Experimental study of the contribution of gear tooth finishing processes to friction noise

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Keywords:
Friction noise
Multiscale surface characterization
Dry and lubricated contact
Finishing processes

A B S T R A C T

Micro geometry of a gear tooth influences the contact durability and wear performance. In this paper, different gear tooth flanks have been manufactured by different finishing processes, which were then characterized using multiscale surface analysis, based on wavelet transform. The friction noise was then measured before and after meshing in dry and lubricated conditions, to quantify the acoustic performance of the surfaces. To accomplish this objective, a new non-destructive sensory measurement technique was developed to characterize the friction noise generated by teeth flank surface. Results show the ability of the new method to discriminate functional behavior of different surfaces as well as give possible explanations as to the contribution of tooth flank asperities during the meshing on the gear in terms of gear noise performances.

1. Introduction

In applications where there are two bodies in contact, the characteristics of their surfaces influence heavily the functional performance, be it the friction and wear between the two [1,2] or the vibrations generated [3,4]. In gear mechanics, contact between teeth occurs periodically with the meshing. Friction is generated during most of the contact except at the primitive diameter, where the relative friction speeds are inverted [5]. For spur and helical gears, the contact is theoretically linear though the load on the teeth makes it more of an area. The influence of the macro-geometry of the teeth on gear noise has been largely studied [6–9] but the impact of lower scales is still not completely understood [10].

The dynamics of contact, be it in gears or otherwise, can be approached acoustically by studying the friction noise [11–15]. Friction noise can be classified in two categories according to the contact load. In high load contacts, the noise is generated by mechanical instabilities such as sprag-slip and stick-slip. In low load contacts, the surface roughness becomes crucial and thus this type of noise is called roughness noise [15]. The radiated vibrations are then attributed to the impacts between the asperities of the surfaces in contact. In order to discriminate surfaces directly considering their functional performance, we developed a new method to qualify and quantify friction noise. It is then used to determine its influence on gear vibrations in order to solve the issues directly at the source. Indeed, the actual gear vibration tests are quite costly and time-consuming if done at industrial level.

In the present study, the influence of the tooth finishing process on friction noise has been studied, as well as the effect of lubrication conditions and meshing on surface modifications. The topographic teeth surface modification during the meshing tests has been used as signature of the considered finishing process on the teeth surface. The relationship between the surface irregularities, the lubrication and the generated vibrations were analyzed in a wide range of wavelength from roughness to waviness.

2. Multiscale signature of finished surfaces

The surfaces studied are the tooth flanks of an automotive powertrain transmission gear. Three configurations were considered after the carbonitriding and the shot-peening operations: finished by an abrasive

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http://dx.doi.org/10.1016/j.triboint.2017.05.004

process, powerhoning or grinding, and not finished. The macro-geometric parameters of the teeth are the same. To access the three-dimensional texture of the flank surfaces, non-destructive replicates of the primary shaft's teeth using a silicone-based resin (Struers, Repliset F1) were made. The topography measurements with a white-light interferometer (WYKO 3300 NT –WLI)) were situated on the primitive diameter of the teeth. The topographies were sampled at 515 × 515 points with a 3.88 μm step in both x and y directions. The form error component was removed from the acquired 3D data using least squared approximation method based on a bi-cubic spline function.

Examples of 3D topographies of the three configurations (without finishing operation, after grinding and after powerhoning) before the meshing test are represented in Fig. 1. It is particularly clear that the surface not finished (Fig. 1a) is much rougher than the other two. While the finished surfaces seem to have similar roughness, their morphologies are quite different, due to differences in the process kinematics. On powerhoned surfaces (Fig. 1b), grooves appear to be curved downwards, the angle with the profile direction (vertical axis) being much smaller along the tooth flank. The grooves generated by grinding (Fig. 1c) follow the helix direction.

These topographies were then characterized using parameters issued from the ISO 25178 standard (Fig. 2). As seen on the topographies, the 3D aeral arithmetic average roughness (Sa) between finished and un-finished surfaces is quite different, with a factor of two between them. This parameter doesn't discriminate clearly between powerhoned and grinded surfaces as their roughness are close and standard deviations overlap. The three main parameters of the bearing curve were also studied. The core roughness depth (Sk) is a measure of the surface with the predominant peaks and valleys removed. The Reduced Valley Depth (Svk) is a measure of the valley depth below the core roughness while the Reduced Peak Height (Spk) is a measure of the peak height above the core roughness. The differences between powerhoned and grinded surfaces are not significant for the three topographic elements.

However, as several operations were used to obtain the final topographies of the flanks, there is a need to characterize the surfaces on a large wavelength band to enhance discrimination between generated surfaces. Wavelets transform was then used in the present study to decompose surface topography at various scale from roughness to waviness [1,3,16]. The analysis was done along the profile direction of the teeth. Then, the SMa spectra regrouping the arithmetical roughness average of surface components h_a at each scale "a" is computed [17,18] according to the following equation:

$$SMa(a) = \sum_{x=1}^{M} \sum_{y=1}^{N} \frac{|h_a(x,y)|}{MN} \tag{1}$$

where M and N are the analyzed surface sizes.

Fig. 3 shows an example of these surface roughness spectra on pieces directly after manufacturing. It depicts clearly that grinded surfaces

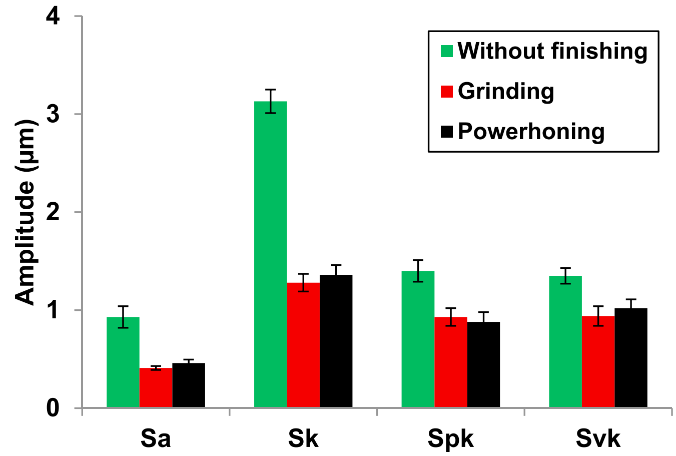

Fig. 2. ISO 25178 3D areal surface parameters of gear tooth flanks for the three studied configurations.

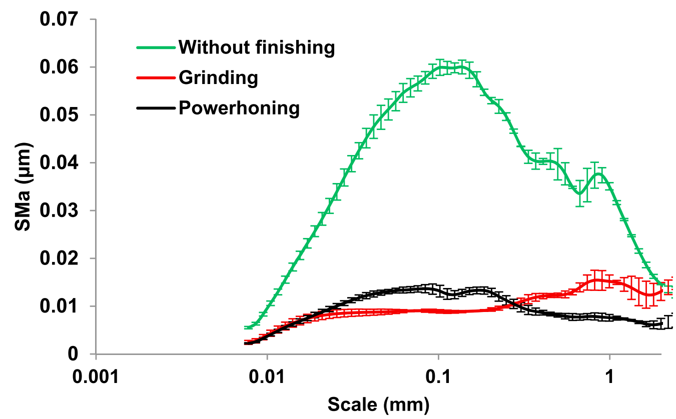

Fig. 3. Example of surface roughness spectra for three considered gears surface finishes before gear testing.

exhibit more waviness than powerhoned ones which are, on the contrary, rougher. Thus, surface texture components of grinded tooth are more spaced than of powerhoned ones.

3. Friction noise measurements on finished surfaces

In order to discriminate the surfaces according to their performance in term of vibrations, a new sensory method was developed. Friction noise, the vibrations generated by the contact asperities, measurements were made using a polymer stylus with a radius of curvature of 6 mm

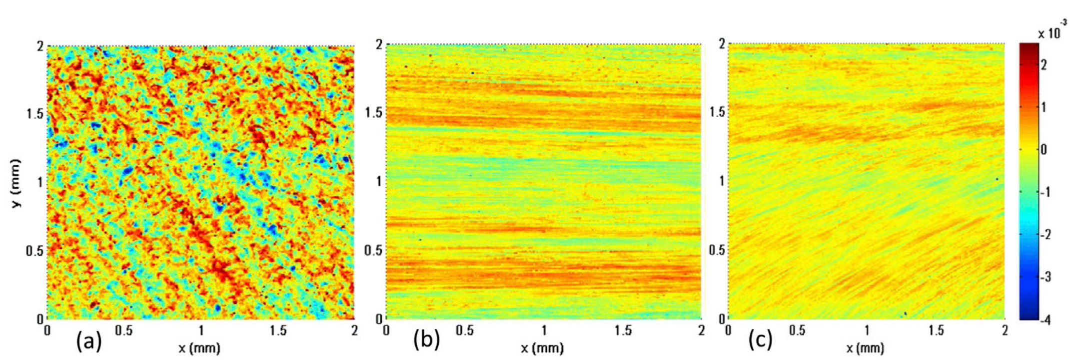

Fig. 1. 3D micro-topographies (2 mm × 2 mm) of tooth surfaces generated respectively (a) after shot peening (without hard finishing), (b) by power honing and (c) by grinding. The color scale is in millimeters. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

which was put into contact with the flank (Fig. 4). This probe applies a load equal to 0.3 N on the tooth surface during the friction test. The course was about 3 mm long in the helix direction and the friction speeds were 10 mm s^{-1} and 20 mm s^{-1}. In order to verify the repeatability of the experiments and to assure the representativeness of the results, each test was repeated five times and six teeth by gear were measured.

The average vibratory levels were computed. In order to analyze the results more finely, we used the Empiric Mode Decomposition (EMD) [19]. Each mode is characterized by its main frequency i.e. the one with the maximal amplitude. The results were then grouped as described in Fig. 5.

Friction noise was measured before and after 2 h meshing on a low power test rig, at a 1500 rpm speed and 8 Nm load, which was described in a previous study [4]. Dry and wet regimes using two different lubricant respectively “Oil A” (80W90) and “Oil B” (5W20) were considered. Over the 2 h meshing period, the temperature varied averagely from 20 to 33 °C. In this temperature range, the dynamic viscosities varied from 40 to 70 cP for the “Oil B” and from 250 to 500 cP for the “Oil A” (Fig. 6). Three gears were tested for each configuration.

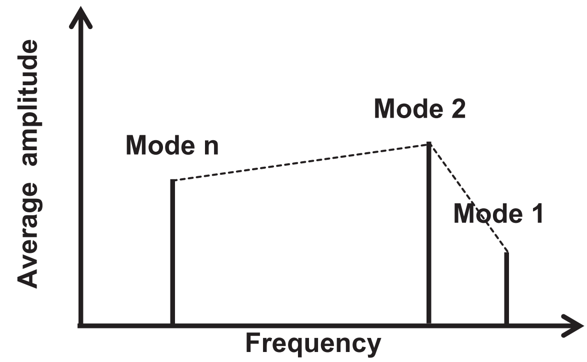

Fig. 5. Spectrum of the mean amplitude of the main mode as a function of its frequency.

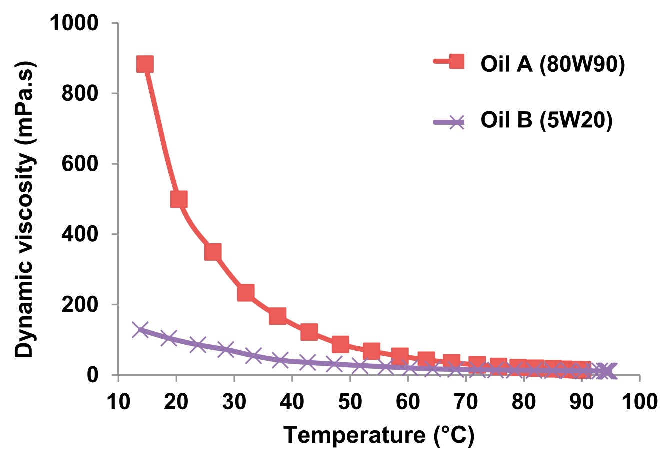

Fig. 6. The dynamic viscosity functions of temperature of the two considered oil.

4. Results and discussion

4.1. Influence of the finishing process on friction noise

First and foremost, the influence of the finishing process choice was investigated. In order to explain this, measurements of the noise generated by the surface asperities were made on the pieces for all three processes. Fig. 7 shows the raw signal measured for each process. Qualitatively, it is interesting to note that the average amplitude of the vibratory signal when the flanks are not finished (Fig. 7a) is higher than for the other two, grinding in Fig. 7b and powerhoning in Fig. 7c, which are very close. Their respective power spectral densities (PSD) are given in Fig. 8. In this figure, the not finished flanks give clearly higher amplitude than the other two processes in the all analyzed frequency range. Also, a difference appears between the two finishing processes. Indeed, at frequencies lower than 300 Hz, the powerhoned flanks generate less noise than the grinded ones. However, this tendency is inverted at higher frequency.

Fig. 9 shows the friction noise results after empirical modes decompositions. The levels of the modes for the non-finished surface are clearly higher than for the finished ones. As these measurements on the pieces were made right after being manufactured, the initial states of the surfaces used for the tests, their SMa, are shown on Fig. 3. It is interesting to note that the inversion of the main modes' trends (Fig. 9) between the powerhoning and the grinding processes is well represented on their surface signatures shown in Fig. 3. Indeed, the grinding process gives higher modes in the lower frequencies than the powerhoning, which correlates directly with the higher amplitude of the SMa in the higher scales related to the waviness components, of the flank surfaces.

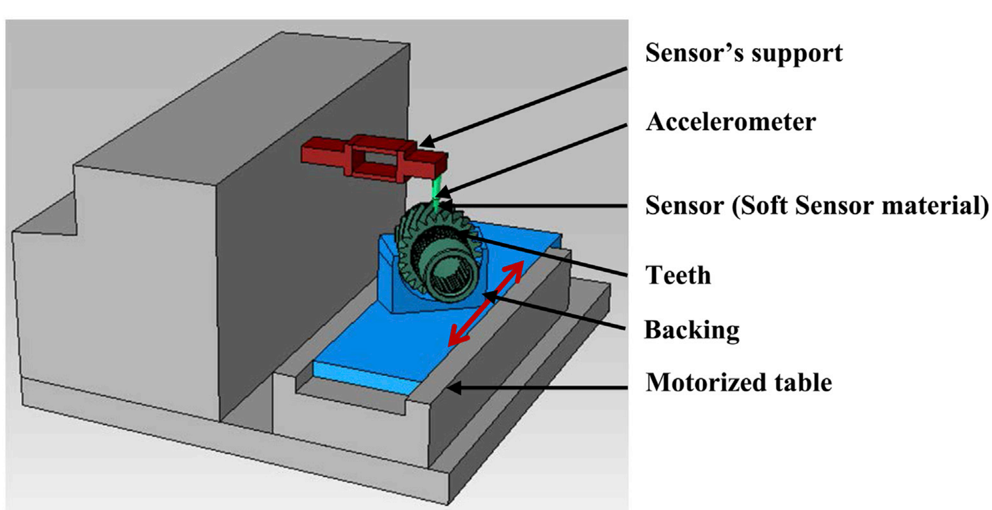

Fig. 4. Drawing of the support, workpiece and sensor.

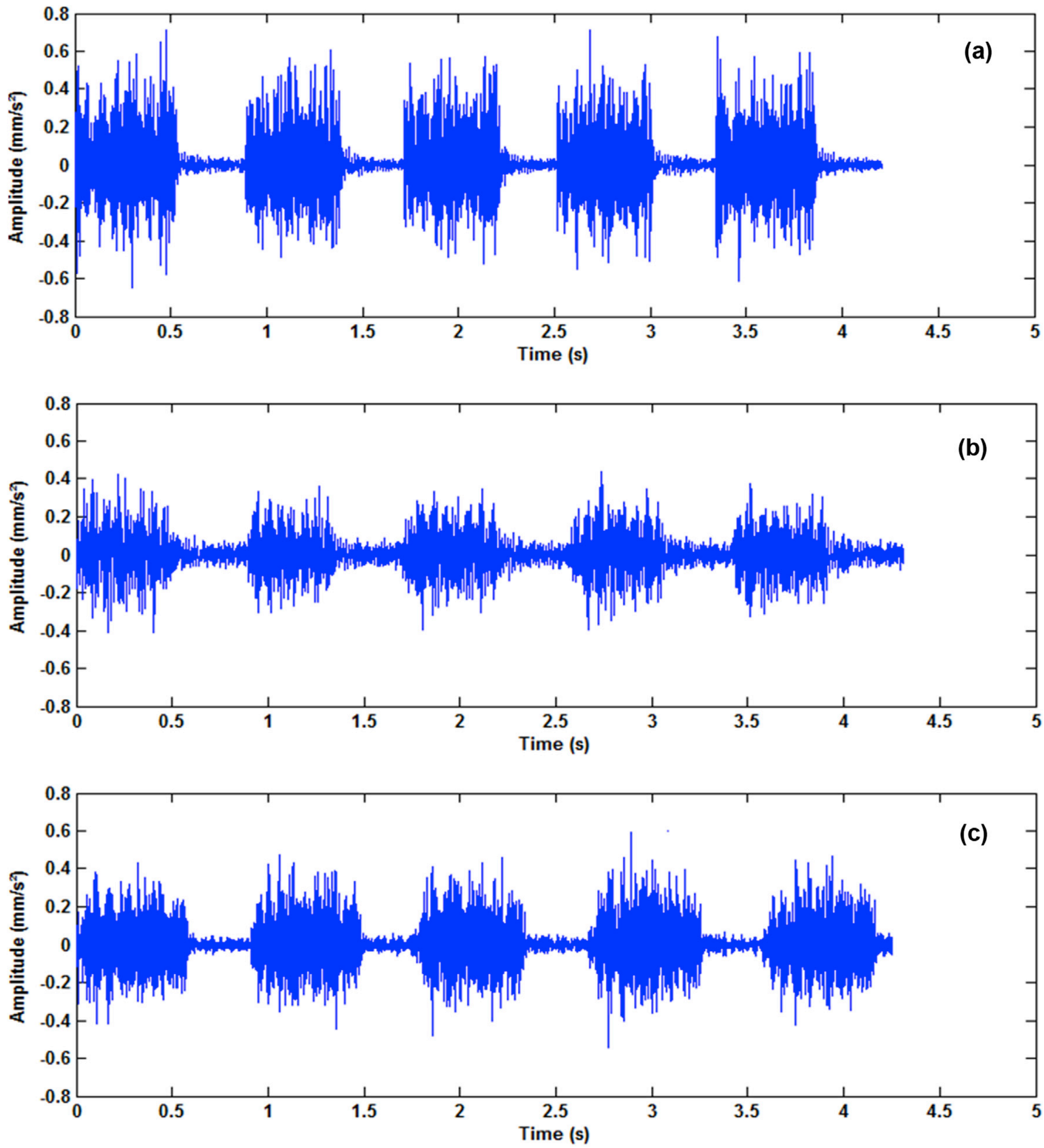

Fig. 7. Measured raw vibratory signal at 10 mm s^{-1} for (a) not finished, (b) grinded and (c) powerhoned tooth surfaces.

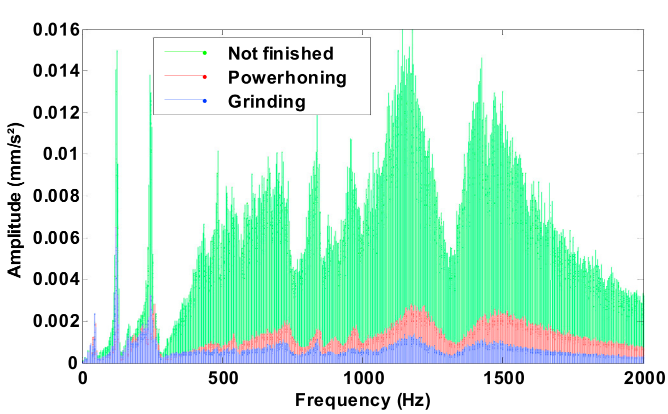

Fig. 8. Power Spectral Density (PSD) for each measured signal for measurements at 10 mm s^{-1}.

Inversely, the higher amplitudes of the SMa for the powerhoning concerns the roughness scales and thus generate modes in the higher frequencies.

4.2. Influence of the meshing and lubricant on friction noise

Fig. 10 show 3D tooth surfaces topographies of the three configurations (without finishing operation, after grinding and after powerhoning) before the meshing test (a) after shot peening (without hard finishing), (b) by power honing and (c) by grinding, after meshing (1) in dry condition, in wet condition using (2) oil A and (3) oil B.

With measurements done after meshing on the test rig, we were able to study the effect of lubrication as well as the meshing itself on friction noise. Without finishing (Fig. 11a), the meshing has a clear impact on friction noise. Indeed, in the frequency range from 100 Hz to 1 KHz, the amplitude decreases from 30 to 60% after meshing. It is interesting to

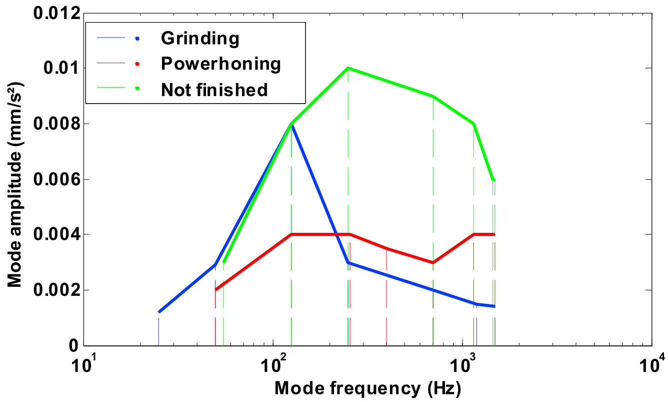

Fig. 9. Amplitudes and frequencies of the friction noise modes and its associated amplitude for measurements at 10 mm s^{-1}, for each studied process before any meshing on the test rig [20].

note that this decrease is more important for lower viscous lubricant. A decrease in friction noise can also be seen on finished flanks (Fig. 11b and c), but on a narrower frequency band, from 100 to 300 Hz, and the lubricant doesn't seem to affect the amplitude in this case. At higher frequencies, friction noise amplitude is higher after meshing, indicating a change in higher surface scales.

Moreover, we can also note that the grinded and powerhoned flanks behaviors are coming closer together for every mode when the viscosity is decreasing. It is particularly visible at around 100 Hz frequencies and the ones above 700 Hz. For these finished pieces, the comparison of the SMa for each lubrication condition is given in Fig. 12.

The inversion scale of the tendencies between the two processes isn't modified by the meshing, with an average around 0.28 mm and a standard deviation of 0.04 mm. The general tendency with wear and the diminution of the lubricant's viscosity is that the SMa is coming closer for the scales lower than 0.2 mm. This behavior correlates with the observation done previously on friction noise measurements and its behavior at higher frequency. Finally, we can conclude that the lubricant (or lack thereof) has more impact on friction noise for not finished teeth than finished ones. Therefore, the meshing leading to surfaces accommodation is the major influent factor.

4.3. Relationship between contact kinematics, vibration modes and surface scales

Figs. 13 and 14 gives the amplitudes of the main modes for the

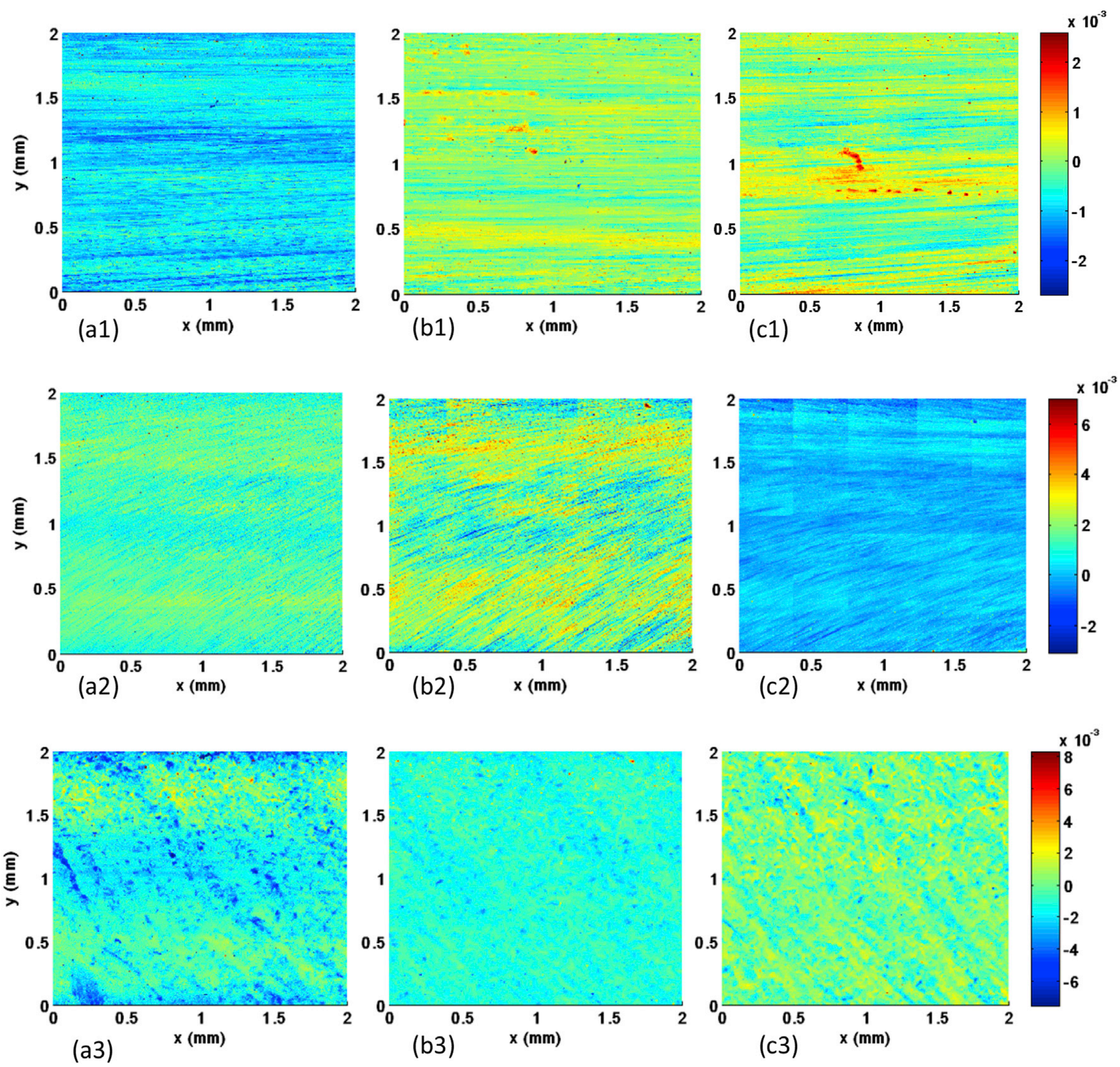

Fig. 10. 3D micro-topographies (2 mm × 2 mm) of tooth surfaces generated respectively (a) after shot peening (without hard finishing), (b) by power honing and (c) by grinding, after meshing (1) in dry condition, in wet condition using (2) oil A and (3) oil B. The color scale is in millimeters. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

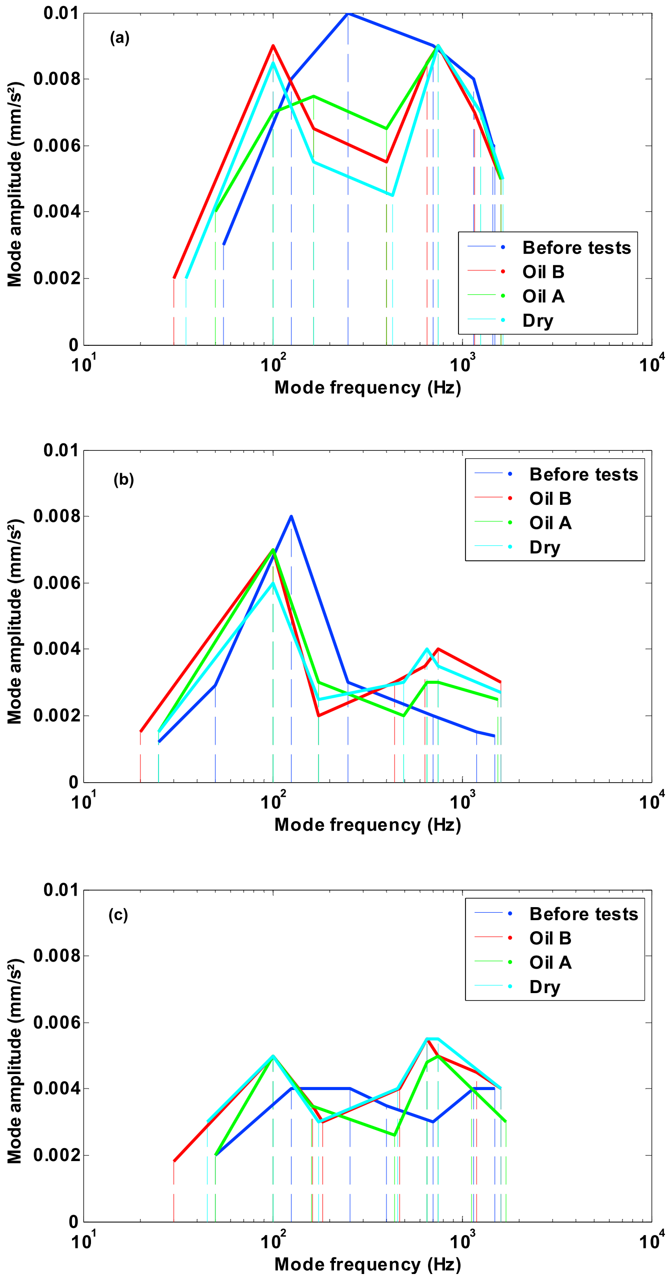

Fig. 11. Amplitudes and frequencies of the friction noise modes and its associated amplitude for measurements at 10 mm s^{-1} and each considered test configuration: before and after meshing in different lubrication conditions for (a) not finished (b) finished by grinding and (c) finished by powerhoning gear pieces.

measurements on the pieces after meshing on the test rig in each lubrication condition respectively for a contact speed of 10 mm s^{-1} and 20 mm s^{-1}. The increase of friction speed solicits more the higher friction modes. In fact, by increases friction speed, the principal (highest amplitude) friction mode translates from 250 to 750 Hz. Besides, it is interesting to note that when the oil is viscous (Fig. 14a), the modes for the grinded and powerhoned teeth are very close to each other in the lower frequencies, below 700 Hz. This is less the case in the other configurations (Fig. 14b and c) where grinded pieces give higher amplitudes. In the highest frequencies, the two finished configurations are close in terms of amplitude but the powerhoned surfaces give higher amplitudes. In the end, the choice of the lubrication type is imposed by the quality of the teeth finishing to reduce contact vibrations. For not finished teeth,

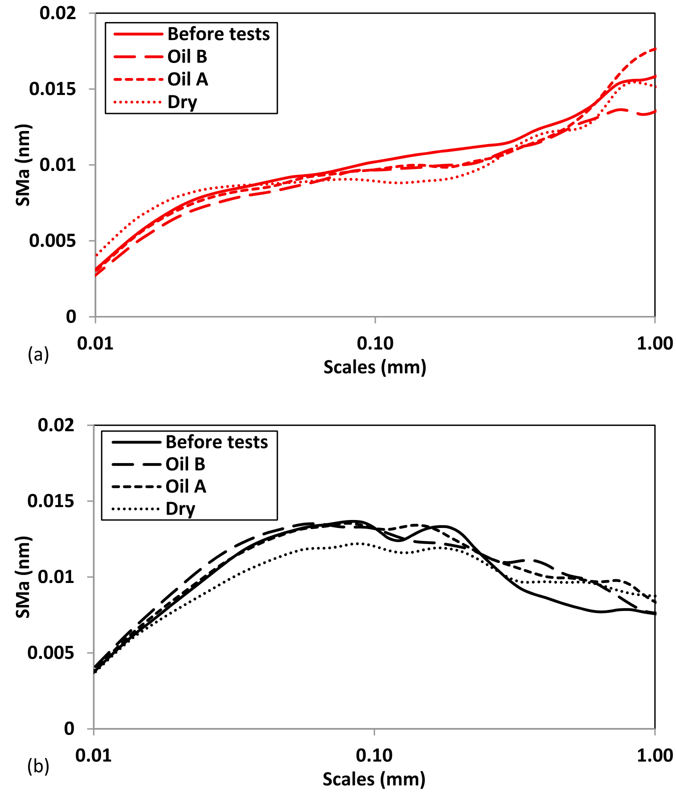

Fig. 12. Multiscale roughness spectra, SMa, of respectively (a) grinded and (b) powerhoned pieces before meshing and after meshing in various conditions.

friction vibrations were more attenuated after meshing using more viscous oil. However, the influence of lubricant viscosity is less apparent for finished teeth. By reducing teeth surface asperities with a hard finishing process, the required lubricant viscosity for noise reduction is also reduced.

In order to establish the relationship between the main friction noise mode and the roughness characteristic scale, the highest scales on the SMa spectra have been plotted as a function of the main mode frequency. The obtained graphics are given in Fig. 15.

From these, we used an inverse model for which the "α" coefficient was determined through the least squared method. The X^2 obtained are respectively of 1.15% and 5.1% and the model is given by Eq. (2), where the surface scale is in millimeters, the friction speed in millimeters per second and the frequency in Hertz.

$$Scale = \alpha \cdot \frac{Friction\ speed}{Frequency} \tag{2}$$

"α" is a dimensionless constant depending on the contact geometry and also on the sensor. In our case, "α" was equal to 6.15 ± 0.08.

From there, this equation was extrapolated to higher speeds, for example the ones happening in the meshing on our test rig. At 1500 rpm, the friction speeds varied from 0 to 900 mm s^{-1}. By applying the presented model to these conditions, waviness scale of the millimeter order will intervene in gear vibrations. This is in accordance with previous study results on gear vibration as we've shown that waviness scales had a great impact on gear noise behavior [4].

5. Conclusions

The new experimental method developed in this work, allows the qualification and quantification of the generated frictional noise on

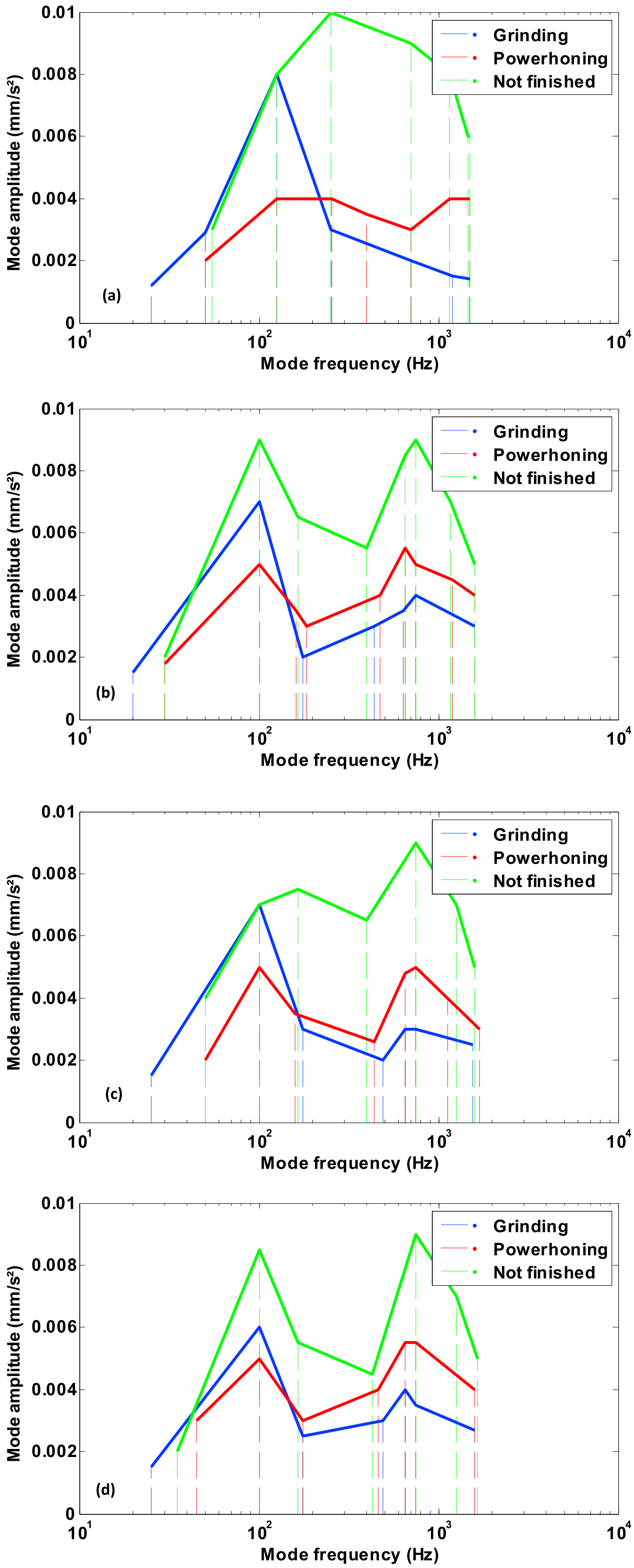

Fig. 13. Frequencies of the friction noise modes and its associated amplitude for measurements at 10 mm s^{-1}, for each considered test configuration: (a) before meshing; (b) after meshing using oil A; (c) after meshing using oil B; (d) after meshing in dry condition.

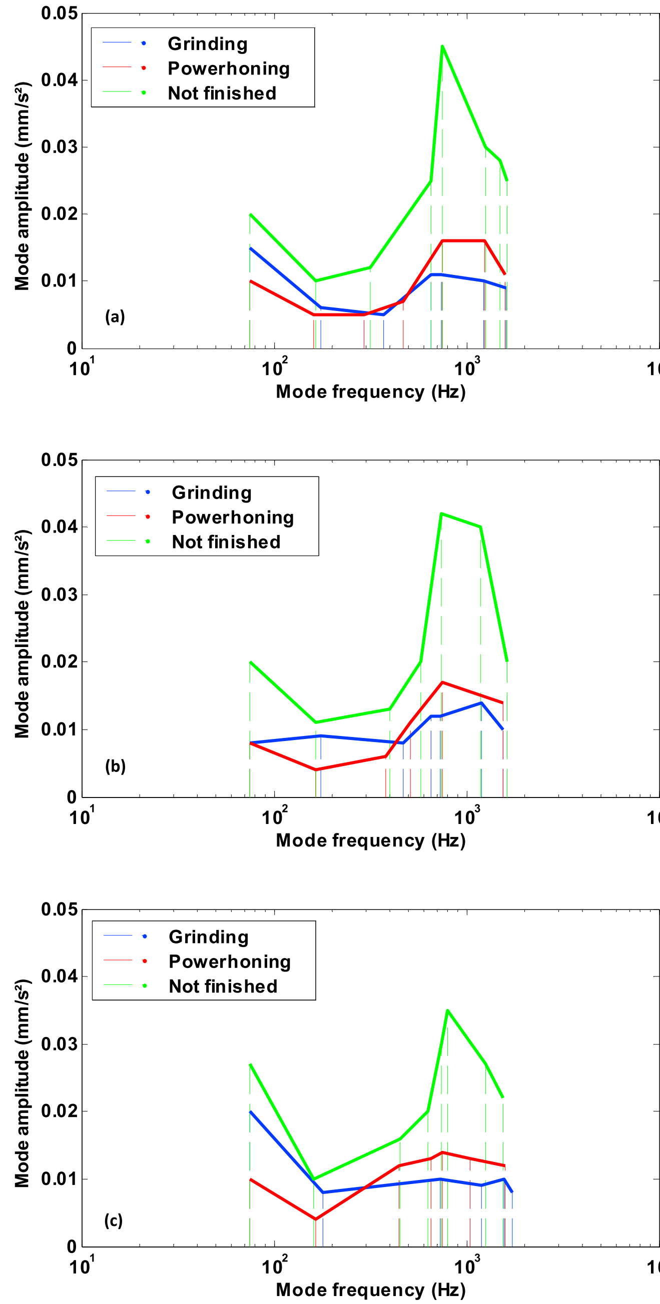

Fig. 14. Frequencies of the friction noise modes and its associated amplitude for measurements at 20 mm s^{-1}, for each considered test configuration: (a) after meshing using oil A; (b) after meshing using oil B; (c) after meshing in dry condition.

finished surfaces. It was applied to explore gear noise at the source and to study the influence of the tooth finishing process, the gear meshing and lubrication conditions on noise generation. Results demonstrate particularly that the finishing process of tooth surfaces has more impact on friction noise generation than the lubricant viscosity and roughness attenuation due to previous meshing. Furthermore, the relationship implemented between friction speed, surface scale and the main modes frequency demonstrate that is possible to replace actual gear vibration tests which are quite costly and time-consuming if done at industrial level by a simpler frictional test at lower scale.

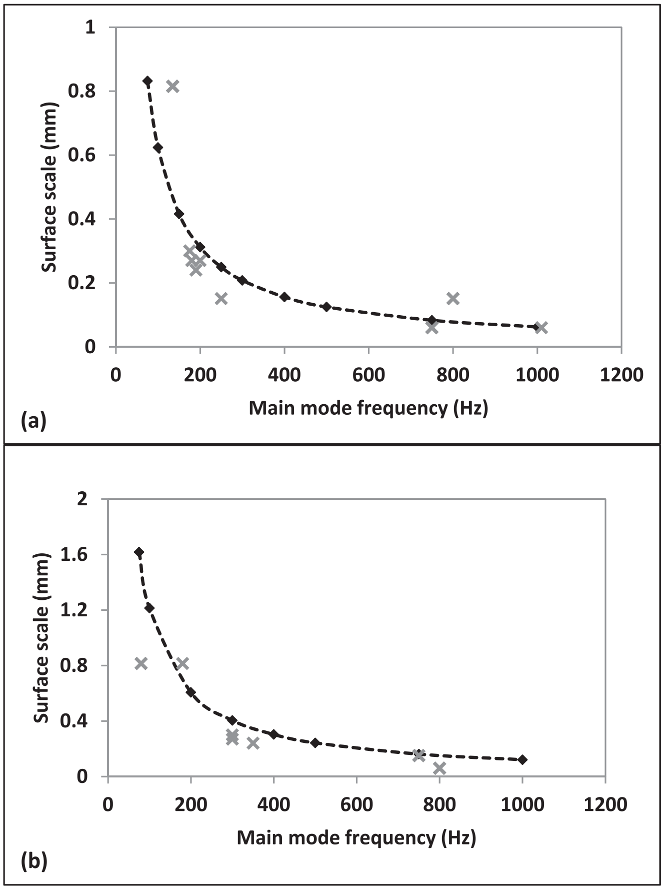

Fig. 15. Relationship between roughness characteristic scale and the main vibration mode for the friction noise measurements at (a) 10 mm s^{-1} and (b) 20 mm s^{-1}. The ' × ' represent the points from the measures and the dotted line is obtained using Eq. (2).

Acknowledgements

Funding and samples for this research were provided by Renault SAS.

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