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NOISE CONTROL FOR A BETTER ENVIRONMENT

Test method for objective evaluation of automotive friction-induced squeak noises regarding complete vehicle conditions on a laboratory scale

Kathrin Arzt[1,2], Peter Lochbrunner[2], Olaf Hinrichsen [1,3]

[1] **Technical University of Munich, Department of Chemistry, Lichtenbergstr. 4, 85748 Garching b. München, Germany**

[2] **BMW Group, Knorrstr. 147, 80788 München, Germany**

[3] **Technical University of Munich, Catalysis Research Center, Ernst-Otto-Fischer-Str. 1, 85748 Garching b. München, Germany**

ABSTRACT

Due to the decreasing interior sound level in passenger cars, BSR noise (buzz, squeak, rattle) is perceived more and more intensely by the passengers. For car manufacturers, BSR noise is therefore one of the top quality issues.
In particular, squeaking noises are generated by friction-induced vibrations between components in relative motion. State-of-the-art setups for experimental stick-slip testing only simulate a two-dimensional motion between two components, which, however, cannot reproduce the characteristics of the complete vehicle system. Moreover, an adequate measurement of acoustic properties cannot be conducted due to the transient background noise of the setups.
The presented work introduces a new test method for the experimental simulation of automotive squeaking noise issues on a laboratory scale while incorporating the conditions present in real vehicles in order to predict stick-slip-risks in early development stages. The setup is based on a three-axis shaker system simulating the actual three-dimensional relative motion of two components in a vehicle. We show that a simultaneous measurement of acoustic and tribological properties is possible and distinct correlations between them can be found. A statistical study verifies the repeatability of the method with very high accuracy. Vibration characterization in the time and frequency domain suggests a four-group noise classification.

Keywords: friction-induced vibration, squeak noise, experimental measurement setup
I-INCE Classification of Subject Number: 72

1. INTRODUCTION

In recent years, the issue of buzz, squeak and rattle noise (BSR) has gained more and more attention in automotive applications since the interior sound levels in passenger cars

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[1] Kathrin.Arzt@bmw.de
[2] Peter.Lochbrunner@bmw.de
[3] Olaf.Hinrichsen@ch.tum.de

decreased continuously. BSR noises are assumed to be one of the top quality issues for car manufacturers nowadays [1,2].

While rattling noise issues can be identified and prevented by component gap and relative motion modeling [3,4], squeaking and squealing issues of the complete vehicle cannot be modeled reliably since the generation of vibrations and noise on tangentially moving surfaces is a complex coaction of numerous system parameters (tribological, physicochemical parameters) and still subject of current research [5,6]. Therefore, reliable testing of squeaking and squealing issues can thus far only be done on the complete vehicle. Since tests involving the complete vehicle exhibit several disadvantages like high costs and restricted countermeasures during the later stages of development, testing is also done with laboratory setups in earlier development stages. These setups imply a reduction of relevant boundary conditions, which influence the occurrence of friction-induced vibrations. For example, testing is only done with planar geometries or the relative motion between the test specimens is only performed one- or two-dimensionally with a constant normal force and, therefore, not comparable with the conditions of a complete vehicle. Objective acoustic measurements often cannot be performed due to the loud and transient background noise of the setups. In this study, a test setup is presented which is able to experimentally simulate all relevant boundary conditions present at the complete vehicle in order to enhance the reliability of laboratory friction-induced vibration testing. The test method is exemplified for two characteristic sample pairs used at the automotive door sealing system.

2. EXPERIMENTAL SETUP AND METHODOLOGY

2.1 Laboratory Test-Setup

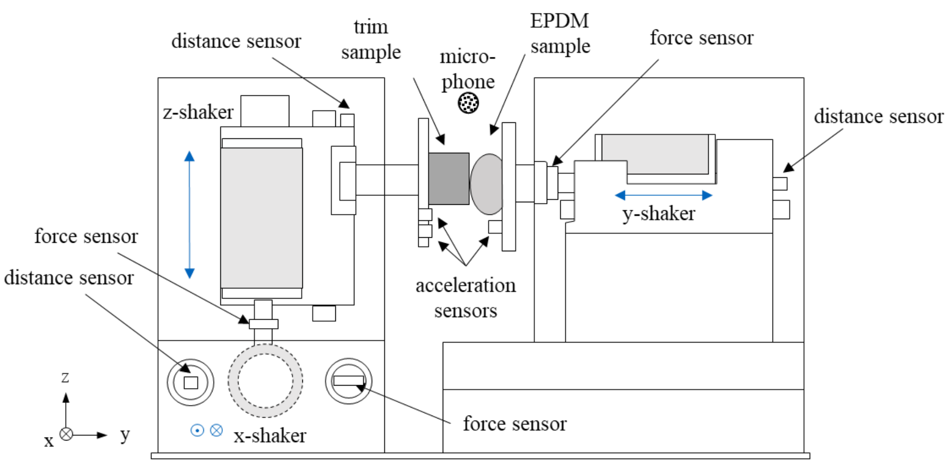

Figure 1: Laboratory test setup for examination of friction-induced vibrations.

For experimental investigation of automotive squeak issues, a test setup was developed, which is able to simulate a three-dimensional relative motion between two test specimens (Fig. 1). The general configuration of the setup consists of two electrodynamic shaker units, each of them exciting one sample fixture. One of the fixtures is excited in y-direction, whereas the other fixture is moved in combined x- and z-direction, consequently performing a three-dimensional relative motion between two samples mounted on the fixtures. The setpoint-signals for the excitation are measured in terms of three-axis relative acceleration waveforms on the complete vehicle and transferred to the test setup afterwards. Stochastic relative motions like they occur when driving on specific pavements (e.g. cobblestone, rough road) can be simulated via acceleration controlling in time domain. The acceleration signal is measured by three acceleration sensors mounted on the fixtures. These sensors are also able to detect potentially occurring, high frequency friction-induced vibrations between the samples. The relative displacement between the samples is measured by three magneto-inductive distance sensors located at the shaker axes. Moreover, the axes of the shakers are radially air-beared, so actuator friction can be neglected. The force to move the samples applied by the shakers is therefore assumed to equal the tangential and normal forces, respectively. Therefore, it is possible to measure kinetic coefficients of friction. Since the shaker units are located in housings the background noise of the setup is very low (< 31 dB) with a non-transient character. This allows to record the acoustic behavior of the sample pair via a microphone located 50 mm above the sample contact face.

The component pairs investigated in this study were two sealings used at the automotive door sealing system. On the one hand, a sample pair consisting of an EPDM gap seal (polyamide flock coating) and a polyurethan-coated trim strip, which is known to exhibit noises, was examined. On the other hand, an EPDM foam rubber edge protection in contact with a polyamide cover was investigated. This sample pair is known to not exhibit any noises.

The relative motion waveforms of a complete vehicle were measured on the complete vehicle for two different test tracks in preliminary studies. Test track 1 was a cobblestone pavement with a recording length of 35 s. Test track 2 was a rough road pavement with a recording length of 55 s.

2.2 Validation Measures

For validation of the simulation quality of the setup, four different error measures known from literature were used [7,8]. In order to characterize the linear relationship between the setpoint (≙ acceleration signal measured on complete vehicle) and the response signal (≙ acceleration signal measured on test setup), the normalized correlation coefficient r for the waveform signals was calculated according to Equation 1 by using the cross correlation in time domain:

$$r = \max[r(i)] = \frac{\frac{1}{N}\sum_{j=1}^{N} x_{j+i-1} \cdot y_j}{\sqrt{\frac{1}{N}\sum_{j=1}^{N} x_j^2 \cdot \frac{1}{N}\sum_{j=1}^{N} y_j^2}} \quad , -1 \leq r \leq 1 \tag{1}$$

Here, N represents the number of discrete signal points, x and y represent the signal values of the setpoint and response waveform, respectively, and i represents the signal point index.

Moreover, the simulation quality was characterized by the error of square sums, which compares the energy level of the setpoint and response signal.

The error of square sum is defined by Equation 2:

$$e(S^2) = \frac{\sum_{i=1}^{N} y_i^2 - \sum_{i=1}^{N} x_i^2}{\sum_{i=1}^{N} x_i^2} \tag{2}$$

Another important characterization is the local maximum relative error, averaged over the global minimum and maximum signal values. The error measure is expressed by Equation 3:

$$e(LMR) = \frac{1}{2}\left[\frac{\max(y_i - x_i)}{\max(x_i)} + \frac{\min(y_i - x_i)}{\min(x_i)}\right] \tag{3}$$

With definition of the $e(RMS)$ measure in Equation 4, a comparison of the setpoint and response signal can be done with regard to phase shifts between the signals. $e(RMS)$ is highly sensitive as the two signals are compared for each discrete signal point.

$$e(RMS) = \sqrt{\frac{\sum_{i=1}^{N}(y_i - x_i)^2}{\sum_{i=1}^{N} x_i)^2}} \tag{4}$$

3. RESULTS AND DISCUSSION

3.1 Road Excitation Simulation Quality and Laboratory Repeatability

In a first study an evaluation was performed on how accurate the relative motion data measured at the complete vehicle were simulated with the laboratory setup. For this purpose, the excitation signal was transferred to the setup in terms of a three-dimensional acceleration time signal dataset. This data is used as the setpoint signal in the following. In order to evaluate the simulation quality of the pure setup and the influence of a frictional sample contact, measurements were carried out both with and without the samples in contact for two different test track signals. The results of the tests are summarized in Table 1.

Table 1: Error measures for setup simulation quality of acceleration signals from complete vehicle. The respective signals marked with (a) and (b) are shown in Figure 2.

values in %		without sample contact				with sample contact			
	axis	r	e(S²)	e(LMR)	e(RMS)	r	e(S²)	e(LMR)	e(RMS)
test track 1 (cobblestone)	x	97.0	10.9	27.2	26.1	97.8	11.3	22.1	29.3
	y	97.7	10.9	29.9	21.4	96.5	11.2	33.5	22.8
	z	96.1	12.8	27.6	27.2	95.3	12.3	33.4	28.8 (a)
test track 2 (rough road)	x	97.7	17.6	30.1	24.5	96.5	16.4	31.5	27.8
	y	99.0	12.1	15.9	14.5	98.4	12.2	15.3	16.2 (b)
	z	97.5	20.0	23.6	24.0	97.0	20.3	25.0	27.3

With r being above 96% for all measurements a distinct linear relationship between the setpoint and the respective response time signals is identified. Comparing the error of square sums of the two signals values from 10 % to 20 % are achieved, which states an adequate simulation quality for both test track qualities. The average maximum relative error $e(LPM)$ shows peak differences of up to 32 %, while the time signal error $e(RMS)$

reaches maximum values of 29 %. The y-axis shows the lowest error values and, therefore, the best controllability, while the z-axis tends to show the highest error measures. This is associated to the higher instability of the z-axis due to the junction on the dynamically moving x-axis. In general, the error measures of the tests with the samples being in contact are slightly higher than without being in contact. This indicates a minor influence of the frictional contact on the simulation quality as the setup controller is not able to compensate all of the sample interactions.

Since the $e(RMS)$ is highly sensitive to any deviation of the setpoint and response signal, a detailed comparison of the time signals with the highest and the lowest error measures (for the tests with sample contact) are shown in Figure 2. For both tests, the very high accuracy of the experimental simulation can be seen, even for the worst case measurement in Figure 2 (a). Essentially, the differences in time domain are caused by peak deviations, which is associated to sample interactions of the highly elastic contact. This is confirmed by the comparatively high $e(LMR)$ values in Table 1. In summary, it is shown that the relative motion measured at the complete vehicle can be simulated on the laboratory setup with a very high accuracy.

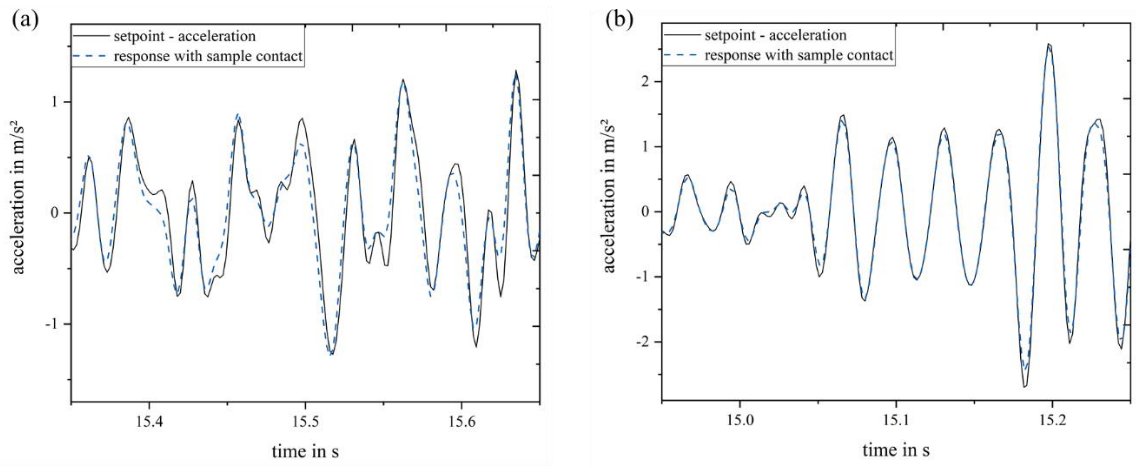

Figure 2: Comparison of details in the time signal of the acceleration setpoint and the response for (a) test track 1 in z-direction with r=95.3% and (b) test track 2 in y-direction with r=98.4%.

Besides the simulation quality the repeatability of the relative motion data is essential in order to validate the laboratory setup. For evaluation of the excitation repeatability of the setup, equal measurements were carried out consecutively with the number of iterations being $N = 31$. In order to reduce the time signal data volume, the root mean square (rms) of the measured acceleration signals was defined as an evaluation measure. Figure 3 shows box plots of the measurements for each of the three axis. The measurements show a very high repeatability with Gaussian distributions and the coefficients of variation ν (relative standard deviation) being 2.89 %, 0.55 % and 1.23 % for the x-, y- and z-direction, respectively.

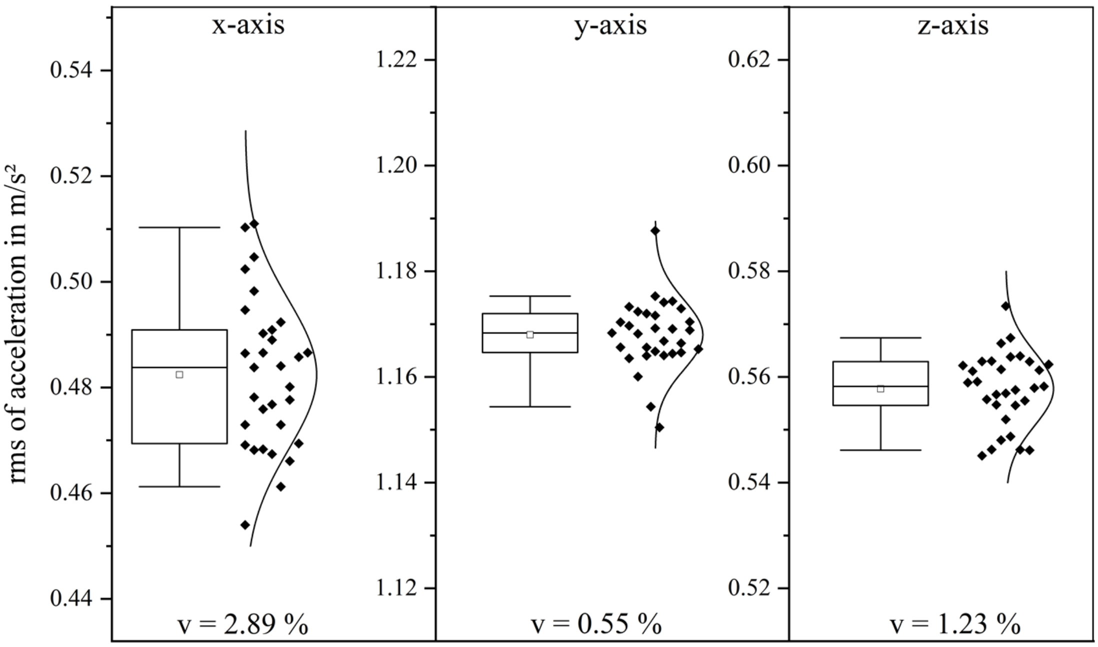

Figure 3: Repeatability of the excitation acceleration root mean square for x-, y- and z-shaker axis with the samples being in contact.

3.2 Detection of Friction-Induced Vibrations

Preliminary studies showed that the relative motions for automotive door sealing applications exhibit relevant frequencies (f) up to 80 Hz. Therefore, the setpoint signal for the laboratory setup was resampled to 170 Hz. According to the Nyquist-Shannon sampling theorem this ensures that the setup controller only regulates up to 85 Hz and does not regulate vibrations of higher frequencies which are induced by the frictional contact of the samples. Consequently, friction-induced vibration with $f > 85$ Hz are not compensated by the axes controlling system.

For detection of friction-induced vibrations, tests with two different sample pairs were carried out. One of the sample pairs exhibited squeaking noise, whereas the other one didn’t exhibit noise. The occurrence of noise was determined by evaluation of the sound signal of the attached microphone. Figure 4a shows a detail comparison of the setpoint excitation signal and the response signals of the two sample pairs. The sample pair without generated noise continues along the setpoint time signal. For the sample with noise generation, high frequency interferences can be observed. A comparison of the power spectral densities (PSD) of a sample pair generating noise and a sample pair without noise generation is shown in Figure 4b. The PSD content lower than 85 Hz can be assigned to the excitation acceleration (setpoint). When friction-induced vibrations are generated, vibrations with $f > 100$ Hz can be determined for all three axis. For the sample pair, which does not exhibit noises, no vibrations in this area can be found. Consequently, friction-induced vibrations can be detected by high-pass filtering of the acceleration raw signal in order to separate the excitation acceleration and induced vibrations.

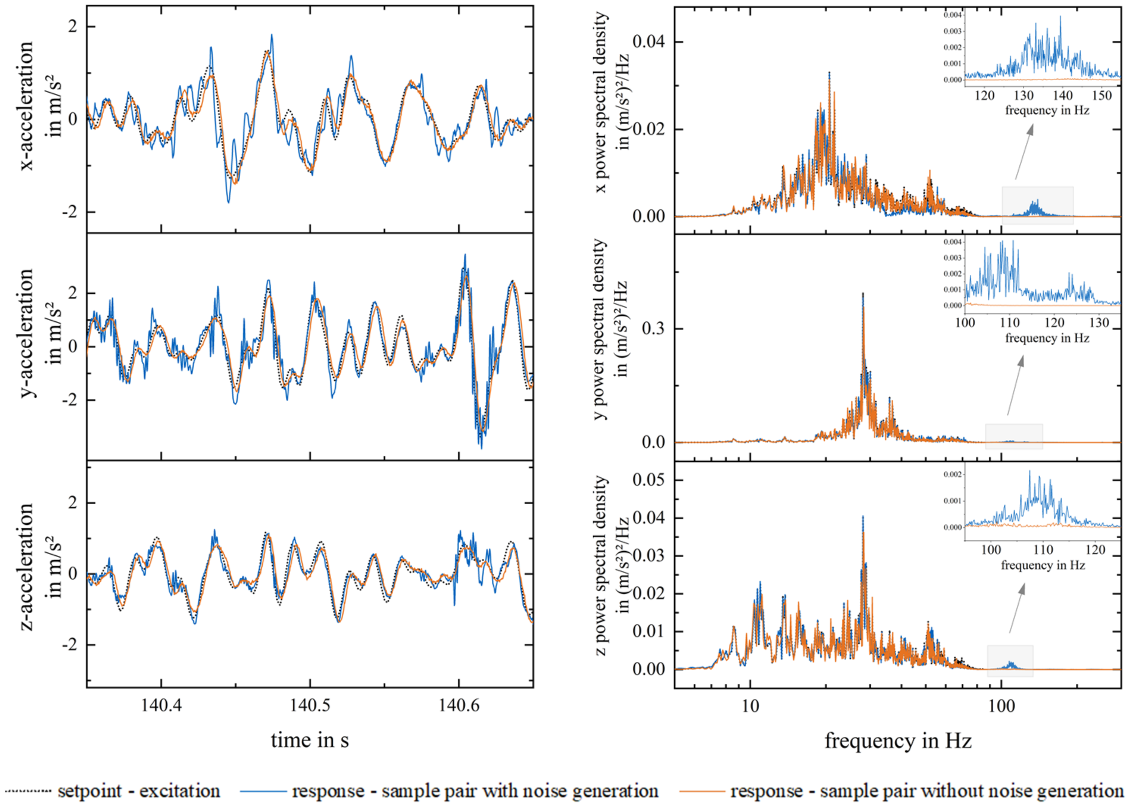

Figure 4: Time signal (a) and power spectral density spectrum (b) for sample test with and without generation of friction induced vibrations.

For the examined EPDM sealing samples high-pass filtering with a filter frequency of 100 Hz has been carried out. The remaining time signal only shows the content of the friction-induced vibrations and is defined as stick-slip acceleration in the following.

Table 2: Results of the repeatability study for a sample pair with and without generation of friction-induced vibrations.

		sample pair without noise generation			sample pair with noise generation		
		x	y	z	x	y	z
rms of stick-slip acceleration	absolute SD in m/s²	0.001	0.001	0.001	0.005	0.011	0.004
	relative SD in %	3.91	4.23	7.83	2.80	4.80	4.74
N10 of stick-slip acceleration	absolute SD in m/s²	0.001	0.001	0.001	0.006	0.008	0.004
	relative SD in %	2.96	2.71	2.00	2.59	3.13	3.37

Since the analysis and comparison of time signal data requires a high effort, a reduction of the data volume is needed in order to create an efficient method for detection and characterization of friction-induced vibrations. For analyses, the rms and N10 percentile of these signals were used as characteristic measures for the occurrence of friction-induced vibrations. These measures can be produced with a high repeatability as shown in Table 2, where 30 equal tests for a sample pair with and without noise generation were performed each. The relative standard deviations exhibit values below 3.5 % and, therefore, the rms and N10 percentile are suitable measures for detecting the occurrence of vibrations.

3.3 Evaluation of Generated Noise by means of Acceleration Measurements

Due to the housing of the shaker axes the laboratory setup has a very low and non-transient background noise (average sound level below 31 dB). This enables an evaluation of simultaneously measured acceleration and sound signals generated by the frictional contact of the sample pairs. In Figure 5 the relationship of the stick-slip accelerations and the properties of the generated noise is depicted. As a measure for the acceleration associated with vibrations, the sum of the rms values of the x-, y- and z-stick-slip accelerations was used. For the acoustic evaluation, the difference between the rms values of the sound levels of the pure setup noise (recorded when the samples surfaces are not in contact) and the noise of the samples being in contact was used. A distinct linear correlation between the two measures can be found with the correlation coefficient being $r = 0.98975$.

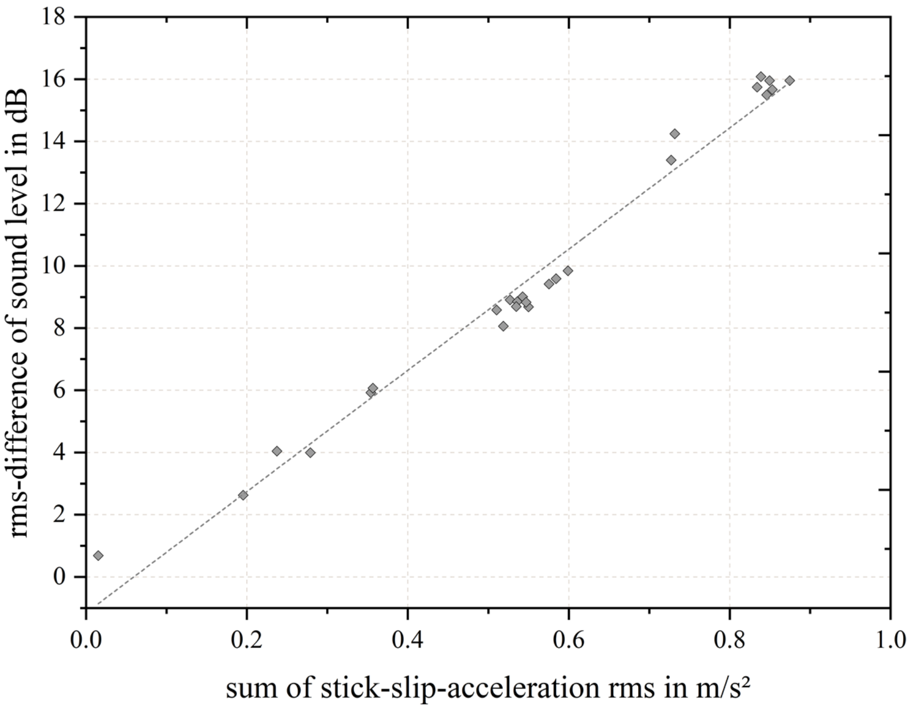

Figure 5: Correlation of acceleration and sound signal of the laboratory test setup.

3.4 Characterization of Friction-Induced Vibrations

For an evaluation of the frictional and acoustic behavior of a sample pair, an automatic characterization can be performed based on two parameters. On the one hand, the friction-induced vibrations and the sound pressure can be distinguished by the overall occurrence and intensity of vibration events, which can be expressed by the rms-sum of the three-dimensional stick-slip accelerations or the rms value of the sound pressure waveform, respectively. The rms increases with the number of vibration events and the amplitudes of the vibrations. On the other hand, the transience of the vibration events can be characterized by the sum of the excess kurtosis values in x-, y- and z-direction, where the excess kurtosis increases when exhibiting a more transient signal. In Figure 6, the results of this characterization is shown. Four different classes can be distinguished:

[1] low rms / low kurtosis: no friction-induced vibrations, no noise generation

[2] high rms / low kurtosis: friction-induced vibrations and noise with persistent character

[3] high rms / low kurtosis: friction-induced vibrations and noise with transient character

[4] low rms / high kurtosis: single-slip events

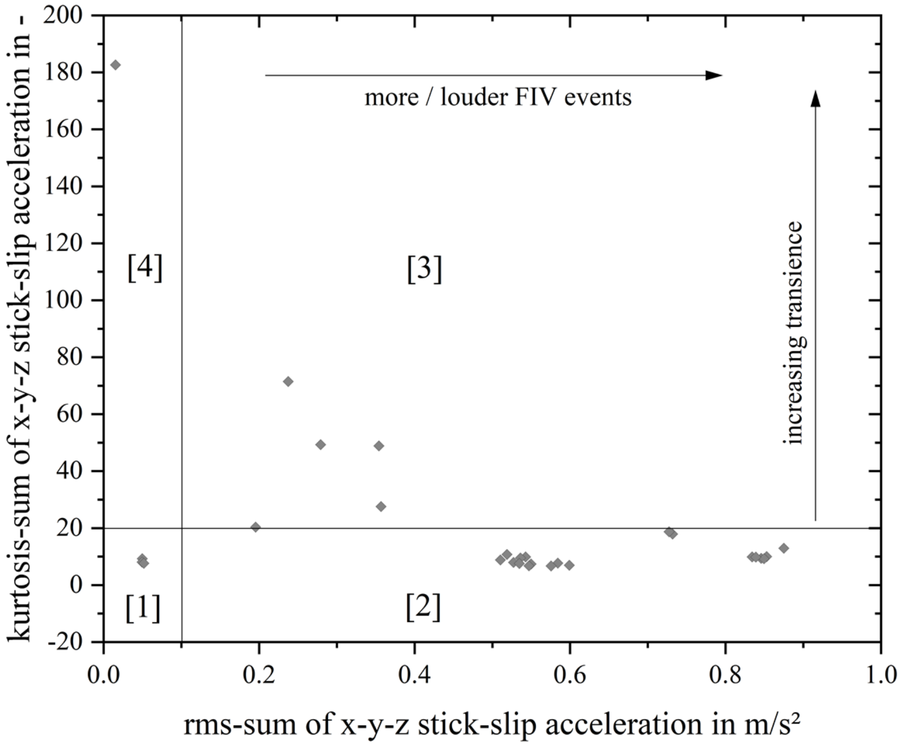

Figure 6: Characterization of friction-induced vibrations test results based on rms and excess kurtosis values.

4. CONCLUSIONS

The present work introduces a laboratory test setup for the objective evaluation of automotive friction-induced vibrations. The setup is able to incorporate complete vehicle conditions since tests can be performed with the component geometries of a real vehicle and the incorporation of relative motions of a real vehicle. It is shown that the three-dimensional relative motions measured at the complete vehicle can be accurately and repeatably simulated with the laboratory setup, which advances to the reliability of the laboratory tests dramatically. The occurrence of friction-induced vibrations can be detected by separating the excitation accelerations from higher frequency components.

The correlation of the sound and acceleration data exhibits a highly linear relationship. Therefore, the detection of friction-induced noise events can be done by pure evaluation of the acceleration signals. Moreover, the characterization of the vibration events can be done by introducing the excess kurtosis and rms values of all three dimensions as parameters, resulting in a four type classification of the vibration events.

5. ACKNOWLEDGEMENTS

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