possible explanation for the phenomenon observed in the test process. The final goal of this work is to verify the role of grooves’ trapping wear debris behaviour in reducing FIVN, and study the effect of space-varying contact state surface on stability of friction system, which can further understand the generation mechanism of FIVN and seek a potential surface modification way to reduce squeal instability.

1. **Experimental details**

2.1 Sample preparation

A special surface which possesses the time-varying contact state property during sliding is deliberately designed. Flat samples which are made of cast iron are cut to the size of 10 mm×10 mm×20 mm, and polished to a surface roughness of approximately 0.04 μm *R*a. Several grooves with width of 2 mm, pitch 4 mm and depth of 1 mm are manufactured on the sample surface through machining. Subsequently, some aluminium slices and forged steel slices whose sizes that are slightly thicker than grooves are manufactured, and then pressured into the grooves to form an interference fit. Such treated surfaces are polished to flat by grinding. This material processing helps obtain a special flat surface which possesses the property of time-varying contact states when another structure slides over it. The flat samples used in this study are shown in Fig. 1(a). In addition, optical images of the flat samples are presented to illustrate their morphology in their initial state, as shown in Fig. 1(b). The ball sample used in the FIVN test is a chromium bearing steel ball (AISI 52100, HV0.05 510 kg/mm2, *E*=210 GPa, ~0.02 mm *R*a) with a diameter of 10 mm. To distinguish the different flat samples, some abbreviations are defined as the following: cast iron indicates the original cast iron flat sample, cast iron+Al indicates that the grooves are filled with aluminium material, and cast iron+steel indicates that the grooves are filled with steel material.

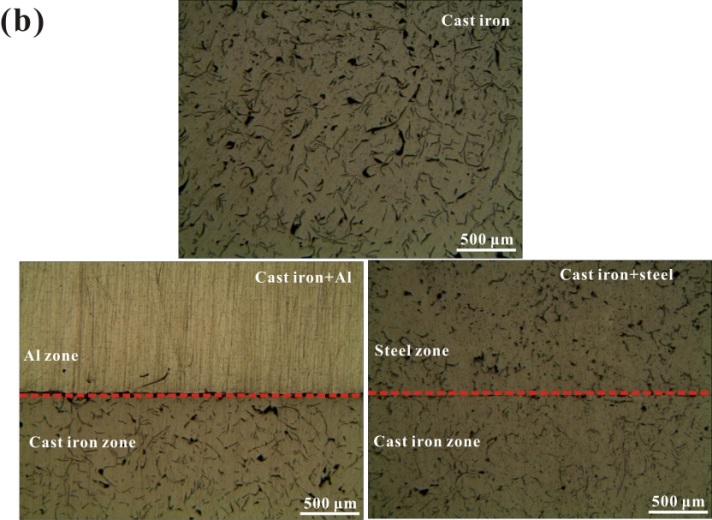
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Fig. 1. Schematic diagram and side view (a) and the surfaces optical images (b) of the flat samples.

It is worth noting that the Young’s moduli of aluminium and steel are smaller and greater than that of the cast iron, respectively, thus adding these two different kinds of materials into the grooves can create two kinds of very different contact interfaces. In the following analysis, these two surfaces will be shown to exhibit very different dynamic behaviour and they are shown to reveal the role of wear debris in affecting the FIVN behaviour.

2.2 Details of test rig

Ensuing FIVN tests are performed on a ball-on-flat tribometer. The ball-on-flat tribometer is mainly composed of a frictional contact system, a holder system and a signal processing system. The schematic of the tribometer is shown in Fig. 2. A flat specimen is fixed to the sliding table which is fixed on the driving device. A ball sample is carried by the ball fixture, which is held by a suspension and pressed down to sliding against the lower flat sample. A 3-D acceleration sensor is mounted on the ball fixture to measure the real-time vibration signals. A microphone is used to measure the noise signal emitted from the contact surface. All the vibration and noise signals are synchronously measured and analysed during the testing period. More details about the test rig can be seen in [32].

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Fig. 2. Schematic of the FIVN test rig: (1) Flat sample, (2) Sliding table, (3) reciprocating sliding device, (4) Ball sample, (5) Ball fixture, (6) 3-D acceleration sensor, (7) Suspension, (8) Strain-gauge force sensor, (9) Microphone, (10) Acquisition and analysis system, (11) Computer.

Considering the aim of this work is to detect the role of wear debris in affecting FIVN, and the contact discontinuities in modifying the squeal instability behaviours, the experimental tests are performed in the condition that the ball-on-cast iron system has a strong tendency in generating squeal instability. Accordingly, many preliminary tests have been performed in the ball-on-cast iron system to find out which set of test parameters leads to sustainable and stable squeal instability phenomenon, by varying different test parameters. Finally, when setting a normal force of 20 N, and a reciprocating sliding frequency of 0.5 Hz at a stroke of 18 mm, the friction system is found to be able to generate strong and sustainable FIVN. Therefore, these values are chosen as the FIVN testing parameters. The FIVN tests are performed for three different testing periods, i.e. 300 s, 600 s and 900 s, respectively, and the corresponding tribological and dynamic signals are measured to conduct comparative analysis. All the tests in this study are performed in strict atmospheric conditions (60%±10% RH, 25 ℃). To ensure that the test results have good repeatability, each test is repeated at least four times.

1. **Experimental results and discussion**

3.1 FIVN result analysis

To better show the noise and vibration evolution for three flat samples in different sliding stages, the root-mean-square (RMS) values of the noise pressure and vibration accelerations are calculated and shown in Fig. 3. In the testing period from 296 s to 300 s, squeal noise and visible higher amplitude vibration are detected from the cast iron+Al surface, and the noise is classified as squeal by analysing the noise frequency as shown in the following section. This phenomenon indicates that the friction system with the cast iron+Al surface has already become unstable and generated squeal in this duration. In contrast, nearly no visible high amplitude vibration signals and no strong squeal noise generation can be observed from the cast iron surface or cast iron+steel surface (Fig. 3(a)). When the tests proceed to 600 s, the noise and vibration signals measured from the cast iron+Al surface still stays at a higher level, whilst squeal noise is found to emit from the cast iron surface, where the RMS values of both the normal and tangential vibration acceleration signals of the cast iron surface show significant increase, compared with the duration of 296 s-300 s. While for the cast iron+steel surface, the noise and vibration magnitude still stays at a relatively low level (Fig. 3(b)). When the tests come to the final stage, i.e. the sliding process lasts for 900 s, for the cast iron+Al surface, the noise and vibration acceleration magnitude still maintains the highest level among the three surfaces during this period, whereas the vibration amplitudes from the cast iron+steel surface are visibly lower than those of the cast iron and cast iron+Al surfaces. Therefore, cast iron+Al surface can increase the tendency of friction system in generating squeal instability; in contrast, the cast iron+steel surface has a good potential in delaying the generation of FIVN from the contact surface. This result indicates that although the cast iron+Al surface possesses the time-varying contact property during sliding, strong squeal instability can emit from this surface.

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Fig. 3. The noise and vibration signals for the three flat samples in the period of 296 s to 300 s (a), 596 s to 600 s (a) and 896 s to 900 s (c).

The corresponding [spectral](file:///G:\Youdao\Dict\7.0.1.0214\resultui\dict\result.html?keyword=spectral) [analysis](file:///G:\Youdao\Dict\7.0.1.0214\resultui\dict\result.html?keyword=analysis) of noise signals are carried for the three contact configurations, as shown in Fig. 4. The main frequencies of noise emitted from the ball-on-cast iron system are characterised by two fundamental frequencies of 1074 Hz and 2363 Hz. Compared with the ball-on-cast iron system, the ball-on-cast iron+Al system shows similar frequency distribution with higher energy. However, for the ball-on-cast iron+steel system, similar frequencies with much lower energy distribution is observed, which indicates that adding steel material into the grooves can reduce FIVN effectively. Therefore, adding aluminium/steel material into the grooves has no strong effect on the distribution of fundamental noise frequency, but it can significantly affect the intensity of FIVN.

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Fig. 4. The FFT analysis of the sound pressure signals for the cast iron (a), cast iron+Al (b), and cast iron+steel (c) surfaces.

3.2 Friction and wear analysis

The variation of friction coefficient curves of the three flat surfaces are shown in Fig. 5. The cast iron+Al surface exhibits higher value of friction coefficient (close to 0.5) compared with the cast iron and cast iron+steel surfaces, and the cast iron+steel shows the lowest value of friction coefficient (approximately 0.25) among the three surfaces. Therefore, adding aluminium material into the grooves can increase the friction coefficient; in contrast, adding steel material into the grooves causes the reduction of friction coefficient.

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Fig. 5. Friction coefficient curves fort the three surfaces throughout the test.

To further illustrate the different dynamical behaviour of the flat samples, worn surface morphologies of these three contact configurations are examined. The aim of performing the worn surface morphology analysis is to link the wear behaviour with the noise and vibration performance, and establish a relationship between them. Fig. 6 shows the worn surface morphologies of the ball and three flat surfaces in three different sliding stages. At the time of 300 s, for the ball-on-cast iron+Al configuration, the surfaces of the contact pairs exhibit relatively complicated wear morphologies than other two configurations, and a noticeable amount of surface damage and wear debris accumulation can be observed in the wear track; meanwhile, the counterface ball shows a large area and an irregularly shaped wear scar, and a large amount of wear debris accumulation and visible ploughing behaviour. In contrast, for both the ball-on-case iron and ball-on-case iron+steel configurations, the surfaces of the contact pairs undergo mild wear in this duration, no severe wear damage and wear debris accumulation can be observed in the wear track and wear scar. Linking this result with the vibration and noise results shown in Fig. 3(a), it is found that for the contact pairs which suffer from more serious wear and more wear debris accumulation, higher intensity vibration and noise can appear from the interfaces.

At the time of 600 s, for the ball-on-cast iron+Al configuration, the contact surfaces maintain complicated and serious wear behaviour, and more wear debris are accumulatedin the wear area. Meanwhile, it is founded that some wear debris have accumulated at the contact interfaces of the ball-on-cast iron configuration, and the degree of wear becomes more severe on the wear track. In contrast, the contact surfaces of the ball-on-cast iron+steel configuration remain relatively clean, no visible wear debris piles up in the wear area. Accordingly, linking this result with the vibration and noise results shown in Fig. 3(b) further demonstrates that the contact surfaces which undergo severe wear and lead to large wear debris accumulation can promote the occurrence of unstable vibration and the emission of squeal.

In the 900 s, more and more wear debris are accumulated at the contact interfaces for both the ball-on-cast iron and ball-on-cast iron+Al configurations, thus high intensity squeal instability is detected from these two configurations. [In contrast](javascript:;), the wear condition of the ball-on-cast iron+steel configuration stays relatively mild, and only a small amount of wear debris is found around the wear scar and wear track. But, it is worth noting that ball-on-cast iron+steel friction system starts to shows a detectable but lower level of squeal instability in this condition, as shown in Fig. 3(c).

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Fig. 6. Optical images of the ball and three flat surfaces at different time instants

To further exhibit the wear states of these surfaces, the 3D surface topographies of both the cast iron+Al and cast iron+steel surfaces at different time instants are obtained by a white light interferometer, as shown in Fig. 7. For both surfaces, more serious wear can be observed with the increase of sliding time. It is visible that the cast iron+Al surface suffers severer wear than cast iron+steel surface, as lager and deeper wear track than that of the cast iron+steel surface in this three time instants. In addition, for the cast iron+Al surface, the wear track in the Al zone is apparently larger and deeper compared with the cast iron zone. This is because Al is softer than cast iron, and thus the Al zone will suffer severe wear damage during sliding process. This phenomenon is consistent with the results measured from the optical microscope, as shown in Fig. 6.

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Fig. 7. 3D surface topographies of both cast iron+Al and cast iron+steel surfaces at different time instants.

Furthermore, the wear morphologies of the cast iron+Al and cast iron+steel surfaces are observed by using SEM, as shown in Fig. 8. The cast iron+Al surface exhibits very complicated and serious wear behaviour: a large amount of wear debris is accumulatedin the wear area, visible material micro-exfoliation and detachments can be observed on the third body layer. In addition, small plateau with micro-exfoliation occurs on the surface [36]. Since the main component of the ball sample is iron, its chemical affinity to aluminum will determine the adhesive wear behavior between the contact interfaces [37]. It has been reported that the aluminum material has affinity for the steel (iron) surface. This affinity results in chemisorption and adhesion of aluminum and formation of intermetallics on the steel surface [38]. Therefore, in this work, severe wear that occurs in the ball-on-cast iron+Al system should be linked with the high affinity between steel and Al. This affinity plays an important role in generating adhesive wear between the contact surfaces. While for the cast iron+steel surface, the degree of wear is very mild, and only a small amount of wear debris appears in the wear area. A smooth and compact layer of third body is formed on the contact surface, but no visible material micro-exfoliations and detachments can be detected on the third body layer. This phenomenon supports the results measured from optical [microscope](file:///G:\Youdao\Dict\7.0.1.0214\resultui\dict\result.html?keyword=microscope) that the wear of cast iron+Al surface is severer and more complicated compared with the cast iron+steel surface.

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Fig. 8. SEM images of both cast iron+Al (a) and cast iron+steel (b) surfaces at the time of 600 s.

3.3 Further verification of the role of wear debris

The above results have shown that the ball-on-cast iron+Al configuration with severer wear states possesses a stronger tendency to generate squeal instability in comparison with the ball-on-cast iron and ball-on-cast iron+steel configurations. Thus, one question should be further answered is that whether the higher intensity of squeal instability of the ball-on-cast iron+Al configuration is originated from the severer wear states, or whether the different material properties among the cast iron, [aluminum](file:///G:\Youdao\Dict\7.0.1.0214\resultui\dict\result.html?keyword=aluminum) and forged steel are responsible for promotion of squeal. Only understanding this point can a reliable conclusion on the relationship between the wear debris and FIVN be reached.

Fig. 9 shows the time history records of tangential vibration acceleration signals for all the three contact configurations at two different sliding stages. For both the cast iron and cast iron+steel surfaces, their vibration signals are fairly stable, while for the cast iron+Al surface, the vibration signals exhibit some intermittent and high frequency oscillations. Each oscillation cycle indicates that the system experiences a growth and then a decrease and even disappearance of vibration. According to the sliding velocity and sliding distance, it can be deduced that when [aluminum](file:///G:\Youdao\Dict\7.0.1.0214\resultui\dict\result.html?keyword=aluminum) zone is sliding against the ball sample, the vibration amplitude starts to decrease and gradually disappear. This phenomenon suggests that the presence of [aluminum](file:///G:\Youdao\Dict\7.0.1.0214\resultui\dict\result.html?keyword=aluminum) material in the grooves can reduce the vibration intensity during the period when the ball slides on the [aluminum](file:///G:\Youdao\Dict\7.0.1.0214\resultui\dict\result.html?keyword=aluminum) area. However, the large accumulation of wear debris and severer wear states occurring at the contact interface can increase the level of vibration.

The above analysis, indicates that the wear behaviour has a strong relationship with the stability of friction system. The larger amount of wear debris accumulation on the wear area can promote the generation of FIVN, and consequently increase the intensity of FIVN. [In other words](javascript:;), if the wear debris accumulated in the interface can be removed through a reasonable way, FIVN will be reduced. Therefore, the experimental phenomenon shown in this work can effectively verify our previous views: in the case of groove-textured surfaces, the grooves’ trapping wear debris behaviour is beneficial for removing the accumulation of wear debris in the interface, which plays an effective role in reducing the FIVN of the friction system.

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Fig. 9. Tangential vibration acceleration signals for the three contact configurations in the period of 597 s to 598 s (a) and 898 s to 899 s (b).

1. **Numerical simulation**

4.1 Creation of the finite element model

In this section, numerical analysis is carried out by using ABAQUS to simulate the test process and give a reasonable explanation for the experimental phenomenon. A finite element model of the ball-on-flat friction system is presented in Fig. 10 (a). This model covers seven main components of the real test system, and the material parameters defined for all the components can well reflect the real experimental system, as listed in Table 1. The flat sample is in frictional contact with the ball sample during the sliding process. Considering that the ball sample is made from a harder material, thus the ball surface is set as the master surface and has a coarse mesh than that of the flat sample. The flat sample surface is accordingly set as the slave surface. Fig. 10(b) illustrated the boundary condition set in this numerical model: a constant force (20 N) is applied on the surface of the force sensor, and the velocity boundary condition is applied on the sliding table along the X-direction.

Table 1. The material parameters of the finite element model

|  |  |  |  |
| --- | --- | --- | --- |
| Parts | [Density](app:ds:range) (kg/m3) | Young’s modulus (GPa) | Poisson’s ratio |
| Force sensor | 2700 | 69 | 0.27 |
| Suspension (plate) | 7800 | 210 | 0.305 |
| Suspension (shim) | 2200 | 0.5 | 0.27 |
| Ball holder | 7800 | 210 | 0.305 |
| Ball sample | 7800 | 200 | 0.305 |
| Flat (cast iron) | 7300 | 158 | 0.3 |
| Flat (Al) | 2700 | 69 | 0.27 |
| Flat (Steel) | 7800 | 210 | 0.305 |
| Table | 7800 | 210 | 0.305 |

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Fig. 10. Finite element model (a) and the corresponding load and boundary condition (b).

4.2 Complex eigenvalue analysis (CEA)

CEA is adopted in this section to evaluate the squeal propensity of the friction systems under different cases of flat samples. The eigenvalues results can be used to evaluate the squeal tendency of the friction system. The system will show a strong potential in generating squeal instability when the eigenvalues occur positive real parts, the positive real parts and the corresponding imaginary parts indicate the level of unstable vibration and the excited squeal frequencies, respectively. In addition, the squeal instability trend of the friction system can be evaluated through the value of the effective negative damping ratio (*ξ*), the smaller of the value of *ξ*, the easier for the unstable vibration to occur. It worth noting that that the energy indexes (MAI index, feed-in index) are appropriated to reflect the squeal propensity of friction system, which are able to identify the mode with the highest propensity to squeal in the steady state unstable response [39]. However, in many published papers, the negative damping effective ratio can also be commonly used as an approximate index to reflect the FIVN trend of friction system. Therefore, in this work, the squeal instability trend of the friction system is evaluated through the value of the effective negative damping ratio. More detailed introduction about this analysis method can be seen in references [40].

In the CEA analysis, the real surface condition is not introduced into the finite element model, considering the uncertain size and shape of wear debris and the higher cost of the computation time. However, it is found from the experimental results that the values of friction coefficient vary with the accumulation of wear debris and wear statues, thus the average friction coefficient values of the three friction systems measured during different sliding durations (296 s-300 s, 596 s-600 s and 896 s-900 s) are used in the CEA of the same friction systems for the corresponding time durations, respectively. Table 2 lists the average friction coefficient values of the three friction systems in these different periods. In fact, many studies have shown that the friction coefficient can strongly affect the model coupling characterise and stability of a friction system, thus the aim of this CEA analysis is to verify the accuracy of the finite element model, and further verify the effect of space-varying contact state surface on system stability. In addition, it is worth noting that for both the ball-on-cast iron+Al and ball-on-cast iron+steel configurations, their eigenvalues are calculated at two different contact conditions (i.e., the ball is contacting with the cast iron zone and ball is contacting with the Al or Steel zone), because of their space-varying surface property. For the ball-on-cast iron+Al configuration, whether the ball is in contact with the Al or cast iron, the boundary conditions of the finite element model are consistent. In addition, the friction coefficient value used in the CEA analysis is the average friction coefficient measured within a certain time period, in which ball slides on the Al section and the cast iron section. Thus, the main difference of parameter values in the simulations between the two contact situations (ball-on-Al and ball-on cast iron) is in the stiffness of the surface. The softer material (Al) possesses lower surface contact stiffness than the cast iron material. Similarly, for the ball-on-cast iron+steel configuration, the harder material (steel) possesses larger surface contact stiffness than the cast iron material.

Table 2. The average friction coefficient values of the three friction systems in different times

|  |  |  |  |
| --- | --- | --- | --- |
|  | 296 s-300 s | 596 s-600 s | 896 s-900 s |
| *μ* of Ball-on-cast iron | 0.254 | 0.32 | 0.36 |
| *μ* ofBall-on-cast iron+Al | 0.455 | 0.471 | 0.50 |
| *μ* of Ball-on-cast iron+Steel | 0.15 | 0.2 | 0.24 |

The complex eigenvalue analysis result of the friction systems in the experimental duration of 296 s-300 s is shown in Fig. 11(a), no negative effective damping ratio can be observed for both the ball-on-cast iron and ball-on-cast iron+steel configurations, which indicates that these two friction systems are stable during this period. While for the ball-on-cast iron+Al configuration which possesses a larger value of friction coefficient at this time instant, two visible negative effective damping ratios appear (*ξ*1=-0.05863, *ξ*2=-0.02312), suggesting that the ball-on-cast iron+Al friction system has a strong potential in generating unstable vibration and accordingly squeal noise. Thus, this result is consistent with the vibration and noise results shown in Fig. 3(a). Moreover, it is found that the calculated unstable vibration frequency are 995 Hz and 2350 Hz, respectively, which are very close to the experimental vibration frequencies shown in Fig. 4.

Fig. 11(b) presents the complex eigenvalue analysis result of the friction system in the experimental period of 596 s-600 s. Both the ball-on-cast iron and ball-on-cast iron+Al friction systems show negative effective damping ratio, which suggests that both of these two friction systems will generate unstable vibration and squeal noise in this moment. It is worth noting that the effective damping ratio of the ball-on-cast iron+Al friction system is smaller than that of the ball-on-cast iron friction system. This indicates that the ball-on-cast iron+Al friction system still has a stronger potential to generate unstable vibration compared to the ball-on-cast iron friction system. While for the ball-on-cast iron+steel friction system, there is still no negative effective damping ratio shows in this moment, thus the ball-on-cast iron+steel friction system still stays in a stable condition. This result is still consistent with the vibration and noise results shown in Fig. 3(b).

Fig. 11(c) provides the complex eigenvalue analysis result of the friction system in the experimental duration of 896 s-900 s. With the increase of friction coefficient, both ball-on-cast iron and ball-on-cast iron+Al friction systems show smaller negative effective damping compared to case of the 596 s-600 s, in addition, the negative effective damping ratio of ball-on-cast iron+Al friction system is still lower than that of ball-on-cast iron friction system, suggesting that ball-on-cast iron+Al friction system still has stronger tendency to generate unstable vibration. In contrast, no negative effective damping ratio obtained from the ball-on-cast iron+steel friction system in this moment, thus the ball-on-cast iron+steel friction system still stays in a stable condition.

Linking the numerical results with the experimental results shown in Fig. 3 and 4, it can be seen that the finite element model created in this study can well reflect the stability of the friction system, and the cast iron+Al surface has a stronger tendency to generate squeal instability compared with the cast iron+steel surface. In addition, the calculated results show that for the cast iron+Al surface, the unstable vibration occurs in the area where the ball contacts with the cast iron, which is consistent with the experimental results shown in Fig. 9.

Two dominant unstable mode shapes of the ball-on-flat system are shown in Fig. 11(d). It is seen that unstable model shape occurs in the suspension and the long bar-shaped ball holder, which consists of the bending deformation of suspension and ball holder carrying ball sample sliding on the flat surface along the X direction. In addition, it is found that the unstable mode shape of the systems for the cases of the space-varying surface are almost the same as that of the plain cast iron surface, thus the space-varying surface will not affect the mode shapes of these ball-on-flat system.

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Fig. 11. CEA results of the friction systems at the testing time of 296 s-300 s (a), 596 s-600 s (b), 896 s-900 s (c) and the unstable mode shapes (d).

4.3 Transient dynamic analysis (TDA)

In this section, the TDA is conducted to simulate the vibration evolution of the friction systems in time domain. The TDA accounts for the contact nonlinearity and allows the system vibration characteristics in different contact situations to be studied. In the TDA process, the velocity and the displacement can be obtained by calculating the kinetic equations of the friction system, according to the explicit central difference integration rule. On the detail introduction of TDA performed in ABAQUS can be seen in [41].

In the TDA, the system parameters, such as the time variation of the normal load and the sliding speed of the flat sample are the used to simulate the real operating condition. Firstly, a gradually increased normal load is acted on the top surface of the finite element model until it reaches 20 N. This load remains constant thereafter and brings the ball sample and flat sample into contact. Then, a sinusoidally varying velocity is applied to the flat subsystem to make it achieve reciprocating sliding behaviour. The observation point (as labelled in Fig. 10) of the vibration acceleration is on the ball holder surface, which is the location that the acceleration signal is placed in the tests. During the TDA, all the surfaces are considered as flat planes. Considering that the real friction coefficient values varies with the accumulation of wear debris and wear statues, different friction coefficient values (as shown in Table 2) of the three friction systems are set in the numerical model, to represent different degrees of wear.

Fig. 12 illustrates the simulated vibration acceleration in the friction direction in time domain. Fig. 12(a) shows the vibration behaviour of the three friction systems at the testing period of 296 s-300 s, in which a significant high-frequency burst is found to appear from the ball-on-cast iron+Al system; conversely, nearly no visible high amplitude vibration signals can be observed from the ball-on-cast iron or ball-on-cast iron+steel systems. This phenomenon suggests that the friction system with the cast iron+Al surface has already generated unstable vibration in this period. Fig. 12(b) presents the vibration behaviour of the three friction systems in the testing duration of 596 s-600 s, in which visible high-frequency bursts are seen from both the ball-on-cast iron+Al and ball-on-cast iron systems, while the ball-on-cast iron+steel system produces very slight oscillations. When the sliding process lasts for 900 s, corresponding to the results shown in Fig. 12(c), visible higher magnitude bursts are generated from all the three friction systems, whereas the vibration amplitudes from the ball-on-cast iron+steel system are visibly lower than those two other systems. Thus, the simulated results are consistent with the evolution of FIVN results observed in the experimental tests, as shown in Fig. 3, which further verify that adding aluminium material into the grooves can promote the generation of FIVN, and increase vibration level of friction systems, while adding steel material into the grooves can stabilize the friction system. In addition, Fig. 12(d) shows the simulated contact forces of the ball-on-cast iron+Al system at the final sliding stage (900 s), three intermittent and high frequency oscillations of contact forces are observed in both directions. Linking the vibration behaviour of friction system and the corresponding contact positions between the flat and ball samples, it is found that when an Al zone is sliding against the ball counterface, the vibration amplitude starts to decrease and gradually disappear. While when the cast iron zone is sliding underneath the ball sample, the vibration signals begin to appear visible unstable vibration and the vibration level comes to the highest magnitude again. This simulated result is consistent with the vibration accelerations results observed in the experimental measurement, as shown in Fig. 9.

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Fig. 12. TDA results of the friction systems for the testing periods of 296 s-300 s (a), 596 s-600 s (b), 896 s-900 s (c) and the normal force and friction force with the corresponding contact positions at 900 s (d).

1. **Conclusions**

In this work, we design a special kind of flat surfaces which possess space-varying contact states to validate the role of the surface grooves in trapping wear debris in affecting friction-induced vibration (FIVN) of three friction systems, and also study the effects of these specially treated surfaces on stability of friction system. The experimental and theoretical results are summarized as follows:

(1) Experimental results indicate that adding aluminium material into the grooves will promote the generation of FIVN, and increase unstable vibration level of the friction system. In contrast, adding steel material into the grooves can demote the generation of FIVN from the contact surfaces and reduce the unstable vibration level of the friction system.

(2) The experimental results show that severer wear states and wear debris accumulation at interface can promote the generation of FIVN, and increase the intensity of FIVN. Thus, these results validate our previous speculation that the capability of groove-textured surfaces in trapping wear debris is beneficial for reduction of FIVN.

(3) CEA and TDA results can well reflect the experimental phenomenon to a reasonable degree, and they further confirm that different wear states can affect the tendency of squeal instability of these friction systems.

(4) Friction surfaces with grooves filled with suitable materials are shown to be an effective means of reducing friction-induced vibration and noise.

The further research works are focused on the following two aspects to further verify the proposed method. The first aspect is to study the response of these discontinues contact surfaces under different tribological conditions, and the second aspect is to further verify the results and justify the approach through subscale tests for the purpose of reducing FIVN in practical application of tribology.

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