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Brass fillers in friction composite materials: Tribological and brake squeal characterization for suitable effect evaluation

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Abstract. In this paper, brake pad performance of two organic matrix composites namely, Sample 1 (contains no brass filler) and Sample 2 (contains 1.5% brass filler), is studied based on tribological and squeal noise behavior. In the first stage, a pin-on-disc tribometer is used to evaluate the frictional behavior of the two pads. On the following stage, these pads are tested on squeal noise occurrence using a drag-type brake dynamometer. From the two type of tests, the results show that; (i) brass fillers play a dual role; firstly as reinforcing element of the brake pad providing primary contact sites, and secondly as solid lubricant by contributing to the formation of a layer of granular material providing velocity accommodation between the pad and the disc; (ii) brass fillers contribute to friction force stabilization and smooth sliding behavior; (iii) the presence of small weight quantity of brass filler strongly contributes to squeal occurrences; (iv) there is close correlation between pin-on-disc tribometer and brake dynamometer tests in terms of tribological aspect.

Keywords: friction materials; tribological behavior; brass filler; squeal noise; rubbed surface

1. Introduction

Automobile brake friction materials are usually formed by a phenolic resin binder with the addition of more than 10 ingredients to achieve some acceptable levels of braking objectives, e.g. maximum value and stability of friction coefficient (μ), solid state lubricity, wear resistance, vibration damping, long life and low maintenance costs, at a wide range of braking conditions, temperature, humidity (Kumar and Bijwe 2010, Zhu *et al.* 2010, Mini *et al.* 2012, Zhao *et al.* 2006). It should pose good thermal properties, which interact with the other material properties to improve brake pad performance under braking application. Among the friction material ingredients, metallic fillers such as steel, copper and brass were used in different size, shape and weight percentage, especially for the reason of their excellent thermal conductivity (Kumar and

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Bijwe 2008). Thus, the maximum surface temperature can be controlled and the susceptibility of the system to local overheating in the form of hot banding or hot spotting was reduced (Zhao *et al.* 2006). Macro-particles brass – preferably in the form of chips – can serve as reinforcements, similar to the steel fibers described by Eriksson and Jacobson (2001). Due to their good malleability and plasticity, steel particles were used to improve the dynamical performance of the friction material. The effect of copper fiber on the friction materials performance had been studied by several researchers (Kumar and Bijwe 2010, 2008, Eddoumy *et al.* 2015). However, the contributions of brass filler on brake squeal generation and tribological performance has not been fully studied. In fact, the combined actions of frictional heating and interfacial stresses induced the presence of element Cu on the contact which could affect the stability of friction coefficient and influenced the wear resistance (Kumar and Bijwe 2010). Since the influence of thermal conductivity is best reflected in severe operating conditions, Eddoumy *et al.* 2015 showed that the friction material with copper fibers exhibited a good friction behavior. Xian and Xiaomei (2004) reported that when the mass fraction of brass fibers was over 19%, the friction coefficient and wear loss also increased slightly. They noticed that a percentage of 16-20% of brass fibers induced better wear resistance and a copper transfer film was formed on the friction surface of counterpart. They also concluded that brass fibers were responsible to control the friction level while avoiding the aggressiveness against counter surface. Bijwe *et al.* 2008 studied the influence of brass fillers percentages (0%, 4%, 8% and 12%) on friction material performance in terms of sensitivity of friction and wear to load and speed using reduced scale prototype. They observed that increase in brass contents could lead to deterioration in strength and modulus, hardness, etc. and increase in density, void contents, heat swelling, water absorption, specific heat, thermal diffusivity and conductivity. Furthermore, increase in brass contents could also lead to increase in $\mu_{\text{performance}}$ when evaluated under ECR 90 schedule conducive to precipitate the fade and recovery behavior of the materials. Using a small-scale friction tester, Jang *et al.* 2004 studied the effect of (15 vol.%) metallic fibers (low carbon steel, Al and Cu) on performance of NAO friction composites. It was concluded that the Cu fibers led to high fade resistance followed by steel fibers and then Al fibers. It was argued that high thermal conductivity of copper fibers and the formation of copper oxides at the friction interface during high temperature sliding were responsible for a high fade resistance. High temperature strength of the steel fibers was thought responsible for better performance. Kumar *et al.* 2011 studied three non-asbestos organic (NAO) friction composites with three metallic particulate fillers (iron, copper and brass) using a specific tribometer that allowed them to examine hot spots and thermal localization phenomenon under braking conditions. It was found that metallic contents could improve the overall thermal-localization performance of friction material. The best results were obtained with the iron powder (IP) where there were no hot spots occurred up to the high braking power and energy at 1900 rpm, followed by the friction composites with the brass and the copper. Composite without metal powder (NM) showed the worst performance as the hot spots appeared at the earliest speed of 1500 rpm.

Due to the presence of friction between the pin (brake pad) and the disc interface, it can induce dynamic instability in the system. Experimental investigations revealed that the noise and vibration might not only strongly depend on the contact properties, involving parameters like temperature, humidity, surface topography (Sherif 2004, Massi *et al.* 2008), local pressure and loading history (Mat Lazim *et al.* 2012), but also friction materials composition (Mat Lazim *et al.* 2013). The friction characteristics of abrasive particles were examined using silicon carbide, zircon, quartz and magnesia (Kim *et al.* 2011). It was reported that all friction material specimens produced stick–slip during the slow speed drag tests, which was more significant when silicon carbide and zircon were used as abrasives. Jang *et al*. (2001) attempted to investigate creep groan noise propensity by changing the relative amounts of ingredients in a brake friction material containing 12 ingredients. The difference $\Delta\mu$ of kinetic (μ_k) and static (μ_s) coefficients of friction for each friction material was measured since the stick–slip mechanism was believed to excite the creep groan. It was found that a smaller $\Delta\mu$ value induced low propensity of creep groan phenomena during braking application. Previous researchers (Cho *et al*. 2005, Woldman *et al*. 2012) found that certain friction material with high content such phenolic resin, rockwool, zircon, and MgO, had a high squeal noise propensity where the ingredients with high abrasive action (hardness) increased the coefficient of friction while soft ingredients such as copper and rubber decreased the coefficient of friction.

It is seen that the effect of brass filler on brake squeal occurrence has not been fully studied. In this paper, the small quantity effect (1.5%) of brass filler in friction materials is investigated in terms of friction characteristics, rubbed surface evolution and brake squeal behavior.

2. Experimental procedure

2.1 Friction materials

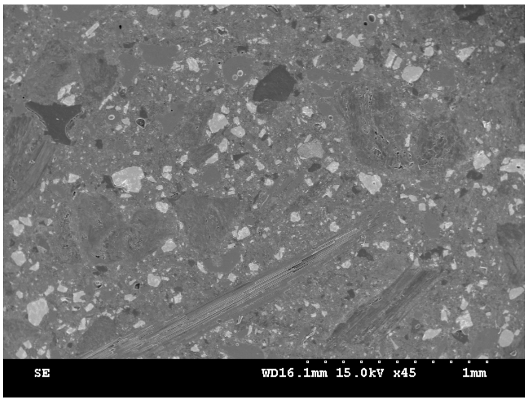
In this work, two pairs of brake pad materials, namely Sample 1 and Sample 2 are tested. These materials contain the most important ingredients, which are commonly used in a conventional friction material, including phenolic resin matrix, reinforcing fiber, friction modifiers, abrasive particles, fillers, solid lubricant and metal oxide. Sample 1 contains no brass filler whilst Sample 2 contains 1.5% brass filler (Table 1). The brass filler is an ingredient that helps to suppress vibration and subsequently reduce the noise (Mat Lazim *et al*. 2014). Fig. 1 shows the microstructure of the two friction material specimens. Physical and mechanical properties of the materials are summarized in Table 2.

Table 1 Organic matrix composites composition

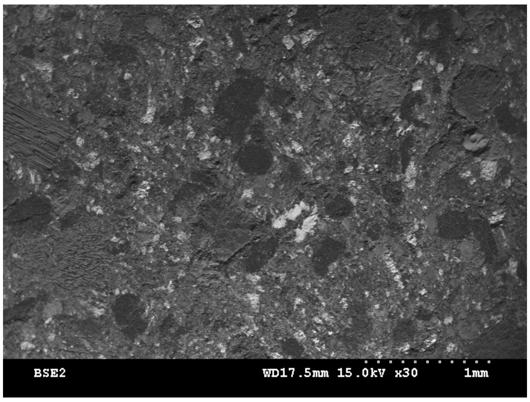
Composition		%Wt	
		Sample 1	Sample 2
Binder (Phenolic resin)		14.2	14.2
Fibres	Brass	0	1.5
	Mineral	22	20.5
Fillers		44.5	44.5
Abrasive		1.7	1.7
Lubricant		10.8	10.8
Friction modifiers		6.8	6.8

Table 2 Organic matrix composites properties

Brake material	Porosity (%)	Thermal conductivity (w $m^{-1}k^{-1}$)	Young’s Modulus [MPa]
Sample 1	15	1.17	19.04
Sample 2	8	1.21	19.51



(a) "Sample 1" without brass fillers



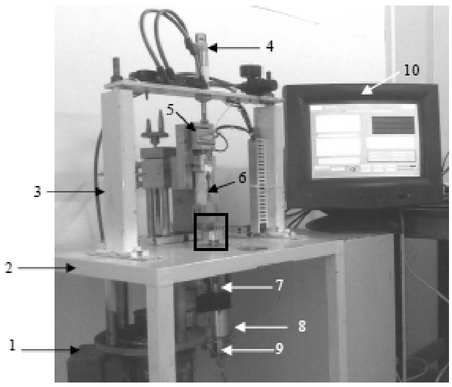
(b) "Sample 2" with brass fillers indicated by arrows

Fig. 1 SEM observations of the two brake lining materials

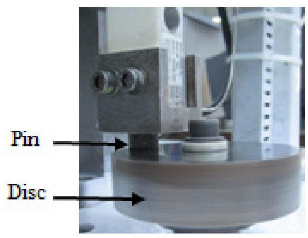
Tribological test and analysis

Tribological tests are performed using a pin-on-disc tribometer as shown in Fig. 2 with a pin made of friction material (diameter of 14 mm and a height of 16 mm) rubbing against a grey cast iron disc (diameter of 50 mm) to compare the friction behavior at elevated temperature. During the test, normal and tangential forces as well as the mean surface temperatures of the disc are measured. Pin and disc temperatures are measured using K-type thermocouples located on the average friction radius respectively, 3 and 2 mm beneath the friction surface.

As illustrated in Fig. 3, the friction test is set in the temperature range of 200-250°C. This test is defined with an increased severity of rubbing conditions, which correspond to a medium energy wear test (ME), allowing reaching a temperature range representative of on road braking severities (200-250°C, reached in service during a severe stop braking and leading to a moderate thermal loading of the friction composite material). Before the tribological test, a preliminary run is performed to achieve smooth plane contact between the pin and the disc surfaces. It consists of



(a) Pin-on-disc tribometer



(b) Detail of the materials contact

Fig. 2 Tribological equipment (1. Motor, 2. Table, 3. Guidance, 4. Pneumatic loading, 5. Normal force sensor, 6. Transducer, 7. Spindle, 8. Four thermocouples type K, 9. Slip ring, 10. Acquisition)

Table 3 Tribological test conditions

Conditions	Run-in	Tribological test
Temperature range (°C)	[70-100]	[200-250]
Sliding speed (m/s)	3	6
Mean pressure (MPa)	0.6	1.2

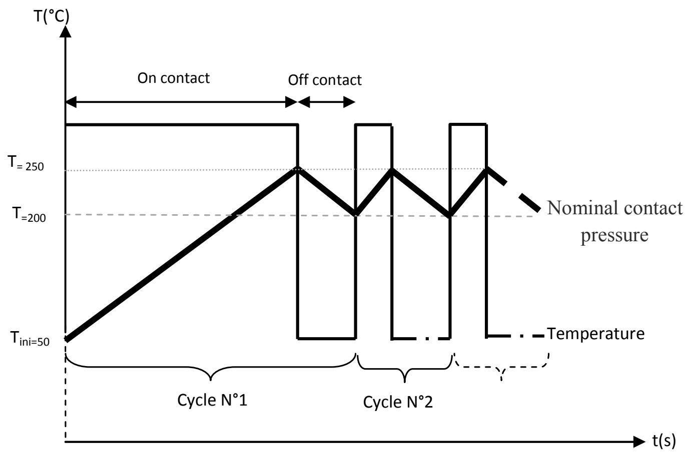


Fig. 3 Experimental protocol of tribological tests

a succession of friction cycles with soft contact conditions (a sliding speed of 3 m/s and a pressure of 0.6 MPa) to avoid degradation by high thermal levels until the pin surface area is entirely rubbed. At the end of the run-in period, the disc temperature is between 70°C and 100°C. After a natural air cooling period, the “on-contact” test period is carried out at a temperature of 200°C with a sliding speed of 6 m/s and a pressure of 1.2 MPa as specified in Table 3. The “off-contact” test period is referred to which the pin is out of the contact and the disc rotation is stopped when the temperature is about 250°C. During this “off-contact” period, the disc temperature decreases due to natural air cooling and once it downs to 200°C, a new on-off contact cycle restarts until it reaches a total of 30 cycles (Hentati *et al*. 2012). A scanning electron microscope (SEM) equipped with an energy-dispersive X-ray spectrometer (X-EDS) is utilized to reveal topography and to identify the different constituents of the third bodies on the rubbed pin surface.

2.2 Brake squeal test

In order to examine brake squeal behavior of the two brake pad samples, a brake noise dynamometer as shown in Fig. 4 is fully utilized. It is powered by an 11 kW DC motor. The test rig can be operated at a maximum torque of 413 Nm and 20 bars of brake hydraulic pressure. In order to capture brake operating parameters and squeal occurrence, a number of transducers such as accelerometers, pressure sensor, microphone, thermocouple, torque sensor and speed sensor are used. These transducers are connected to 16 channels Dewetron data acquisition system (DAQ). The microphone and the uni-axial accelerometers are employed to identify squeal levels (sound pressure level above 70 dBA) and its frequency (above 1 kHz). The squeal frequency can only be

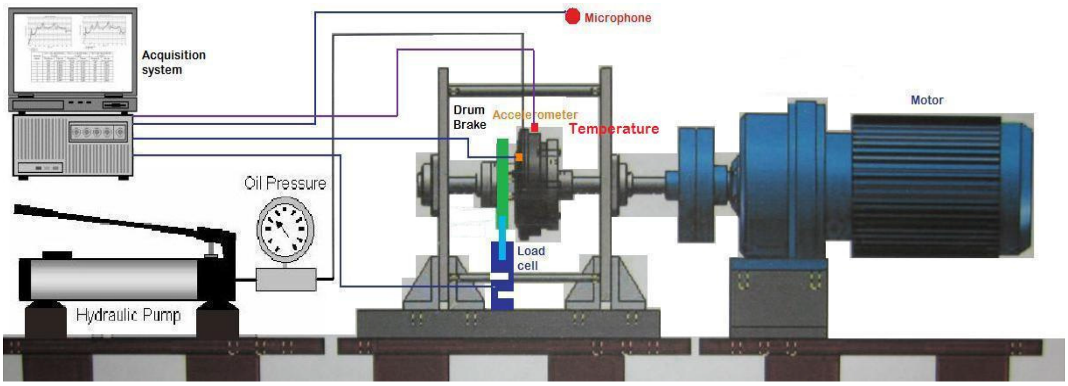


Fig. 4 Brake dynamometer with some attached sensors

confirmed after it is captured both by the microphone and the accelerometer. Squeal events are recorded at various operating conditions such as at disc rotating speeds of 20 to 70 revolutions per minute (rpm) and hydraulic brake line pressure up to 2.0 MPa. The squeal tests are carried out by referring SAE J2521 standard operating procedure. Prior to these tests, warming up (bedding-in) process is conducted to deposit an even layer of friction material, or transfer layer, on the rubbing surface of the rotor disc. This process needs to be done at low disc speed (57 rpm) with a brake pressure between 0.3 to 1.0 MPa for half an hour. Typically, squeal will be easily generated after the bedding-in process. During the squeal tests, seven (7) parameters are monitored and captured, namely disc speed, hydraulic pressure, disc surface temperature, brake torque (friction coefficient), vibration of the pad, humidity and sound pressure levels.

3. Results and discussion

3.1 Frictional behavior

Fig. 5 shows the evolution of the friction coefficient (μ) during the tribological test. The initial value is low, i.e., 0.21, but it increases rapidly up to 0.3 after 50 s. A slight fading effect is detected after 200 s. Some other observations can be described as follows;

- Both samples show the friction coefficient in the range of 0.22-0.5 which is in the acceptable range for cars,
- The friction coefficient is increased from the first to the 30th cycles for the two samples,
- Friction material with brass fillers (Sample 2) exhibits a higher friction coefficient (μ),
- Temperature of the Sample 2 reaches 250°C more rapidly than the Sample 1.

It is known that for an organic matrix composite, the increase in the magnitude of friction coefficient is essentially due to the frictional heat produced on the friction interface as a consequence of pressure increase; leading to the decomposition of the organic ingredients (Zhu *et al*. 2010). The brass filler plays an important role in evolution of friction coefficient (Kumar and Bijwe 2008). In fact, sensitivity of friction coefficient has contributed to the increase of the thermal conductivity of composites. Another reason can be the ability of metallic filler to form a uniform frictional film on mating surface, which keep the friction coefficient at desired level

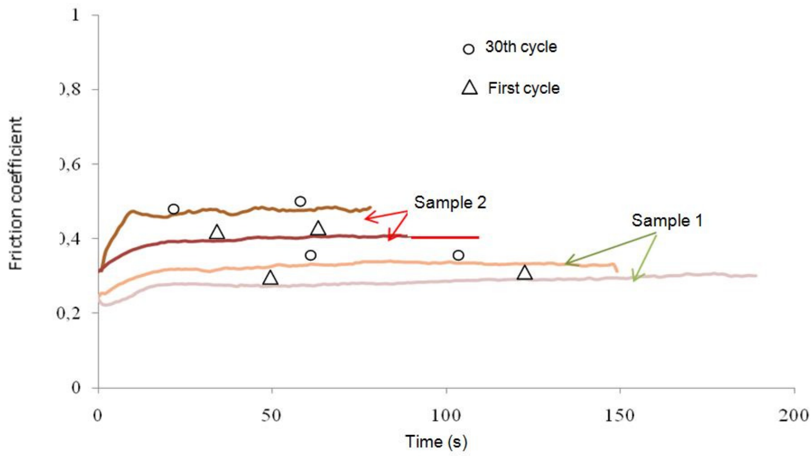


Fig. 5 Friction coefficient evolution for the first and 30th cycles of the two samples

consistently (Kumar and Bijwe 2010, Kumar *et al*. 2011). This behavior will be further discussed in the following section in relation to the worn surface of the rubbed surface.

3.2 Worn surface characterization

SEM observations are carried out to examine characteristics of rubbed surfaces for a possible relationship between the morphological characteristics of the worn surfaces and wear test results. Figs. 6 and 7 exhibit the worn surface of the two samples. Based on Fig. 6, it is seen that the Sample1 has a smooth surface and it is covered by multilayer secondary plateaus which are believed to protect the friction surface from the excessive wear rate at 250°C. The rockwool fibers and shots contribute to the formation of the bearing surface by forming a primary support for the development of secondary plates. The majority of particles are rich in carbon, which appear as dark grey remain totally or partially uncovered. These large particles do not form preferentially supports of the secondary plate development.

As shown in Fig. 7(a), third-body particle at the contact is constituted of numerous powder layers that cover most of the surface between the flat plates. Rubbing surface is marked in some places by the imprints of loose particles. At the beginning of the tribological test, the third body is constituted of powder layers which agglomerate and compact around the harder point, enlarging the effective area of the contact plateaus. This phenomenon is one of the mechanisms involved in the formation and expansion of the flat plates. From Fig. 7(b), it can be observed that the third body particle does not concentrate on the dark particles. Referring to the Backscattered Electron, dark particles are light and rich in carbon such as graphite which is used to stabilize the friction coefficient. As illustrated in Fig. 7(c), the fragmentation of the third body can be attributed to the severity of the solicitation. Eventually, it acts as a protective for the matrix material and that is why it plays a significant role in wear behavior of the brake lining material. The high load may cause the distortion of the flat plates so it can severely affect the wear rate and reduce the lifespan of the friction material. EDX analyses of zone A (see Fig. 8(a)) shows that at points 2, 5 and 6, the large, bright particles are rich of Cu and Zn elements, which can be necessarily from brass filler (Table 4). The brass particles themselves are mostly covered in grey particles. In fact, the presence of Fe elements coming from the disc shows that brass fillers are acting as support to the flat plates

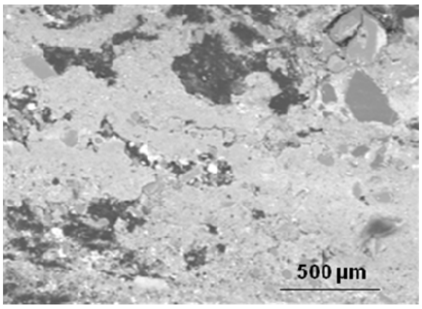
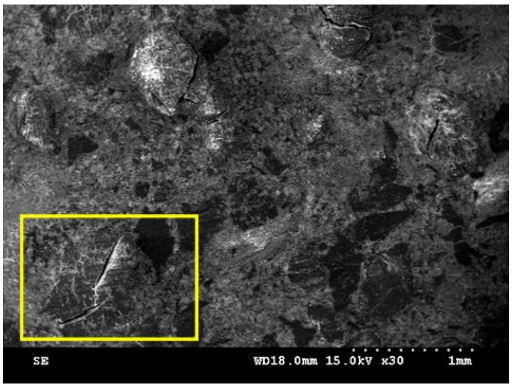
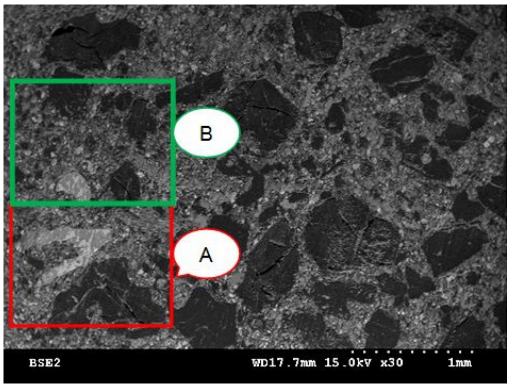


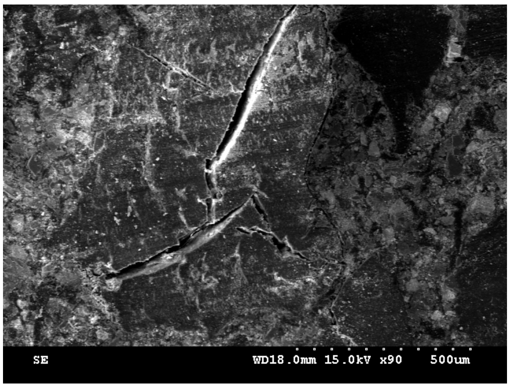
Fig. 6 SEM observation of the rubbed surface of the Sample 1



(a)



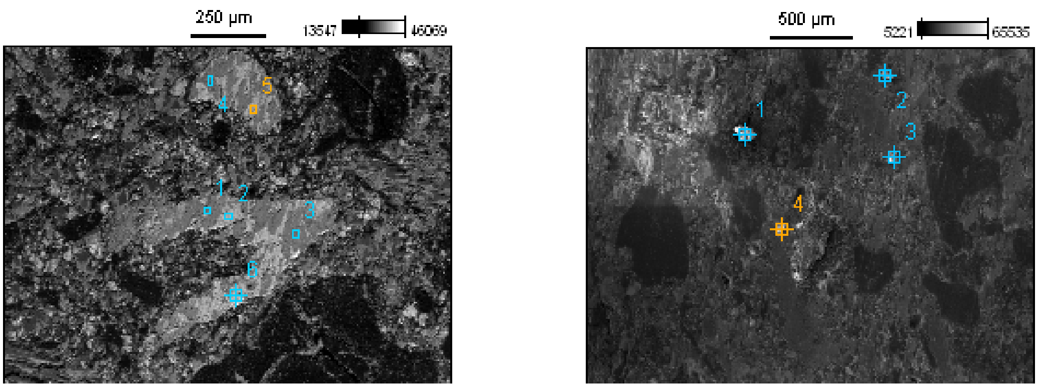
(b)



(c) Detail of the Fig. 7(a)

Fig. 7 SEM observations of the Sample 2 after the friction test

of the third body for the neighboring particles such as mineral element (Ba, Ca, Si, Al, S) and it can be observed at point 1, 3 and 4. These flat plates develop on the worn surfaces of metallic fillers extend outside by compaction of powder accumulated against them. This phenomenon causes the rise of the size of the latter, which reaches 500 μm. To further examine the second zone as illustrated in Fig. 7(b), the same rubbing surface is now analyzed by EDS and the result is shown in Fig. 8(b). From this figure, it shows the presence of elements of mineral constituents such as Si, Al, S, Ca and Ca elements (points 1and 2, Table 5). As shown in point 3, the third body of flat plates is formed on mineral fillers which is rich in Ba element coming from Barite particles. This constituent is considered as a friable particle which is easily worn out and it generates an important part of the third body composition. Whereas in some areas of the matrix, particularly at point 4 from Fig. 8(b) are uncovered and it is rich in C and O elements. In spite of a high level of temperature, which reaches the temperature of degradation of the matrix and leads to physical and mechanical modifications, the small quantity of powders and flat plates that presents on the rubbing surface shows that the third body generally leaves contact during friction. This explains the high level of friction coefficient of the composite material, even at higher temperature.



(a) Zone A (b) Zone B

Fig. 8 Details of zones located on Fig. 6

Table 4 X-EDS analysis of zones located on Fig. 8(a)

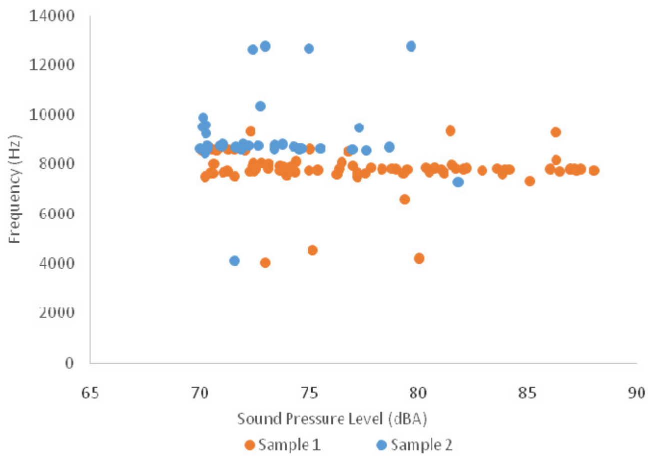
	C	O	Al	Si	S	Ca	Fe	Cu	Zn	Ba
pt1	10.81	20.85	0.83	4.59	1.77	3.42	34.96	8.11	5.35	9.31
pt2	7.77	5.45	----	1.33	----	0.41	4.28	50.01	30.74	----
pt3	11.30	20.20	1.06	4.46	1.13	3.11	31.74	13.21	6.21	7.45
pt4	11.84	21.03	0.77	5.01	1.29	3.23	35.35	8.26	4.46	8.76
pt5	8.65	5.67	----	0.91	----	0.53	4.26	47.44	32.54	----
pt6	5.71	1.97	0.97	0.21	----	----	1.15	53.13	36.88	----

Table 5 X-EDS analysis of zones located on Fig. 8(b)

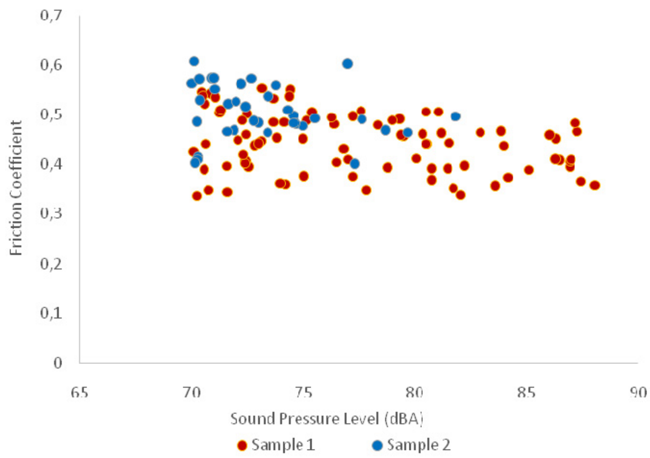
	C	O	Al	Si	S	Cl	K	Ca	Fe	Ba
pt1	56.02	17.87	0.66	2.11	3.77	1.92	1.61	3.81	2.51	9.72
pt2	50.90	16.07	0.23	0.83	0.53	14.14	---	3.24	0.93	6.07
pt3	13.94	30.87	0.50	1.73	0.57	15.73	0.43	3.32	1.39	20.35
pt4	40.37	22.58	0.24	0.74	0.43	14.43	---	2.04	---	3.80

3.3 Brake squeal behavior

As illustrated in Fig. 9(a), it is found that both samples predominantly generate squeal noise at frequencies between 7 kHz to 10 kHz. The interesting observation is that Sample 2 produces less squeal sound i.e., 80 dBA compared to 90 dBA for Sample 1. This finding agrees with the results obtained by Mat Lazim *et al*. 2014 where the presence of brass filler can help to damp the noise level. From Fig. 9(b), it is found that there is a slight difference in friction coefficient value during squeal events between Sample 1 (mu = 0.3~0.55) and Sample 2 (mu = 0.4 ~0.6). This agrees with the result that obtained in pin-on-disc tribometer test as shown in Fig.5. In addition, it indicates that brake squeal is indeed generated due to friction. Fig. 9(c) shows that squeals occur at a humidity range of 60 to 80 for Sample 1 and 40 to 75 for Sample 2. This slight difference may due to tribological aspects of the friction material as well as due to surrounding environment in the laboratory because of it is open dynamometer. In terms of the disc speed as given in Fig. 9(d), both

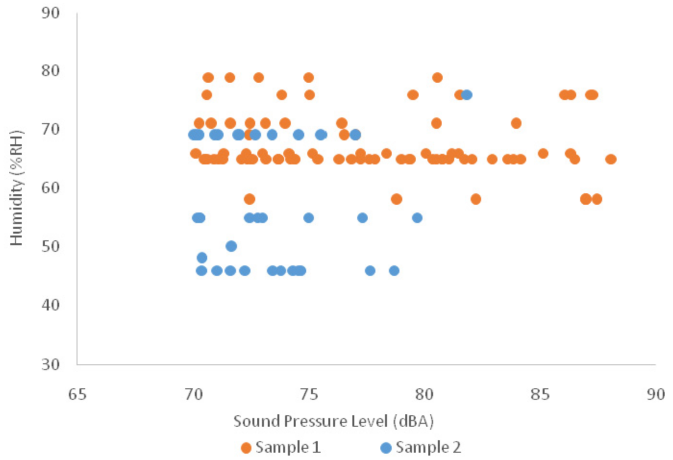


(a) Frequency versus Sound pressure level

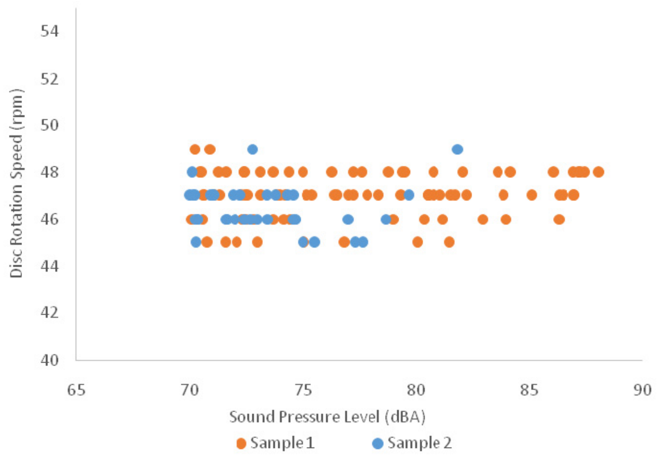


(b) Friction coefficient versus Sound pressure level

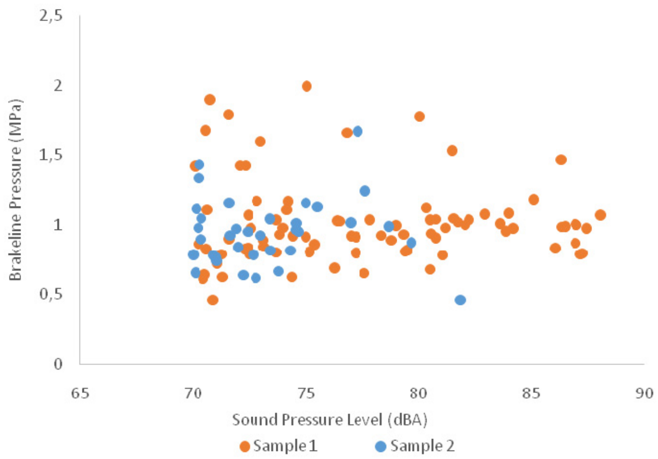
Fig. 9 Results of brake squeal test



(c) Humidity versus Sound pressure level

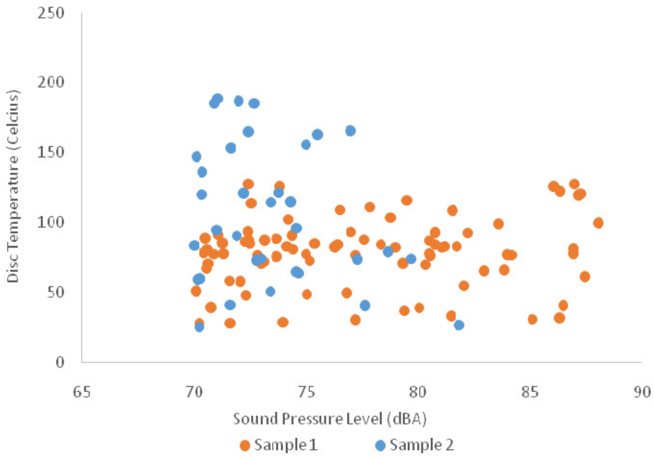


(d) Disc rotation speed versus Sound pressure level



(e) Brakeline pressure versus Sound pressure level

Fig. 9 Continued



(f) Disc temperature versus Sound pressure level

Fig. 9 Continued

samples generate squeals at almost identical levels i.e. between 40 to 50 rpm. This shows that squeal tends to occur at a very low speed of a vehicle, i.e. travelling in a busy traffic or slowing down for the traffic light. As in Fig. 9(e), both samples are largely producing a squeal noise at a hydraulic pressure between 0.5 MPa to 1.5 MPa which indicates light braking applications. Fig. 9(f) shows that Sample 1 experiences squeal at a temperature range of 25~140 degree Celsius while Sample 2 produces squeal at a bit higher temperature i.e. up to 200 degree Celsius. It shows that the brass filler may contribute to squeal generation at higher temperature. This is also observed in the pin-on-disc tribometer where Sample 2 reaches the highest temperature in a time shorter than Sample 1.

4. Conclusions

Based on tribological and dynamic studies conducted on composites containing brass filler, the following conclusions can be drawn:

- It is observed that 1.5% of brass filler influences the friction, wear and squeal noise performance significantly. Thus the brass filler seems to play a dual role; firstly as reinforcing element of the brake pad providing primary contact sites and secondly as solid lubricant by contributing to the formation of a layer of granular material providing velocity accommodation between the rotating disc and fixed pad.
- Brass filler determines conditions for friction force stabilization and smooth sliding behavior.
- Part of the wear debris remains in the contact area forming a discontinuous friction layer by filling troughs whereas the rest is released from the system as loose particles.
- The presence and absence of brass filler indeed affect squeal occurrences. It was found that the friction material without brass filler had slightly higher values in terms of humidity, brake hydraulic pressure and sound pressure level but slightly lower values in temperature and friction coefficient.

- There are strong correlations between pin-on-disc tribometer and brake dynamometer tests in terms of tribological aspect.

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