END-OF-LINE BEARING QUALITY MONITORING ON COMPACT VENTILATORS: A CASE STUDY

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SUMMARY

The present paper deals with bearing quality detection on operating fans and proposes a methodology in order to assess the fan bearing quality on end-of-line test rigs. The method is based on amplitude demodulation. Latest order tracking techniques are used in order to compensate rotation speed fluctuations and thereby enhance the result quality.

INTRODUCTION

Decentralized domestic ventilation is a market in great expansion with specific requirements. Most applications consist of one or two fans and a heat exchanger. Like many other markets, a maximum cooling/heating performance (boost mode) and therefore airflow is expected. Unlike many other markets, the critical acoustic requirements are not evaluated on the boost mode, since it is rarely operated and the customer is rather expecting acoustic feedback at full speed. Customer requirements are related to low rotation speed (sleep mode), so that the application can be mounted in bedrooms without disturbing occupants' sleep. At low rotation speed, structure borne noise dominates fan noise, and any mechanical defect can lead to great noise excess. Among others, bearing quality plays a major role.

The present case study deals with a compact blower with nominal rotation speed 4000 rpm. As can be seen on the figure 1, blowers with a damaged bearing can be much louder than healthy fans (in some case up to 10 dB(A). Bearing noise is mostly located at high frequencies, but it can exhibit as a rattle like noise too, with both properties being subjectively very annoying. Thus, controlling bearing quality both during and at the end of the fabrication process is a very important matter.

The present study is organized as follows. The first part is dedicated to the presentation of end-of-line measuring constraints, and the main technical solutions are presented. In the second part, practical amplitude demodulation methods are introduced. In a third part, order tracking methods allowing rotation speed fluctuation correction are described. All assertions are illustrated with a test case measurement in the fourth part.

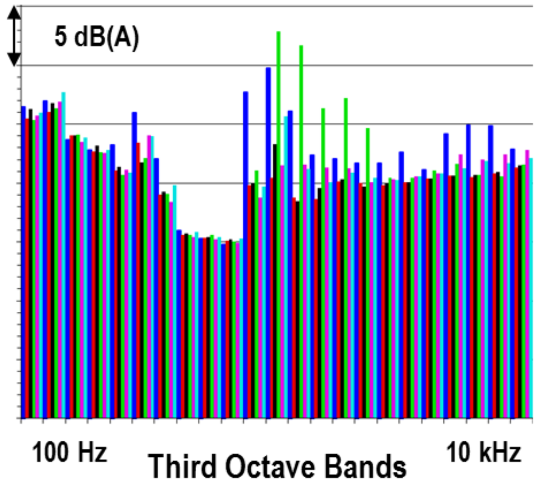


Figure 1: Sound Power Level from selected samples (one sample per color). The blue and green samples exhibit bearing damages.

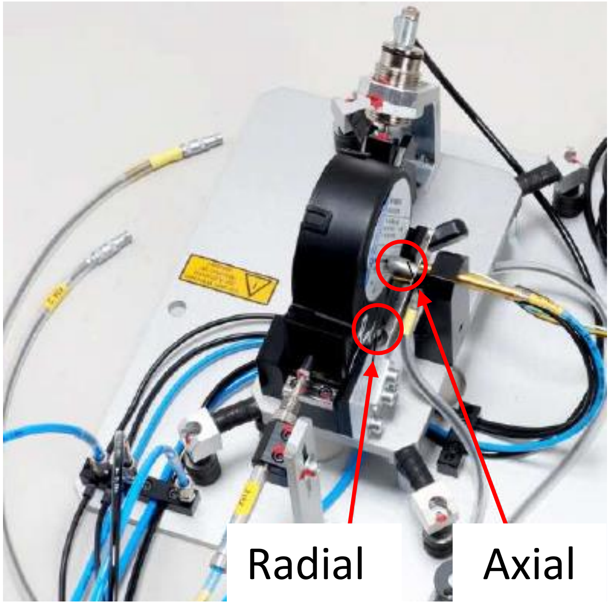
END-OF-LINE TESTING: REQUIREMENTS

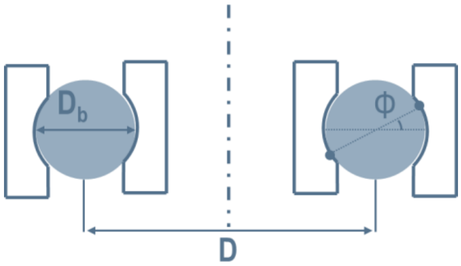
End-of-line testing has to meet demanding requirements. First of all, factories are mostly very noisy environments, which most of the time forbids using microphone techniques. Accelerometers, and more seldom laservibrometers, are greatly preferred. From a general point of view, a correlation between sound power (integral result) and vibro-acoustic (pointwise) measurements must be investigated in order to assess the end-of-line test quality.

Then, the test fixture must allow disturbance free measurements. On one hand, background vibrations shall be damped out, which is achieved by means of elastomer damping devices. On the other hand, the fan vibrations shall not be damped out by the test fixture itself. Special care must therefore be brought to the holding system. Accelerometers cannot be stuck nor glued as in laboratory conditions. Spring plates are a convenient solution, but can lead to vibration damping over 10 dB. When possible, accelerometers can be stuck directly on the test fixture. Alternatively, laservibrometers can be used. The test fixture used for the present project is depicted on figure 2. The fan is mounted on a plate which can freely vibrate on springs. Two accelerometers are mounted on plate springs. The first one is mounted radially below the fan and shall help investigating the fan imbalance. After investigations, it has been found that the best position and direction to investigate bearing quality is axial, in the shaft neighborhood.

Last but not least, the test has to be carried out in a very short time, since short production cycle times are a strong economical constraint. In the present case, only 5 seconds are allowed, whereas measurements in laboratory are usually carried out with 30 seconds.

End-of-line testing is the final step of manufacturing related quality management. As a consequence, the test rig's capability (reproducibility and repeatability) has to be assessed. Whereas statistical approach can be used for geometric measurements [1,2], this approach is not fully relevant for vibro-acoustic based end-of-line testing. Beyond statistical convergence problems (see next section), the test samples physical properties are changing with repeated measurements: the motor can warm up, grease can spread in the mechanical parts, etc. In the case of bearing testing, two major phenomena are to be mentioned: the grease in young bearings is not always very uniformly widespread in the rings. More important, bearing noise occurring from any fabrication related Brinell's mark will disappear during fan operation. This does not mean that the fan is healed: a damaged bearing is and stays a bearing with reduced life time. In the present case, a fan has been measured over 600 times on the test rig. Despite strong damage detection in the early measurements, no significant damage could be detected in the latter measurements.





$$BPFO = \frac{n \cdot f_r}{2}\left(1 - \frac{D_b}{D}\cos\phi\right)$$

$$BSF = \frac{D}{2D_b}\left(1 - \left(\frac{D_b}{D}\cos\phi\right)^2\right)$$

$$BPFI = \frac{n \cdot f_r}{2}\left(1 + \frac{D_b}{D}\cos\phi\right)$$

Figure 2: Overview of the fan test fixture and bearing overroll frequencies. BPFO, BSF and BPFI stand respectively for ball passing frequency outer race, ball spin frequency and ball passing frequency outer race.

BEARING MONITORING

The most widespread technique in vibro-acoustic end-of-line testing is third octave analysis. This kind of analysis is quite robust to external disturbances. Since bearing noise is mostly visible at high frequencies (in the kHz range), the related bands are becoming very large. Thus no distinction between damage on the inner race, ball or outer race is possible. Besides, the damage related vibrations can be quite small and masked by other mechanical phenomena.

In order to solve these problems, a different technique has been developed and widely accepted in the community. The basic idea is quite simple: as the shaft rotates, the bearing balls roll over the inner ring and outer ring at different relative speeds. Besides, the balls rotate over themselves with a third speed. Each time a ball rolls over a damaged location, a shockwise excitation of the fan is produced, which results in a modulation of the fan vibration amplitude over a broad frequency range. Depending on its location, the damage will therefore be overrolled by the bearing balls at different speeds, resulting in amplitude modulation of the fan vibrations at different frequencies. An overview of the formula predicting these frequencies is given on Figure 2. Parameters are hereby the number of balls n, the ball diameter D_b, the pitch diameter D, the shaft speed f_r, and the contact angle Φ. Conducting amplitude demodulation of the vibration signal in a selected frequency range will allow identifying modulation related to the aforementioned overroll frequencies and thus the damaged bearing part. This technique is called envelope analysis.

Envelope analysis can be carried out using multiple alternative procedures, two of which are covered here. The first step is common to both procedures and consists of selecting a frequency range to be demodulated. There is no standard way to choose this frequency range. Depending on the number of harmonics to be demodulated, it should be defined wide enough. Recent works from Randall & Antoni [3] dealing with spectral kurtosis aimed to automatically investigate the time signal and identify the optimal frequency range. The idea is thereby to dyadically divide the frequency range into subsignals, by appropriately filtering the signal, and then computing the kurtosis of each subsignal. As bearing noise is characterized with shockwise excitation, a high kurtosis in a subsignal is often a hint a bearing is damaged. This method has been applied on the present case without convincing results. The frequency range has thus been manually investigated and defined as an 1800 Hz center frequency and a 1600 Hz bandwidth.

The two procedures are summarized on the Figure 3. The procedure (A) is exclusively based on filtering operations and thus working with real valued signals, whereas procedure (B) implies working with complex valued signals as a result of the Zoom step. As the latter is the standard Brüel & Kjær ® procedure [4], this procedure has been chosen to be implemented on the test rig. In order to better identify hidden modulating phenomena, Randall [5] suggests using the squared envelope before low-pass filtering and correcting the resulting signal amplitude. This has been tried out during the present study, but results (not shown here) were not convincing. It indeed enhanced the modulation peak at the rotation frequency, but did not bring any significant enhancement at the bearing frequencies. It rather had a negative impact on the results when the order tracking (s. next section) has been applied. Therefore, this option has been discarded during the rest of the study.

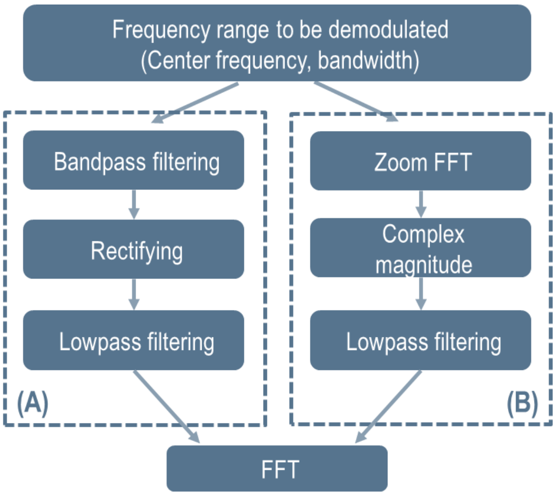


Figure 3: Procedure to compute a signal's envelope spectrum

A compromise must be made during the design of the filter. On one hand, high orders and thus high filter quality prevents unwanted information to disturb the envelope signal. On the other hand, high order filters generate longer transient effects which are to be cut from the resulting signal. In worst cases, this may become a bottleneck and affect the production cycle time.

Once the signal has been demodulated (following either method A or B), the envelope narrow band spectrum has to be computed. The conventional way to compute narrow band spectra from signals is based on subsampling, windowing and averaging. Doing so, the stationary (in the statistical sense) spectrum is evaluated. Though, bearing noise can be a very sporadic phenomenon, as it highly depends on the bearing ball movement. Damages located on the bearing outer or inner race are very likely to be excited by every passing ball, but ball located damage may not be excited so regularly. An intern study (not detailed here) carried out on a blower with ball damage showed at nominal rotation speed that the damage could only be detected every 7th second. A solution to this problem is to increase the test rotation speed, as far as the tested product allows it. As mentioned earlier, this implies enhancing the demodulated bandwidth and may be counterproductive. Another solution is to replace the average of the sample spectra with their maximum value. Doing so, the resulting spectrum will be quite noisy, but any overrolled damage will be detected. This trick is widely used in laboratories, where 30 seconds of a signal allow a considerable amount of samples. But this is a major reason why such a method cannot be evaluated as capable according to the aforementioned norms [1,2]. At this point, a thorough -and in most cases intense- discussion must be carried out with the customer.

ORDER TRACKING

In order to be economically competitive, the blower driving electronics could not be too costly. As a consequence, the blower rotation speed is not constant over the operating time. For most customers, this is an appreciable compromise, since most of them do develop a master driving electronic that compensates for rotation speed fluctuations. But this strongly influences the envelope spectrum. Indeed, an envelope spectrum from a speed fluctuations afflicted signal will exhibit lower, smeared peaks (s. Fig. 6(a)), whereas these peaks will be way higher and sharper without speed fluctuations (s. Fig. 6(b)). Fluctuations of ± 1 % of the nominal rotation speed were enough to significantly affect the averaged envelope spectrum beyond the 4^{th} order. The effects are even stronger if the maximum value of the spectrum is computed (s. Fig. 7).

The same problem is met in the field of gear monitoring, as any motor rotation speed fluctuation prevents inspecting high shaft orders. Compensation techniques have been developed over the past years, based on the following idea: if the rotation speed is known, its integration will deliver the relationship between the shaft angle and time. By mapping the vibration signal from time to the angle domain, any speed fluctuation will be corrected. The resulting order spectrum is then clean from speed fluctuations effects. External sensors (optic/magnetic based sensors, electronic driving output) can be used. This information can be sampled at high rate (see for instance Head Acoustics®) and detailed information is gathered.

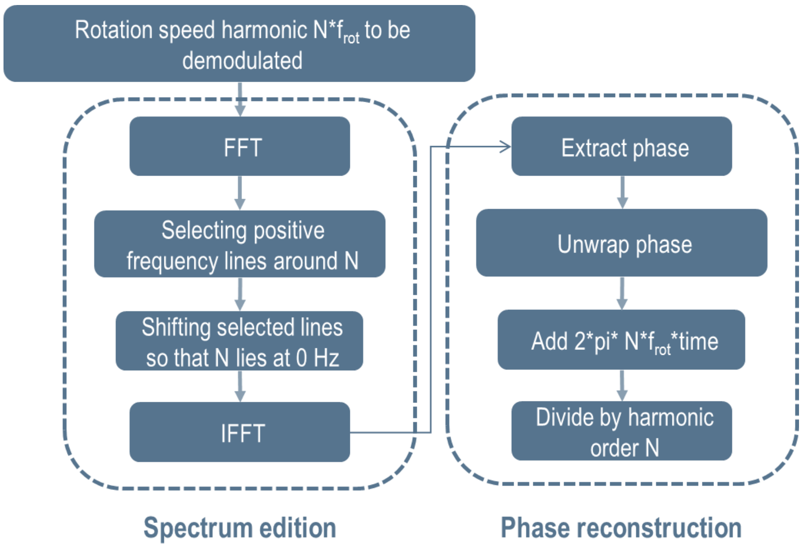


Figure 4: Procedure to extract the rotating phase deviation information.

Recent work from Coats, Coats & Randall, Bonnardot & al. [6,7,8] delivered a very interesting alternative provided the rotation speed fluctuations are neither too high nor occur too sharply. In most cases, this latter point is not problematic since the fan motor has enough inertia. Their idea is to carry out frequency demodulation on rotation speed harmonics from a vibration signal. This way, the shaft angle information is directly extracted from the signal to be analyzed, and rotation speed fluctuations up to 25-30 % can be compensated (Coats & Randall [7]). The procedure to retrieve the rotation speed signal from a vibration signal is depicted in Figure 4. As mentioned by Coats [6], the choice of the bandwidth to be demodulated is a very important step. If this bandwidth is too short, a part of the information will be missing. Conversely, a too wide bandwidth is likely to contain additional information the demodulated signal may be polluted with (s. Fig. 5(b)). In the present case, the bandwidth was set to 10 % of the mean rotation speed in order to cope with potential production quality scatter.

TEST CASE RESULTS

The following figures illustrate results obtained with the axial sensor on a damaged sample. Measurements were carried out in a laboratory setting at nominal rotation speed 4000 rpm, which explains the 30 seconds time duration. The sampling frequency was set to 132 kHz. The bearing orders for outer race, ball spin and inner race overroll are respectively 2.11, 3.05 and 3.89. Amplitudes of the envelope spectrum are investigated at these three order frequencies and its respective first harmonics.

The blue line ('Tacho') on Figure 5(b) shows the variation of the rotation speed during the measurement, recorded with an external laser sensor, which is used as an independent measurement for comparison. The corresponding phase deviation from the laser sensor is depicted on Figure 5(a), also in blue. Results obtained through frequency demodulation of the rotation frequency are superimposed in brown over both figures. As can be observed, the phase deviation is very well caught with both methods. When computing back the instantaneous rotation speed, residual fluctuations are clearly visible on the data extracted from the vibration signal with a 10 % demodulation bandwidth. A deeper analysis of the vibration signal (not shown here) showed that these fluctuations are related to a 7 Hz amplitude modulation of the vibration signal at the rotation frequency. The origin of this modulation is still unclear. Further tries showed that these fluctuations disappear when the bandwidth to be demodulated is reduced below 4 % of the rotation speed (s. Fig. 5(b)), with the extracted result almost perfectly masking the tachometer.

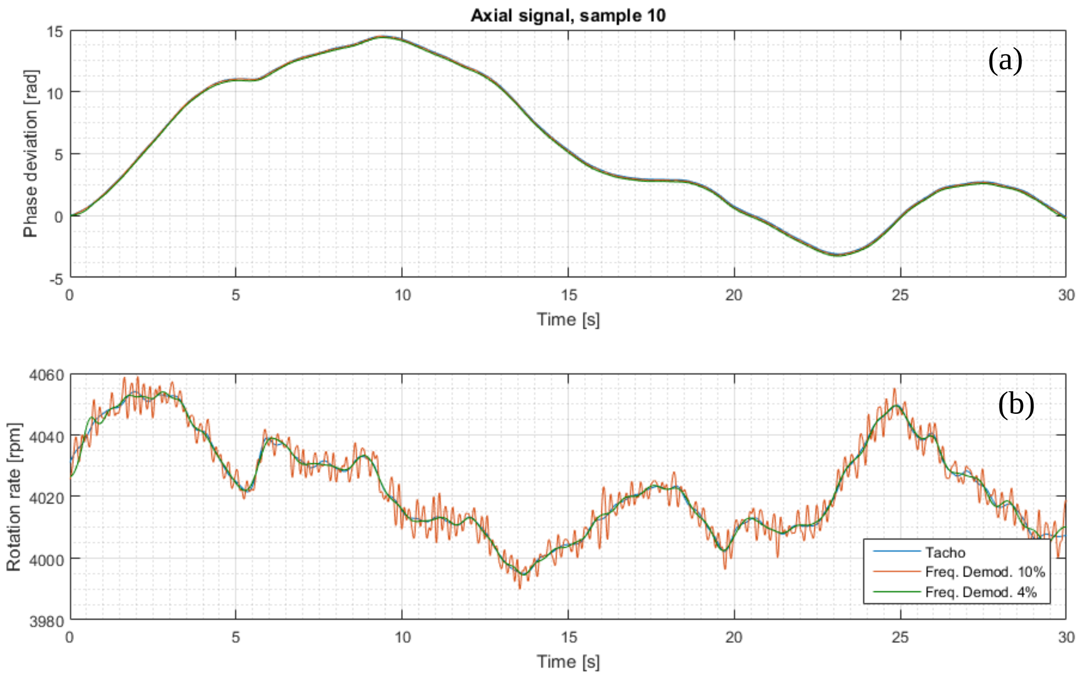


Figure 5: Reconstructed phase deviation (a) and rotation speed (b)

Figure 6 and Figure 7 shows the envelope spectra computed without and with rotation speed correction, respectively in the top and bottom parts. Figure 6 shows the averaged envelope spectrum, whereas Figure 7 shows its maximum counterpart. The bearing frequencies/orders are depicted in red lines.

When looking at the envelope spectra with no speed correction, as already mentioned, the rotation speed fluctuations clearly deteriorate the envelope spectrum at frequencies higher than the 4th order in Fig. 6(a) and 7(a). The deteriorations are greater in the maximum envelope spectrum (s. Fig. 7(a)), which was expected.

In the speed corrected envelope spectra (s. Figs 6(b) and 7(b)), a large amplitude is present at the first harmonic of the outer race ball passing frequency (Order 4.24), which is a sign that the blower bearing is damaged. It should be noted that this can also be observed, although with a lower amplitude, in the non-corrected envelope spectrum in Fig. 6(a) and 7(a), and the same conclusion of ball damage might be diagnosed in this situation, though with less confidence.

Additionally, through comparison of the rotation speed corrected and uncorrected results, it was identified that when automatic detection was applied to the uncorrected envelope spectrum in Fig. 6(a) and 7(a), this automatic detection misinterpreted the amplitude of the 3rd motor order as being the ball spin frequency. This error in automatic detection did not occur in the rotation speed corrected results, highlighting the necessity of speed correction to obtain correct automated results.

Finally, the rotation speed corrected envelope spectra are identical up to the 10th order, which shows that the residual fluctuations in the frequency demodulated envelope are negligible for this frequency range. It should be noted that in [6] Coats showed that frequency demodulation can be applied recursively on higher orders (so called 'multi-pass phase demodulation'), so that the same accuracy can be extended to higher harmonic orders as needed for an application. Employing this alternative variation is still under investigation.

Consecutive to the present study, envelope spectrum based bearing monitoring has been implemented on an EOL test rig. The test rig is equipped with an external rotations speed sensor, which is why phase demodulation based order tracking has not been implemented in this application. Nevertheless, the rotation frequency demodulation could have been implemented on the test rig, provided a given rotation speed fluctuation range is defined. Both for security reasons, and as an external rotation speed sensor is already mounted on the test rig, this option has not been implemented.

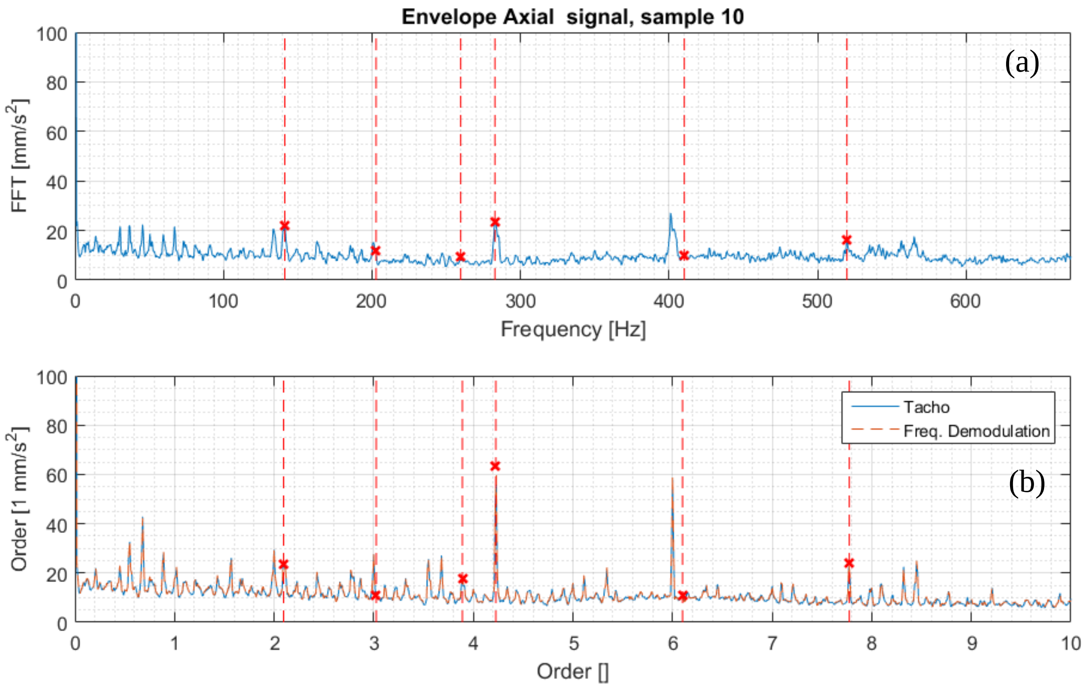


Figure 6: Averaged envelope spectrum without correction (a) and with correction on the basis of external tacho signal and frequency demodulation (b). Bearing frequencies and order are depicted in red.

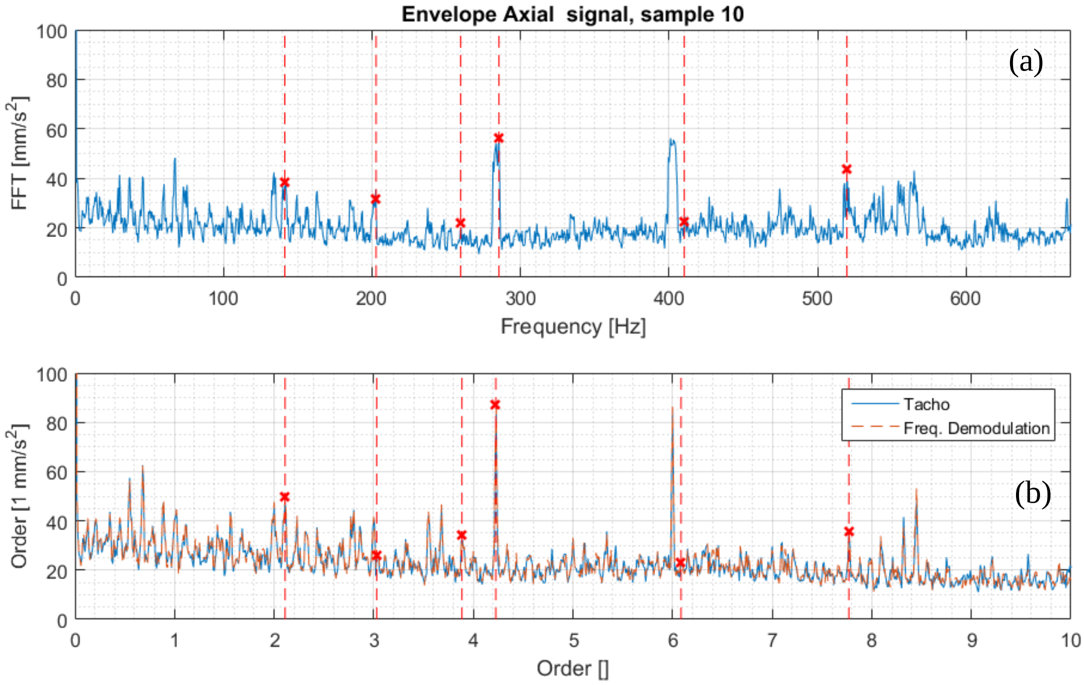


Figure 7: Max envelope spectrum without correction (a) and with correction on the basis of external tacho signal and frequency demodulation (b). Bearing frequencies and order are depicted in red.

CONCLUSION

End-of-line bearing monitoring is one key to avoiding excessive fan noise at low rotation speeds in manufactured products. This paper provided information dealing with the design of end-of-line test rigs and discussed a method to perform envelope analysis in the presence of moderate fan rotation speed fluctuations. It has been shown that even small fluctuations (a few percent of the nominal rotation speed) are sufficient to significantly deteriorate the envelope spectrum without compensation. Order tracking allows compensating for these fluctuations and has been successfully applied using frequency demodulation of the vibration signal itself, and confirmed with comparison to external laser sensor measurements.

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