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# Variable Speed Drive Application Based Acoustic Noise Reduction Strategy

C. Grabner

***Abstract* — The acoustic sound level caused as secondary effects of industrial energy conversion processes is often perceived as an unwelcome noise emitted into the enclosing surrounding. In order to verify design aspects for reducing such undesirable noise emission in particular within variable electric speed drive applications, suggested novel combinations of electrical and mechanical motor topologies are analyzed.**

***Index Terms* — Acoustic, Noise, Squirrel cage motor design, Inverter.**

## I. Introduction

The acoustic noise levels of industrial drives are usually restricted by international standards. This has led to a greater appreciation of the benefits of a quiet environment and a preference for using improved quieter products.

As we are commonly dealing with very complex multi-physical problems, extended investigations to the acoustic behavior of the complete speed variable induction drive system, as it is schematically depicted in Fig.1, have been performed in the lab to obtain reproducible results with high accuracy.

Main interest is thereby given to the influence of the complete mechanical induction motor design to the almost manifold acoustic response under usually converter-fed circumstances. To take remedial measures, totally-closed stator topologies and rotor skewing opportunities, have been deeply investigated in conjunction with converter supply.

Figure 1: Typical drive system consisting of motor and integrated power converter with basic control features.

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Christian Grabner is with the Austrian Institute of Science, Austria (e-mail: christian.grabner@at.atb-motors.com).

## II. Physical Fundamentals of Acoustic Noise

Sound is the mechanical vibration of a gaseous medium through which the energy is transferred away from the source by progressive sound waves. Whenever an object vibrates, a small portion of the energy involved is lost to the surrounding area as unwanted sound – the acoustic noise [1,2].

### A. *Sound power*

Any source of noise is commonly characterized by the emitted sound power, measured in Watt. This fundamental physical property of the source itself is often known as absolute parameter for the acoustic output. It is widely used for direct rating and comparison of different sound sources.

The sound power can be found conceptually by adding the product of the areas times the acoustic intensities for the areas on any hypothetical surface that contains the source. Since equipment for measuring of acoustic intensity is generally not available, sound power inevitably has to be determined indirectly by taking the spatial average of the sound pressure squared measured on equal areas on the surrounding surface.

### B. *Sound pressure level in decibel*

As the acoustic intensity, the power passing through a unit area in space, is proportional in the far field to the square of the sound pressure, a convenient scale for the acoustic measurement can be defined as sound pressure level

$$L_p(t) = 10\log\left(\frac{p(t)}{p_0}\right)^2 = 20\log\left(\frac{p(t)}{p_0}\right) \tag{1}$$

in decibel, with $p_0$ as the reference sound pressure of 20 µPa. The measured sound pressure $p(t)$ in (1) depends on many insecure factors, such as e.g. orientation and distance of receiver, temperature and velocity gradients inside the involved medium.

The analysis of (1) in the frequency domain is fortunately done with the discrete fast Fourier-analysis. Therefore, the time-periodical signal (1) is sampled as $L_{p,n}$ and further processed at a distinct number of N samples as

$$\hat{L}_{p,\upsilon} = \sum_{n=0}^{N-1} L_{p,n} e^{-j(2\pi n/N)\upsilon} \quad , \ \upsilon = 0,1,2\ldots N-1 \tag{2}$$

in order to obtain the Fourier coefficients $\hat{L}_{p,u}$ of the interested frequency-domain.

## III. Sound Pressure Level and Human Hearing

The treatment of acoustic noise and its effects is a complicated problem, which must take a wide variety of parameters into account to achieve good correlations between measurement and the human reaction.

### *C. Human hearing mechanism*

The quietest sound at 1000 Hz which can be heard by the average person is found in Fig.2 to be about 20 µPa and this value has been standardized as the nominal hearing threshold for the purpose of sound level measuring. At the other end of the scale the threshold of pain occurs at a sound pressure of approximately 100 Pa.

An inspection of Fig.2 shows that a pure tone having a sound pressure level of e.g. 20 dB and a frequency of 1000 Hz is plainly audible, whereas one having the same sound pressure level but a frequency of 100 Hz is well below the threshold of audibility and cannot be heard at all.

### *D. Loudness level in phon*

The loudness of a pure tone of constant sound pressure level, perhaps the simplest acoustic signal of all, varies with its frequency, even though the sound pressure may be the same in every case.

Although our hearing mechanism is not well-adapted for making quantity measurements of the relative loudness of different sounds, there is a fair agreement between observers, when two pure tons of different frequencies appear to be equally load. It is therefore possible to establish in Fig.2 contour plots of equal loudness in phone. These subjective felled loudness curves are obtained by alternately sounding a reference tone of 1000 Hz and a second tone of some other frequency. The intensity level of the second tone is then adjusted to the value that makes the two tones appear equally load.

A pure ton having a frequency of 100 Hz and a sound pressure level of about 35 dB sounds as load as a pure 1000 Hz tone whose intensity level is 20 dB, and hence the loudness level of the 100 Hz tone is by definition 20 phon. The 1000 Hz frequency is thus the reference for all loudness measurements, and all contours of equal loudness expressed in phon have the same numerical value as the sound pressure level in dB at 1000 Hz.

### *E. Weighing the sound pressure level*

Due to the human ears assessment of loudness, the defined purely physical sound pressure terminus has to be modified by an implicit weighting process in a way which corresponds to the more complex response of the human being. Therefore, among several possibilities, distinguished by B, C, D, E or SI-weighting in Fig.3, the A-weighted level has be found to be the most suitable for modifying the frequency response to follow approximately the equal loudness of 20 phon in Fig.2.

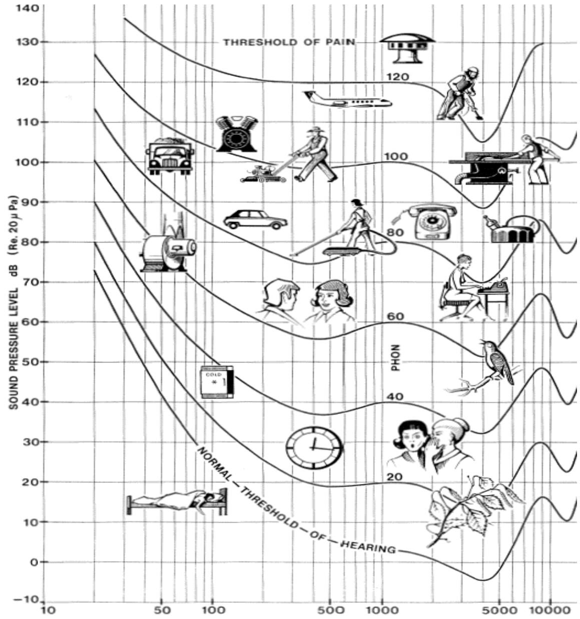


Figure 2: Typical sound pressure levels in dB and equal loudness levels in phon for different noise sources.

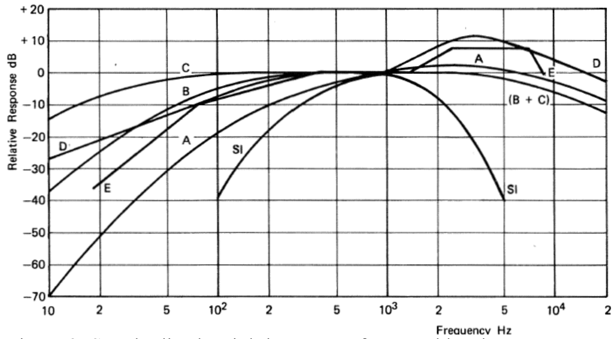


Figure 3: Standardized weighting curves for sound level measurement.

## IV. Acoustic Noise Measurement

Laboratory methods for testing noise specifications are usually very exhaustive, because the equipment and measurement procedures are subjected to international standards in order to guarantee the uniformity and quality of the measured results [3,4].

### *F. Semi-anechoic chamber*

The most accurate values of sound power levels are naturally determined in free field environment conditions. This is not practicable if heavily test specimens such as electrical drives are used. Thus, they are usually placed at a hard reflecting surface with absorption coefficients less than 0.06 in so called semi-anechoic chambers. A typical lab equipped with absorbing walls and several microphones is shown in Fig.4.

According to DIN EN ISO 3744 a fictive rectangular parallelepiped, which is schematically depicted in Fig.5 is applied for defining the exact spatial position of the used nine microphones. Thereby, the total surface of the test box is calculated as

$$S = 4(ab + bc + ca)\ , \quad a = 0.5\,l_1 + d\ ,\ b = 0.5\,l_2 + d\ ,\ c = l_3 + d \qquad (3)$$

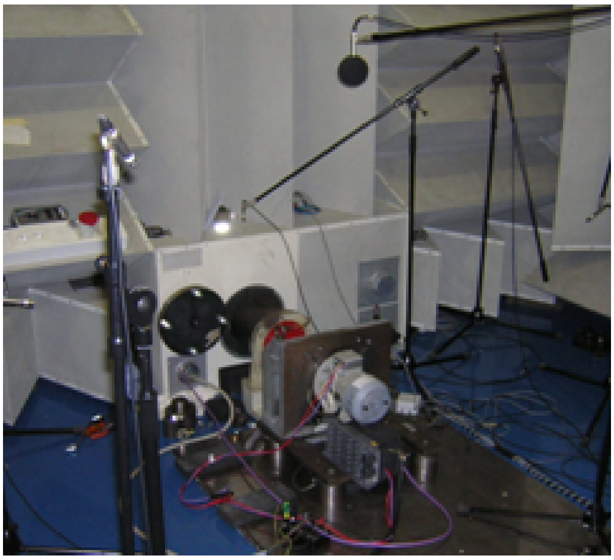

Figure 4: Schematically lab set-up for the acoustic noise measurement according to DIN EN ISO 3744.

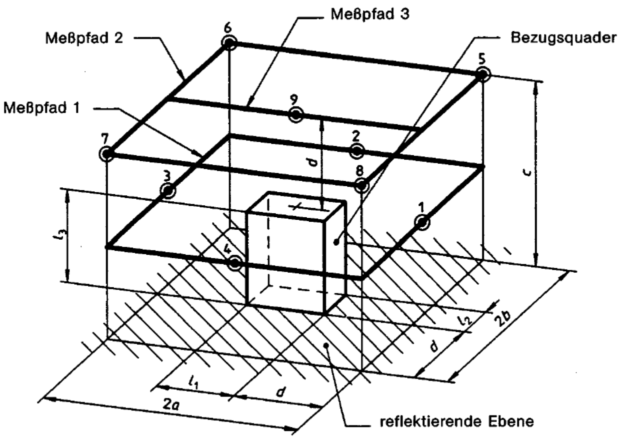


Figure 5: Space dependent microphone positions inside a semi-anechoic chamber.

The noise source to be measured is situated in Fig.5 in the centered area, which is characterized by the length $l_1$, the width $l_2$ and the height $l_3$. The other associated geometric distances in Fig.5 are thereby restricted by $l_1 \le d$, $l_2 \le d$ and $l_3 \le 2d$. The value for the distance d is about 1 m.

*G. Measured continuous A-weighted sound pressure level at each microphone*

With a wide variety of electrical circuits to condition, weight and integrate the signal path, the analysis section of the system shown in Fig.6 is usually the most complex.

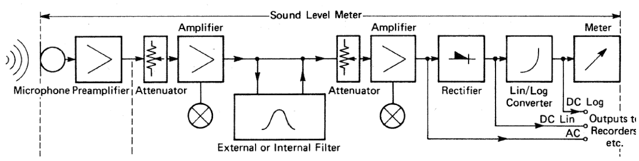


Figure 6: Typical block diagram of a noise measuring system.

The A-weighted sound pressure level

$$L_{pA}(t) = 10 \log \left( \frac{p_A(t)}{p_0} \right)^2 \tag{4}$$

expressed in dB(A), is directly accessible to measurement by incorporating an according filter, as shown in Fig.6. It correlates therefore extremely well with the subjective human response. The term $p_A(t)$ is denoted as the A-weighted instantaneous acoustic sound pressure $p(t)$. The analysis of (4) in the frequency domain takes advantage use of the fast Fourier transformation (2).

Treating time rather than frequency as the important variable, the A-weighted level (4) may be integrated over the measured time-period T as

$$\overline{L}_{pA} = \frac{1}{T} \int_T L_{pA}(t)\,dt \tag{5}$$

in order to give an A-weighted mean of the steady continuously time dependent $L_{pA}(t)$ signal.

*H. Equivalent A-weighted surface sound pressure level over all microphones*

The equivalent surface sound pressure level is obtained from the A-weighted and time-averaged signals (5) at a total number of N microphones as

$$\overline{\overline{L}}_{pA} \approx \frac{1}{N} \sum_{i=1}^{N} \overline{L}_{pAi} - K_{1A} - K_{2A} \tag{6}$$

The first correction coefficient $K_{1A}$ in (6) takes implicit account of eventually disturbing A-weighted environmental noise. The second correction coefficient $K_{2A}$ in (6) considers the A-weighted reflection or absorption of sound with regard to the surface sound power level.

*I. Equivalent A-weighted surface sound power level*

The measured equivalent A-weighted surface sound power level could be directly obtained from (6) with the geometric relation (3) in accordance to the relation

$$\overline{\overline{L}}_{wA} = \overline{\overline{L}}_{pA} + 10 \log \left( \frac{S}{S_0} \right) \tag{7}$$

whereby $S_0$ denotes the reference are of 1 square meter. Thus the difference between the spatial averaged sound pressure level (6) and power level (7) is constant. With an area $S_0$ of 14.3 $m^2$ in case of the used set-up in Fig.5, the constant term in (7) is determined as 11 dB.

## V. Cyclical Mutation of Stator and Rotor Part Combination

The stator part of a squirrel cage induction motor is typically manufactured so far with semi-closed stator slots [5,6]. However, continual progress in the industrial automation process allows the utilization of a novel stator construction with totally-closed stator slots as shown in Fig.7. The squirrel cage rotor could alternatively be built up with a skewed or even non-skewed rotor as depicted in Fig.8. Four different motor designs, which are based on those stator and rotor components, have be consecutively tested with regard to their suitability for acoustic noise minimization.

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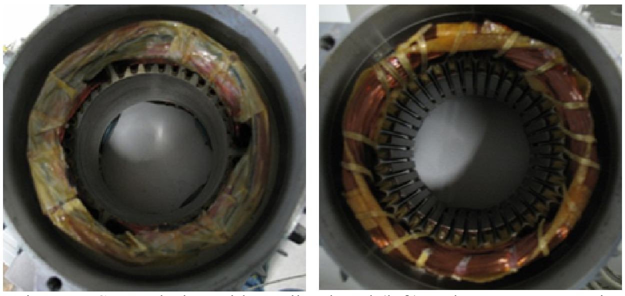

Figure 7: Stator design with totally-closed (left) and semi-closed (right) stator slots.

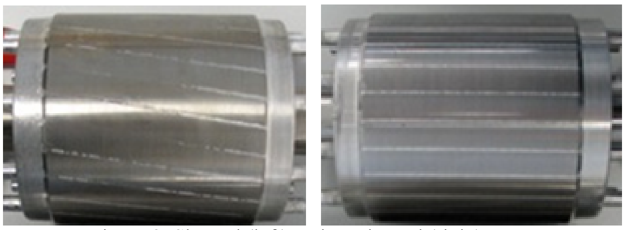

Figure 8: Skewed (left) and un-skewed (right) rotor.

TABLE I

| Combination | Stator slot design | Rotor design |
|---|---|---|
| Motor ① | Totally-closed | skewed |
| Motor ② | Totally-closed | un-skewed |
| Motor ③ | Semi-closed | un-skewed |
| Motor ④ | Semi-closed | skewed |

## VI. Measured Acoustic Noise Emission of Different Converter Driven Motors

Practical investigations of all possible motor combinations as considered in Table I have been performed systematically in the lab. In order to enforce a direct comparison between various test samples, the same motor fan as well as the same power converter technology has been used. The general adjustments of the utilized control unit are thereby also kept unchanged for all tests. Main interest is almost given to the influence of the standard 4 kHz modulation frequency to the acoustic noise emission.

The noise measurement has been carried out for each motor design over a wide speed range beginning from the lower value of 500 rpm up to rated-speed of 1390 rpm at constant mechanical rated-torque of 5 Nm. After reaching the physically quasi-steady operational state for each adjusted speed, the measured data set was stored. Purely dynamic acoustic noise in fact of a transient real-time run-up is herewith not considered. Within increasing higher speed ranges, the drive works in the field weakening range and so the load has been continuously reduced.

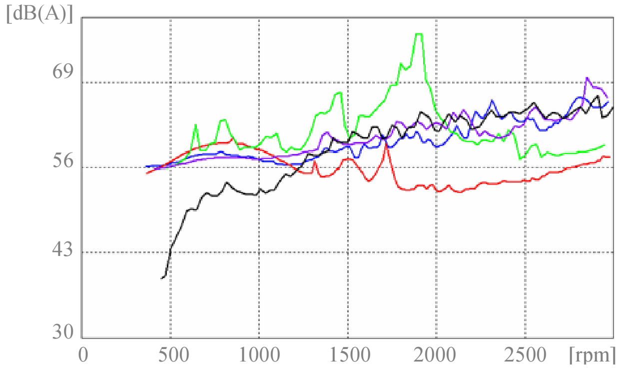


Figure 9: Measured sound pressure level versus speed for the inverter fed motor ① (blue), motor ② (red), motor ③ (green), motor ④(violet) and sinusoidal supply of motor ④ (black) at no-load.

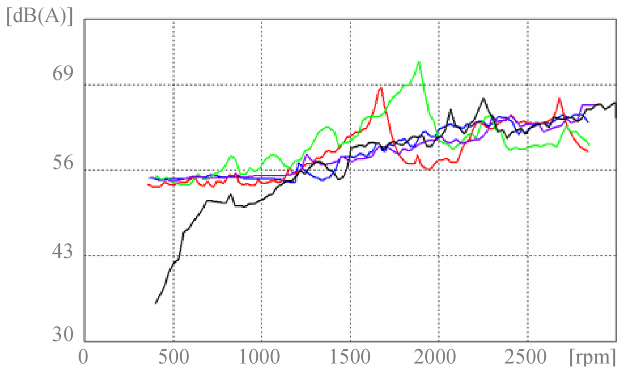


Figure 10: Measured sound pressure level versus speed for the inverter fed motor ① (blue), motor ② (red), motor ③ (green), motor ④(violet) and sinusoidal supply of motor ④ (black) at rated-load.

### *J. Evaluation of design impacts to the equivalent sound pressure level*

The comparatively performed measurement of the speed dependent equivalent continuous A-weighted surface sound pressure level (6) gives a good overview about the principal design influences. The focus is thereby almost given to occurring noise peaks.

It can be clearly seen from Fig.9 and Fig.10 that the courses concerning motor ① and ④ show very smooth tracks with small deviances over the complete speed range. Thus, it can be deduced, that the skewed rotor design implicitly prevents extensively noise peaks at certain speed values at no-load and rated-load operational state.

Contrarily, in case of both objects ② and ③ very discontinuous and even speed sensitive sound level behaviors with distinct peaks are obvious in Fig.9 and Fig.10.

Considering the test object ③, the maximal occurring peak of 77 dB in Fig.9 at 1900 rpm is slightly reduced at the same speed value to 72 dB in Fig.10. A completely contrarily observation could be made for motor ②.

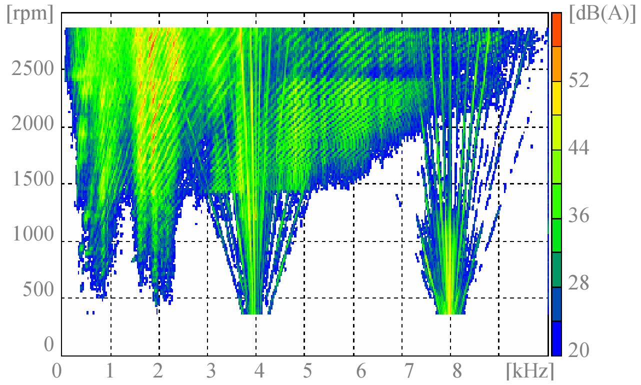


Figure 11: FFT of the sound pressure level of the inverter fed motor ① at rated-load depicted for speed values 0-3000 rpm.

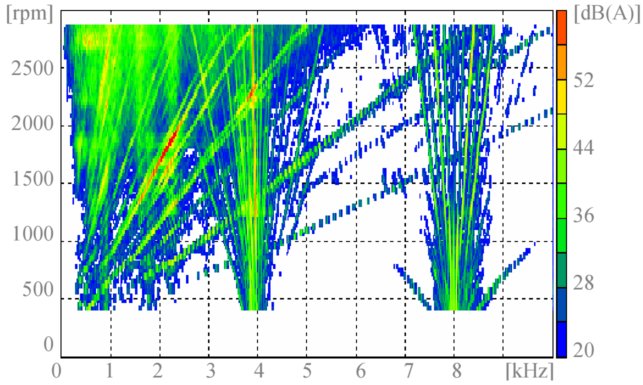


Figure 13: FFT of the sound pressure level of the inverter fed motor ③at rated-load depicted for speed values 0-3000 rpm.

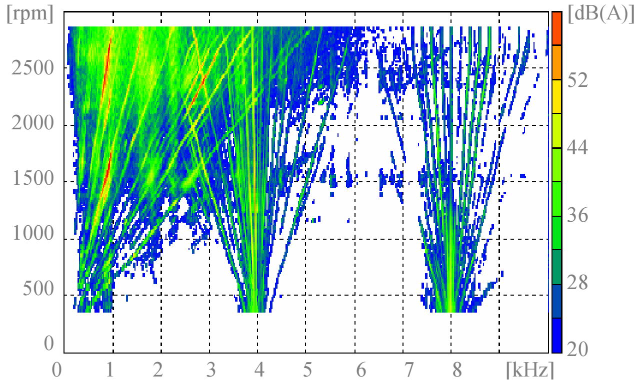


Figure 12: FFT of the sound pressure level of the inverter fed motor ② at rated-load depicted for speed values 0-3000 rpm.

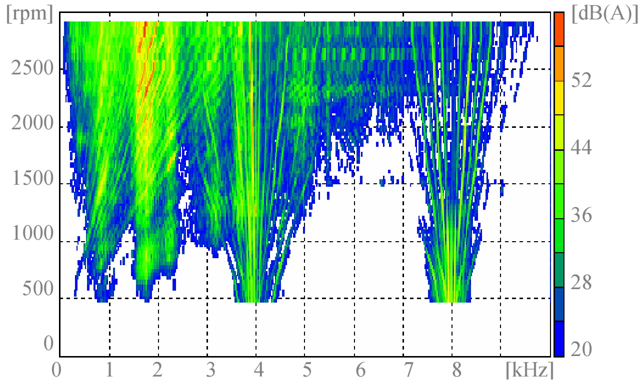


Figure 14: FFT of the sound pressure level of the inverter fed motor ④ at rated-load depicted for speed values 0-3000 rpm.

The noise level course at no-load is significantly shifted to much higher levels with arising additional peaks of 69 dB at the speed of 1650 rpm in case of the rated-load condition in Fig.10. The change from no-load to rated-load causes extended magnetic saturation and affects therefore in particular the totally-closed stator tooth tip regions. This fact has, in dependency on the motor design ② or ③ unexpected inverse impacts to the emitted noise level courses.

Principal differences between inverter-fed and sinusoidal fed motors are exemplarity shown in Fig.9 and Fig.10 for the candidate motor indicated by ④. In both cases, the generated noise level within lower speed range from 500 rpm up to 1200 rpm is much higher within the inverter driven motor ④. This shows impressively the effect of the fundamental 4 kHz modulation frequency within the converter driven motor.

*K. Sound pressure spectrum for each inverter driven motor design*

The description of the time-depended sound pressure in the frequency domain allows a deeper insight into the fundamental aspects of noise generation within the speed variable drive.

The harmonic sound pressure frequency components of the inverter-fed motors ① to ④ are depicted over speed ranges in Fig.11 to Fig.14, respectively. Independently of the considered motor design, it is obvious that especially at the lowest speed range, main attention has to be given to the default modulation frequency 4 kHz and the twice value, which are causing main important noise contribution in the spectrum.

The utilization of a skewed rotor type in motor ① and ④ obviously results in Fig.11 and Fig.14 between 1500 rpm and 3000 rpm in main spectral contributions at several frequencies situated in the bandwidth from 4 kHz to 8 kHz. Contrarily, both different designs ② and ③ prevent in Fig.12 and Fig.13 such significant spectral components between 4 kHz and 8 kHz, which directly leads to a completely different noise behavior.

Unfortunately, two distinct noise peaks of electromagnetic origin are visible in Fig.12 at the speed values of 1670 rpm and 2710 rpm with the same frequency of 940 Hz. Both components have already been verified for object ② in Fig.10. Moreover, from Fig.13, a very dedicated component due to stator slot harmonics of motor ③ at the speed of 1900 rpm could be identified at the frequency of 2300 Hz.


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## VII. Conclusion

The evaluation of distinct influences to the acoustic noise emission of inverter fed squirrel cage motors is fairly a very sensitive and complex research area. In particular the chosen modulation frequency of the power converter is identified as the main reason for the emitted noise of the drive at low speed, namely independent on the tested motor designs. The proposed novel construction with totally-closed stator slots and un-skewed rotor is very responsive to noise emission, but could be utilized within restricted speed ranges. However, it has been shown, that the usage of a skewed rotor always overcomes such problems. The proposed motor design with totally-closed stator slots and skewed rotor seems to be suitable for a slightly noise reduction at full-load and restricted speed values below the nominal even at converter supply.

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