Stresses of an Electric Motor Frame

Jiri Vondrich [1)]

[1)] Czech Technical University in Prague, Faculty of Electrical Engineering/ Department of Drives and Traction, Technicka 2, 166 27 Prague 6, Czech Republic, *vondrich@fel.cvut.cz*

Abstract — **The aim of the research was to apply modal analysis method to investigate the vibration processes of an electric motor in transient states. Transient states can occur, e.g. in the case of the DC motor driving by a voltage step or at sudden loss of the synchronous motor load. Mechanical impacts are transmitted through the frame of the motor and from the motor feet to the base plate.**

Keywords — DC motor, modal analysis, motor frame, mechanical impact.

I. INTRODUCTION

An electric motor is a piece of equipment, which produces noise and mechanical vibration. The causes of this noise and vibration are both electromagnetic and mechanical. In nature, the mechanical reasons are unbalanced rotors, bad couplings, and bad bearings and so on. The electromagnetic reasons are vibration by magnetization, the forces between the stator and rotor, especially in a motor with a high number of poles and torque pulsation. Motors supplied by power conversion equipment always produce torque pulsation. But there are many types of motors, which produce torque pulsation in spite of an ideal form of energy supply. These are universal motors, single-phase induction motors, switch reluctance motors, step motors, vibration motors, etc. The frequencies of the electric motor vibration cover a wide spectrum – from fractions of Hz to tens of kHz. Vibrations produce noise and mechanical stress in the structure of these motors. A reason for torque pulsation can be a non-symmetrical power net. There are special and really dangerous voltage shocks from the power net torque shocks produced by the driven equipment. An electrical motor can be destroyed by these vibrations, especially by torque shocks. There are reasons to study the mechanical quality of a motor, which is a complicated system of form, whose parts are comprised from different materials.

The concept of the modal analysis, introduced in [1], provides a natural foundation for the experimental modal analysis. When the data come from the measured response of the vibration structure (the frame of the electric motor), the relevant eigen values and eigenvectors of the state matrix may be interpreted as those of the underlying vibration system of the electrical motor frame. If the nature of the excitation is known but it is not measured, a mathematical description of the system in terms of natural frequency, damping ratio, and scaled mode shapes can be obtained. Furthermore, if the excitation is known or measured, the scaling of the mode shapes can be recovered to obtain a full mass-spring-dashpot model of the system from the multiple response data.

In contrast to the existing experimental modal analysis methods, such an approach can be used on the operational noisy input-output data recorded during the operation of the system under actual working conditions; the data need not to be collected under tightly controlled conditions.
As the majority of the popular modal testing and analysis methods are primarily based on the fast Fourier transformation (FFT), elucidating their limitations; more details of the methods may be found in the documentation of commercial Fourier analyses implementing these techniques. These limitations refer to FFT without curve fitting; good curve fitting of real data requires more computation than a data dependent system and yet provides less accurate results.

The 72 points were chosen on the surface of the motor frame; 56 of these were on the cylindrical part in 7 transversal planes with 8 points on the circumference, 8 were at the upper part of the frame and 8 were at the feet. The results of the measurements can be useful in designing the motor frame and the construction of its appropriate fixity. It was found to be appropriate to reinforce the vertical walls of the motor feet. After reinforcement, the experimental modal analysis was repeated.

The information obtained from the investigation of the structural responses of the electric motor will make it possible to apply the modal analysis to other electric machines.

II. EXPERIMENTAL SOLUTION

The induction motor (Firm MEZ 3~MOT. 4 AP 90L-4, No 404844-0418, IP 54, IM 1081, 1.5 kW, 1410 /min. 50 Hz, Δ/Y 230/440 V, ISOL.F, cos φ 0.82, 6.0/3.4 A, 98/11) with excitation in the net of the measuring points of the structure was investigated (Fig. 1).

The 72 points in total were chosen on the surface of the motor frame; 56 of those were on the cylindrical part in 7 transversal planes with 8 points on the circumference, at the upper part of the frame 8 points in 2 planes and 8 points at the feet (Fig. 2). Soft rubber in the place of the feet supported the motor.

The modal frequencies and mode shapes were determined by regression analysis from the estimated frequency response function. Measuring point number 2 on the circumference of the 1st transversal plane of the cylindrical part of the enclosure was chosen as the fixity reference point (Fig. 2).

The structure was excited at point 2 by an impact hammer (Firm PCB) with a plastic adapter and with a

Piezoelectric force transducer 3086 Bo 1 sn 3439, PCB Piezoelectoronics.

At the other measuring points, the responses were measured by a releasable miniature sized accelerometer 309 A, PCB Piezoelectoronics. Measuring was performed without moving the rotor. Mode shapes were determined from the computational animation of the frame motion.

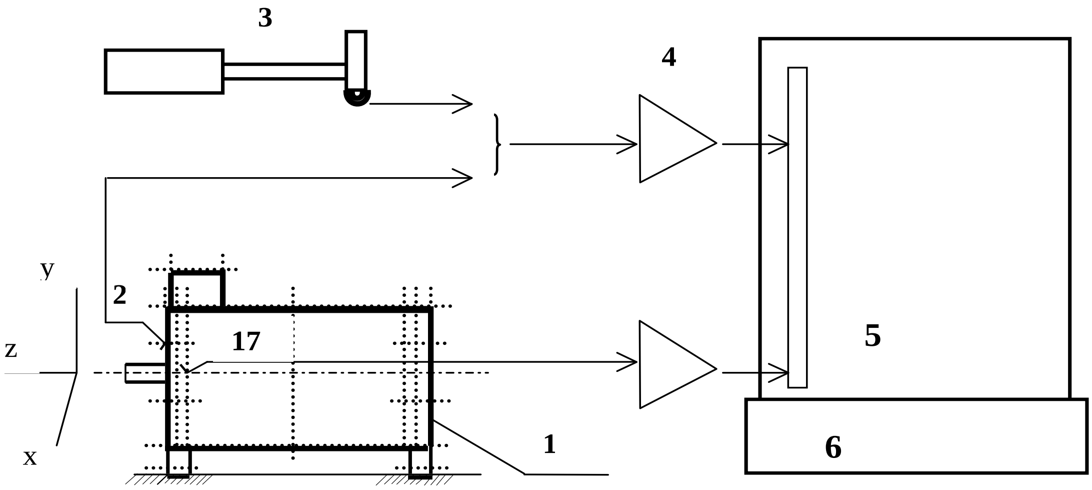

Fig. 1. Experimental modal analysis of a motor frame, (1 – induction motor, 2 – fixityed measuring point, 3 – impact hammer with piezoelectric force transducer, 4 – preamplifiers, 5 – A/D converter, 6 – PC, 17 – migratory points with miniature sized accelerometer)

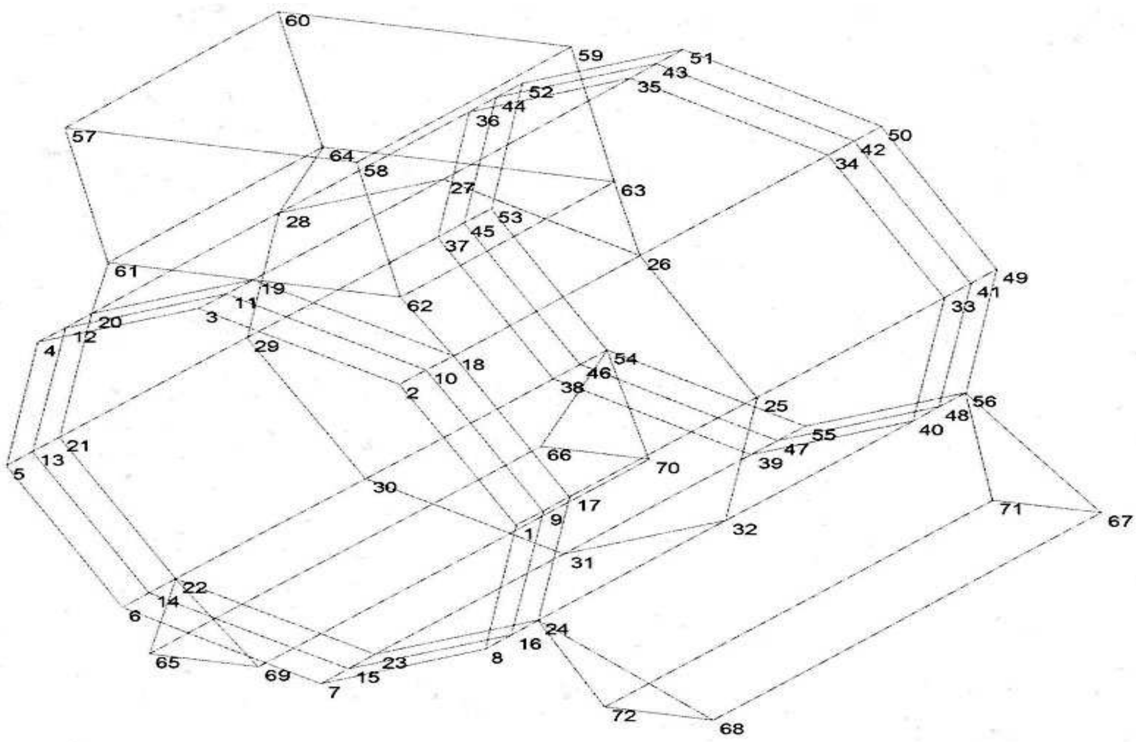

Fig. 2. The net of the 72 measuring points of a motor frame

At particular mode shapes of the motor frame, between measuring point 17 – direction x and impulse excitation point 2 – direction x, significant peaks of the transfer function of frequency (Fig. 3) occurred at:

1. Shape – 319 Hz (spatial oscillation),
2. Shape – 362 Hz (torsion oscillation),
3. Shape – 613 Hz (torsion oscillation),
4. Shape – 2010 Hz (blowing),
5. Shape – 3479 Hz (bending oscillation),
6. Shape – 4300 Hz (blowing).

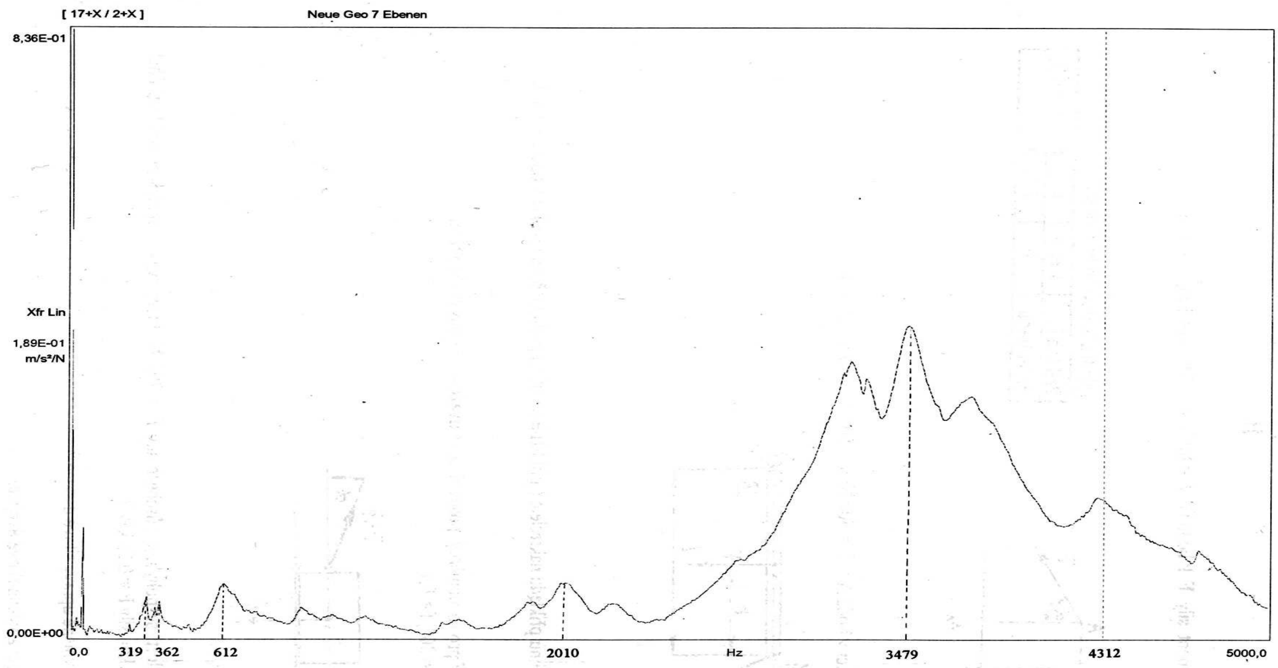

Fig. 3. The frequency transfer of the motor frame between points 17 and 2

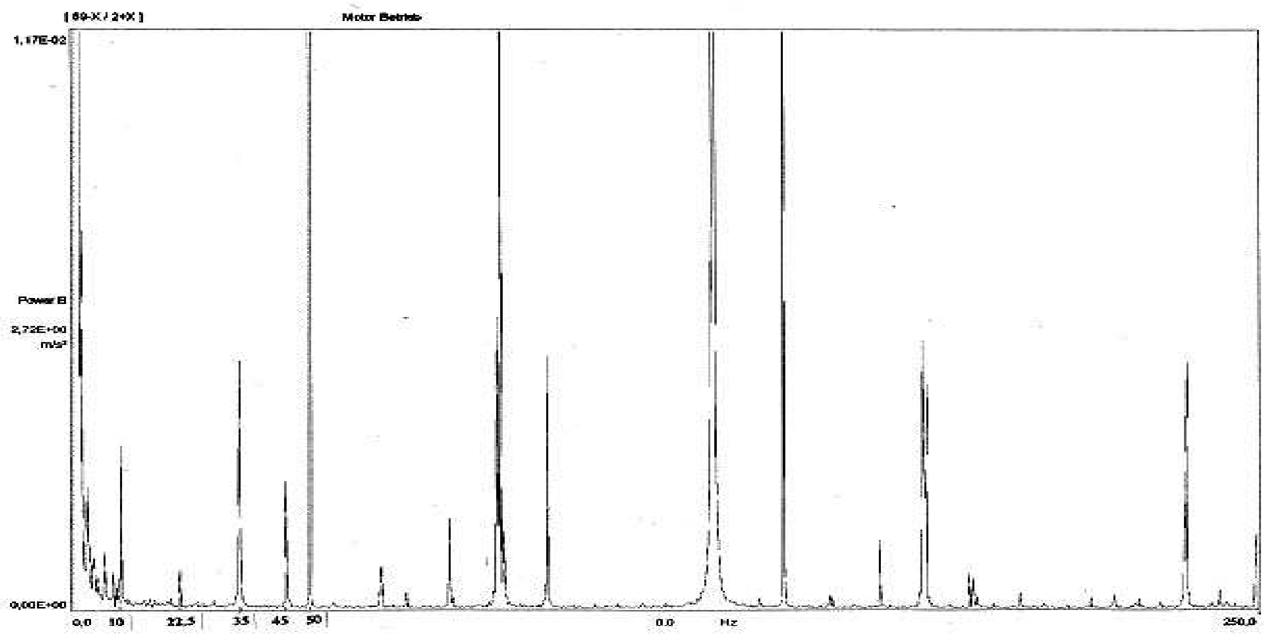

Fig. 4. The frequency transfer of the motor feet between points 69 and 2

It follows from the animation that the motor feet of the frame displacement have maximum amplitudes. In the frequency range below 50 Hz in the feet of the motor frame, between measuring point 69 - direction x and impulse excitation point 2 - direction x, significant peaks of the transfer function of frequencies 10, 23, 35 and 45 Hz (Fig. 4) occur. Therefore, new measurements of mode shapes in operation were performed.

Mode shapes were determined from the measurement points placed at the motor feet of the motor frame in revolutions that correspond to the above mentioned frequencies. The miniature piezoelectric accelerometer 309 A and the miniature accelerometer PCD 338 B SN 2288, 94 mV/g ICP (at point 2) were placed at 8 measuring points (65-72). The motor was fixed on feet with screws to a steel plate with a mass of 60 kg.

III. CONCLUSION

Any physical system can vibrate. The frequencies and the modal shapes which the vibrating system assumes are properties of the system. The frequencies and modal shapes can be determined analytically using modal analysis.

Analysis of vibration modes is a critical component of a design, but is often overlooked. Structural elements such as the frame of an induction motor can be particularly prone to perceptible vibration, thus disturbing sensitive equipment.

Inherent vibration modes in structural components or mechanical support systems can shorten motor life, and cause premature or completely unanticipated failure, often resulting in hazardous situations. Detailed fatigue analysis is often required to assess the potential for failure or damage resulting from the rapid stress cycles of the vibration.

Operational deflection shapes at particular revolutions corresponding to detected frequencies were determined from computational animation of the frame (Fig. 5) to be:

10 Hz – vibration of the feet around the overall centre of feet, up and down in opposition,
22 Hz – vibration around the vertical axis,
35 Hz – vibration around the horizontal axis,
45 Hz – vibration of the feet around the overall centre of feet (up and down in an opposite phase).

It follows from the animation that the motor feet have maximum amplitudes. The oscillation feet amplitudes have maximum values in the range of revolutions corresponding to the frequencies 10 Hz and 22 Hz. After sudden relief, the feet can fail at those frequencies.

The results can be useful for designing the motor frame and for the construction of the appropriate fixity. It is appropriate to reinforce the vertical walls of the motor feet. After reinforcement, the experiment by way of the modal analysis was repeated. The animation information obtained from the investigation of the electric motor structural responses will make it possible to apply modal testing to other electric machines such as transformers, generators, etc.

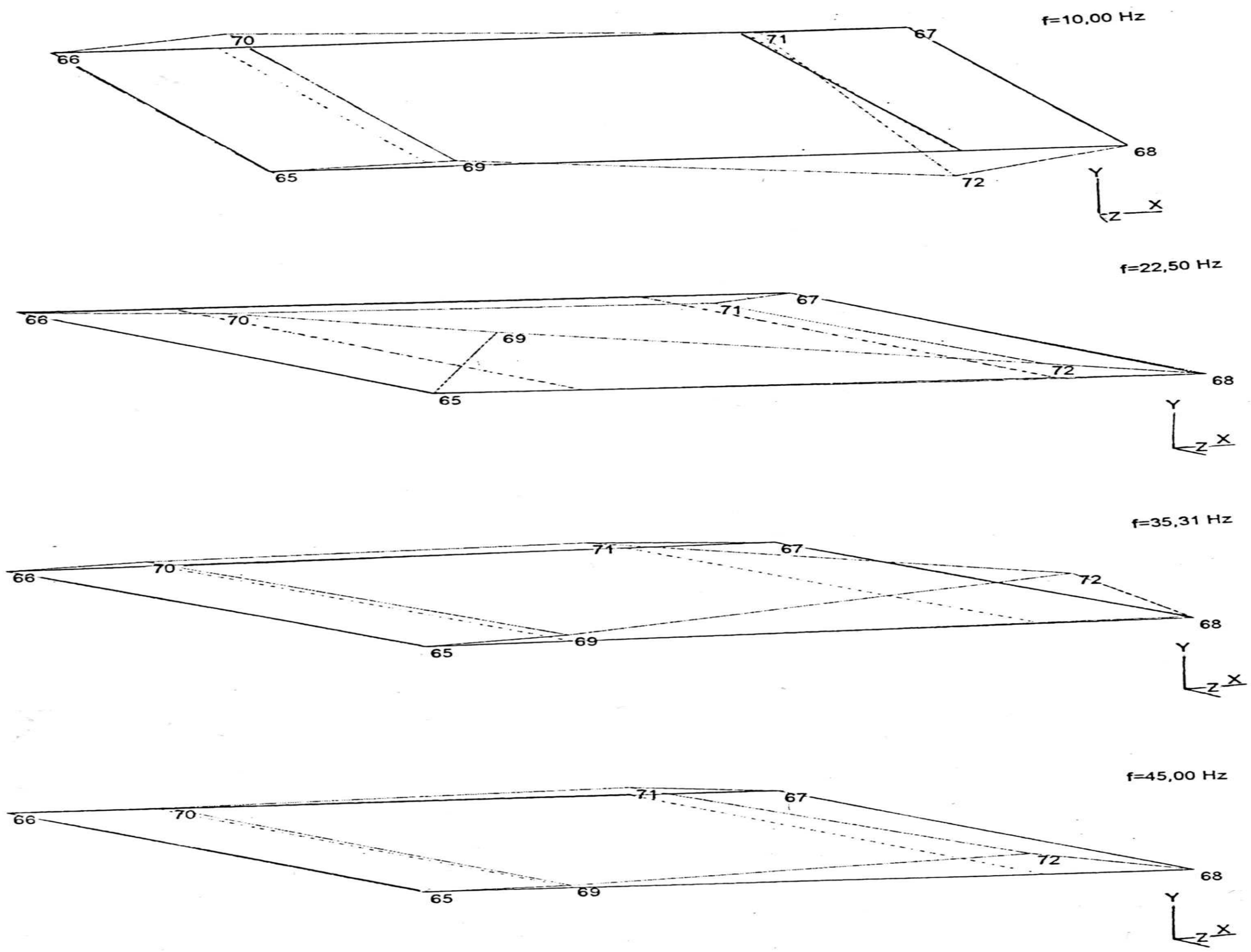

Fig. 5. The computer animation of the operating properties of the motor feet (at points 65 – 72)

REFERENCES

[1] D. J. Ewins, *Modal Testing: Theory and Practice.* Bruel-Kjaer, 1985.