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DYNAMIC PROPERTIES MODELING ANALYSIS OF THE RUBBER-METAL ELEMENTS FOR ELECTRIC DRIVE

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Abstract: The gradual increase in the volume of electric vehicles leads engineers to solve completely new problems of NVH (noise, vibration and harshness) One of the solutions is the use of rubber mounts to hold the electric motor of the vehicle's powertrain. This article deals with a rubber mount used to insulate vibrations from the propulsion system to the rest of the vehicle. Since the electric motor produces a different spectrum of vibrations and noise than the internal combustion engine (it produces lower amplitudes, but in a wider frequency band), it is necessary to adapt the design of the rubber mounts. This article deals with the possibility of using a mathematical model that can be used in a virtual vehicle model. It compares the differences in the vibrations of the internal combustion engine and the electric motor. From the results measured on the experimental model, the parameters for the FEM model and the mathematical model were identified. At the end of the article, the results from the experiment are compared with the results from the simulations.

KEYWORDS: Rubber mount. Virtual model. Electric powertrain.

1 Introduction

Advances in technical production led us to a constant increase in safety and ecology. The current trend in vehicles is to use an electric motor instead of the conventional combustion engine until recently, or to add an electric motor to the combustion engine. This significantly increases the number of possible vehicle powertrain concepts.

One of the problems we commonly encounter with vehicles is acoustic noise, mechanical vibrations, and their subjective feelings by people. Collectively, we call this problem NVH, and it is the electrification of vehicle powertrains that represents a major change in the field of NVH. New types of powertrains create new requirements for NVH, which is summarized in Fig. . [1]

2 Development of trends leading to electrification

In vehicles with a conventional internal combustion engine, the dominant noise of the internal combustion engine prevails. Other broadband noise sources, such as wind noise, tire noise and noise from transmission systems and various other ancillary equipment, are hidden under ICE noise. However, there is nothing to overheat them with electric vehicles. In addition, noise from electrically driven electronics, such as pulse width modulation (PWM) control, is coming to the fore. [2]

The absence of an internal combustion engine makes noise sources, which have not been a problem so far, more visible. The use of an electric motor opens up new areas that need to be included in the solution of NVH problems.

The most common sources of vibration and noise in electric vehicles are:

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- Electric motor(s),
- Direct-drive, single-speed gearbox,
- Power electronics unit,
- Battery pack cooling system.

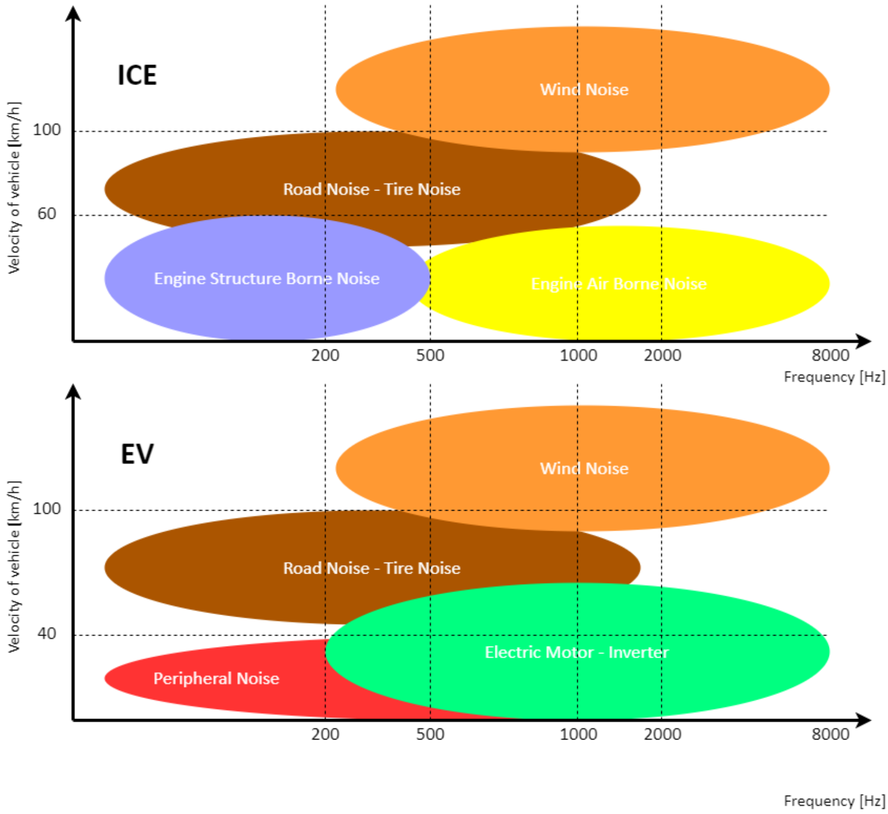

Fig. 1. ICE - EV different frequency ranges of interest

Fig. shows that in vehicles with ICE, the engine and transmission are a significant source of noise and vibration.

For vehicles with an electric motor, the biggest source of noise and vibration is tire noise due to the absence of an internal combustion engine and lower powertrain noise. Electrical propulsion is also a significant source of vibration and noise due to high frequencies. [3,4]

In principle, there are two ways of transmitting vibrations and noise to the vehicle cab from the electric motor. Either directly through the construction of the vehicle or transmitted through the air. Due to the low amplitude and high frequency, it is possible to use conventional rubber-metal mounts to reduce the transmission of noise and vibration. However, their design must also include use in a wide frequency operating range, up to 3000 Hz.

3 Effects of powertrain rubber mount on the NVH and the vehicle

The mounting of the drive system consists of a metal or plastic base on which a rubber insert is permanently vulcanized. Thanks to this connection, it is possible to dampen and eliminate oscillations from metal to rubber. The main functions of the rubber bearings of the drive train are [1]:

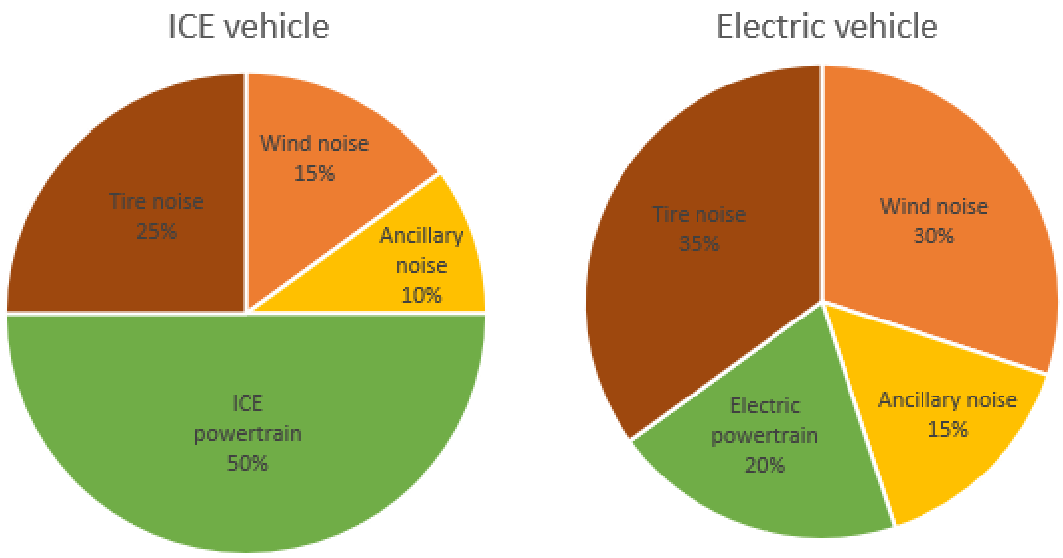

Fig. 2. Comparison of the size of NVH sources for ICE and electric vehicles

- Force transfer,
- Noise isolation,
- Vibration damping,
- Allow defined motion.

These functions create several of the conflicting requirements. These requirements are: [5]

- Force transfer:
  - hard bushing, minimal compression,
  - minimal damping.
- Ride Comfort:
  - the soft bushing in longitudinal direction,
  - the hard bushing in lateral direction,
  - low torsional stiffness.
- Noise Isolation:
  - soft bushing,
  - minimal damping.
- Vibration Damping:
  - maximum damping.

4 Measurements of dynamic properties of the rubber mounting of the drive unit

It is suitable to use an electromagnetic shaker to measure the dynamic properties of the rubber bearing at high frequencies, because hydraulic pulsators are frequency limited [5,6,8]. The experiment may be affected by resonances due to the wide range of frequency measurement by parts of the measuring device, such as the frame of the test device or the fixture of the test specimen (Fig.). To obtain the best possible result, it is necessary to have the frame of such a device designed out of resonance, and modal analyses must be performed on the fixture.

Piezoelectric acceleration sensors IEPE and piezoelectric force transducer with charge amplifiers were used for experimental measurements due to the need to measure at frequencies up to 2000 Hz.

When logging the experimental data, the dynamic stiffness and the loss angle depending on the frequency were captured (Fig. 5). The dynamic stiffness increases with increasing frequency, and the highest dynamic stiffness of the test body is around 1500 Hz, which is due to the resonance of this body.

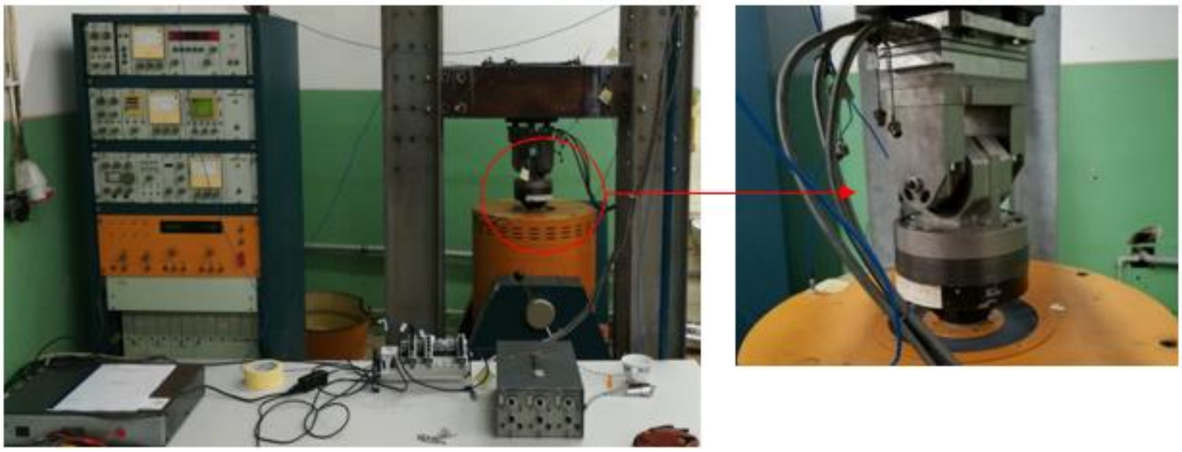

Fig. 3. Arrangement of the measuring test rig and clamping test specimen

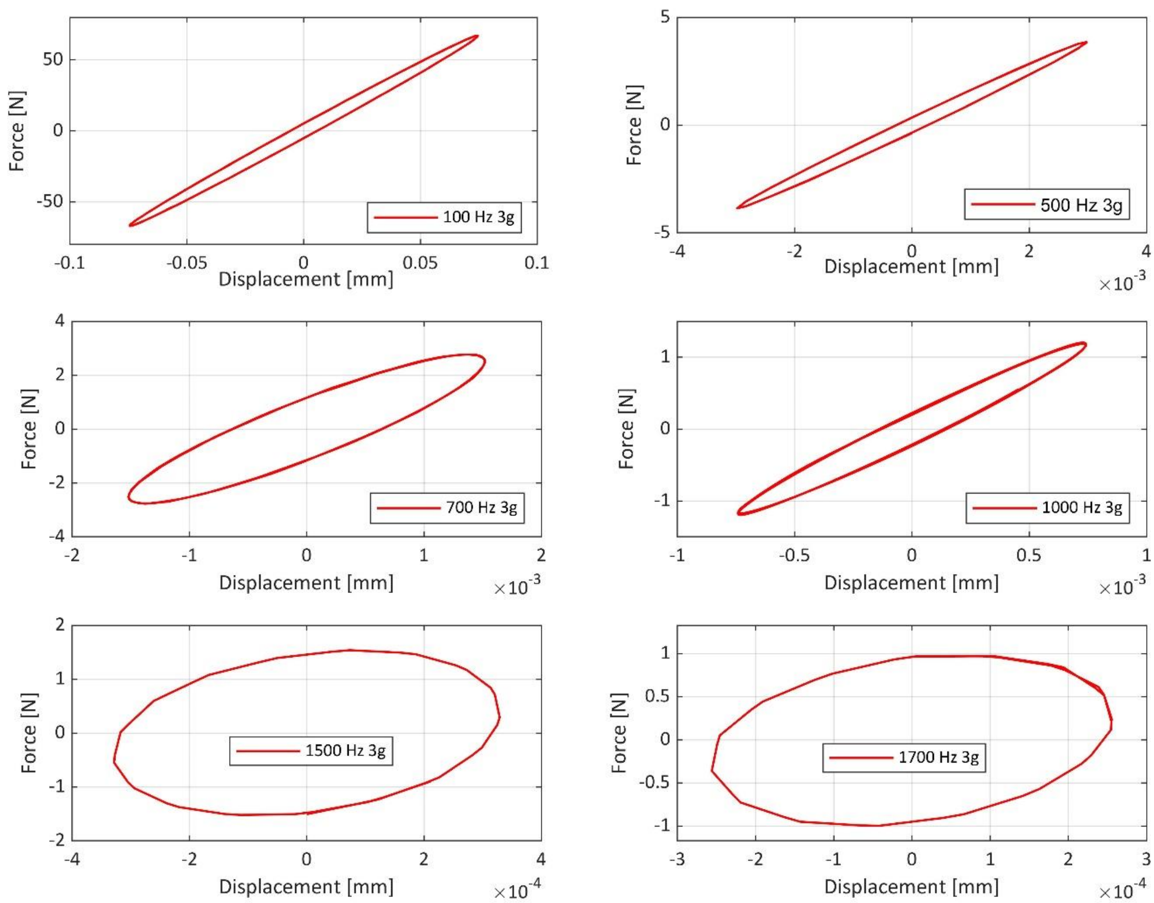

Fig. 4 Force-shift curves for selected frequencies

FEM model of the rubber mounting of the drive unit

For the simulation, a CAD model of the rubber mount was created, from which an FEM model was subsequently created (Fig. 6). Such a model has the advantage of increasing the efficiency of the rubber mount design because it reduces the time and cost of repetitive modified design procedures. Thanks to the accurate FEM model, dynamic properties can be determined by a limited number of experimental measurements. Identification of the parameters of the mathematical model is possible thanks to the results of FEM. Dynamic characteristics of the rubber mount were obtained by means of nonlinear dynamic harmonic analysis.

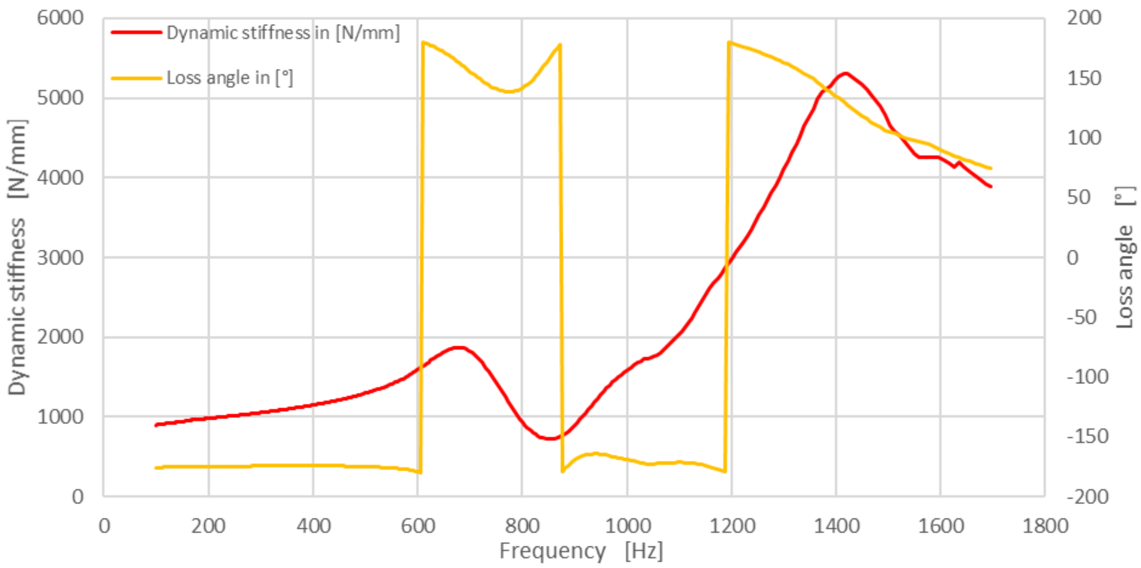

Fig. 5. Dynamic stiffness and angle of loss vs. frequency

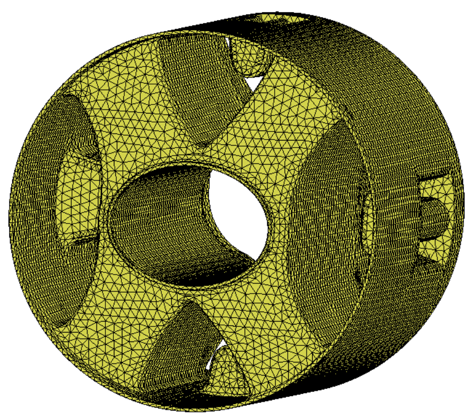

Fig. 6. FEM model of rubber mount

In contrast to the measured data, the rubber mount simulation was performed up to 2500 Hz (The measured data were limited by the resonance of the test rig frame). The resulting dynamic form of the stiffness model is shown in Fig. 7. The measurement and the FEM model together correlate to a frequency value of approximately 1100 Hz. Inaccuracies can be seen at higher frequencies. The resonant states of the test device frame or the clamping device could have caused these inaccuracies.

Mathematical model of dynamic properties of engine rubber mounting

To model the dynamic properties of elastomers, one of the mathematical models used is the so-called Bouc-Wen model. It was designed by Bouc in early 1967 and was generalized by Wen in 1976. It contains the following typical equations:

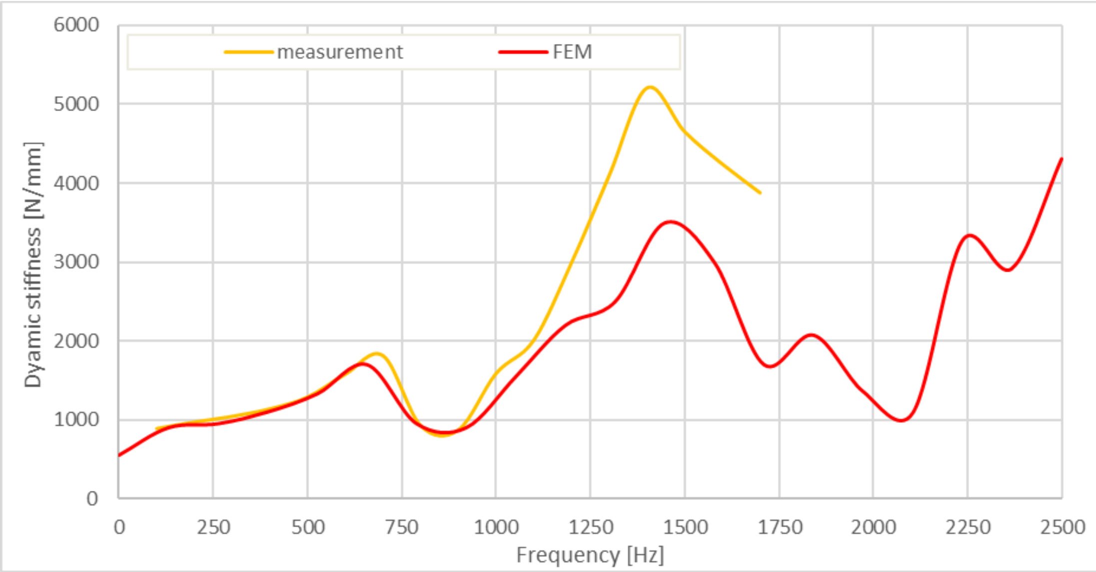

Fig. 7. FEM and measured dynamic stiffness and loss angle vs. frequency

$$\dot{z}(t) = A\dot{x}(t) - \gamma\dot{x}(t)|z(t)|^{n} - \beta|\dot{x}(t)|z(t)|z(t)|^{n-1} \tag{1}$$

$$F^{el}(t) = ak_{i}x(t) \tag{2}$$

$$F^{h}(t) = (1-a)k_{i}z(t) \tag{3}$$

$$F_{BW} = F^{el}(t) + F^{h}(t) \tag{4}$$

$$F_{R} = k_{0}x(t) + b_{0}\dot{x}(t) + F_{BW} \tag{5}$$

The force F_R is the total response, representing the equation of a system with one degree of freedom. The total force can be further reduced to stiffness, hysteresis and damping components. Elastic forces F_{el}, as well as hysteresis force F_h are both included within the hysteresis response component F_{BW}. Yield to pre-yield stiffness ratio (elastic ki) is being represented by the coefficient a. The value of a can never be higher than 1 and lower than 0. Basic shapes of hysteresis are represented by the dimensionless hysteresis parameter $z(t)$. The parameter $z(t)$ can be further reduced to four distinguished parameters. Parameter A for controlling the amplitude, β, γ and n [7,9].

During optimization, there is a problem with the estimation of parameters, while the objective function is the measured data. Parameter estimation was performed using the Matlab Parameter Estimation Toolbox program. In Fig. 8 is possible to see the measurement results in comparison with the simulation as a dependence of the displacement and the force.

7 Comparison of mathematical model and FEM model of rubber mounting

The calculation also determined the dependence of the dynamic stiffness on the frequency range of the rubber mount. Fig. 9 shows a comparison of mathematical, FEM, and measured results. The figure shows that there is a correlation between the measured and simulated data from the mathematical model. The reason is to tune the mathematical model, which was tuned by the measured data, not FEM data. It is also possible to tune the mathematical model from the data of the FEM model, which is the goal of future work.

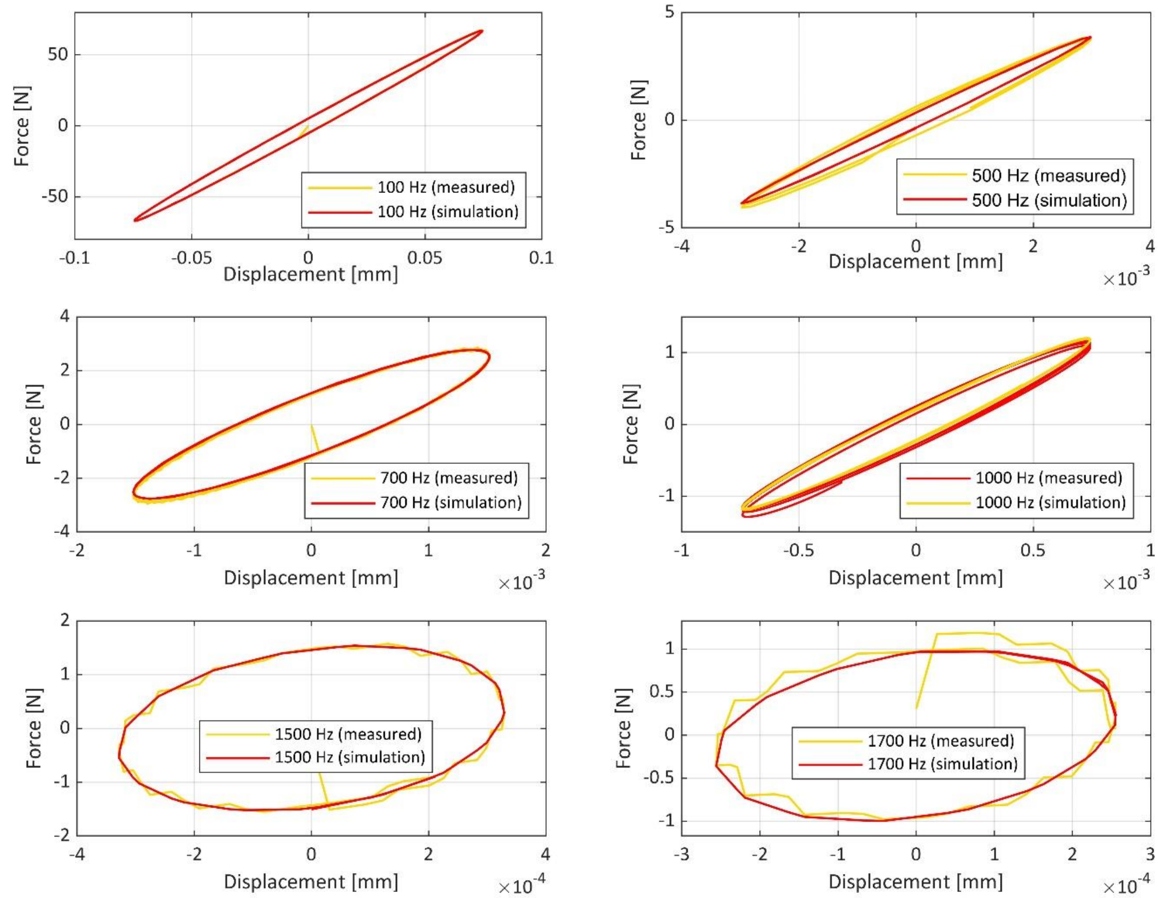

Fig. 8. Comparison of the simulation with the measured data for the selected frequency

CONCLUSION

The aim of this article is to point out the differences in the field of NVH between a propulsion system with an internal combustion engine and an electric propulsion system. Rubber mounts for electric drives are subject to various requirements, such as lower excitation amplitudes, a wide range of operating frequencies (up to 2000 Hz), changing force loads, and different torques. This can lead to various problems such as rubber resonance at high frequency and dynamic hardening.

Determining the dynamic properties of rubber elements by experiment is costly and time consuming, in which case the possibility of using simulation opens up. The aim is to create a corresponding mathematical model that can be used in a virtual vehicle model. In the case of simulation, it is also advisable to have as many experimental measurements as possible to fine-tune the model. The use of an accurate FEM model is suitable for identifying the parameters of a mathematical model.

In this article, the parameters of the mathematical model were tuned using experimental measurements of the rubber mount. The results of the mathematical model and experimental measurements correlated. The results of the experimental measurement might be affected by a test kit that may have resonated during the experiment. The aim of further work may be the design and manufacture of a test kit that can measure the required dynamic properties without being affected by resonance.

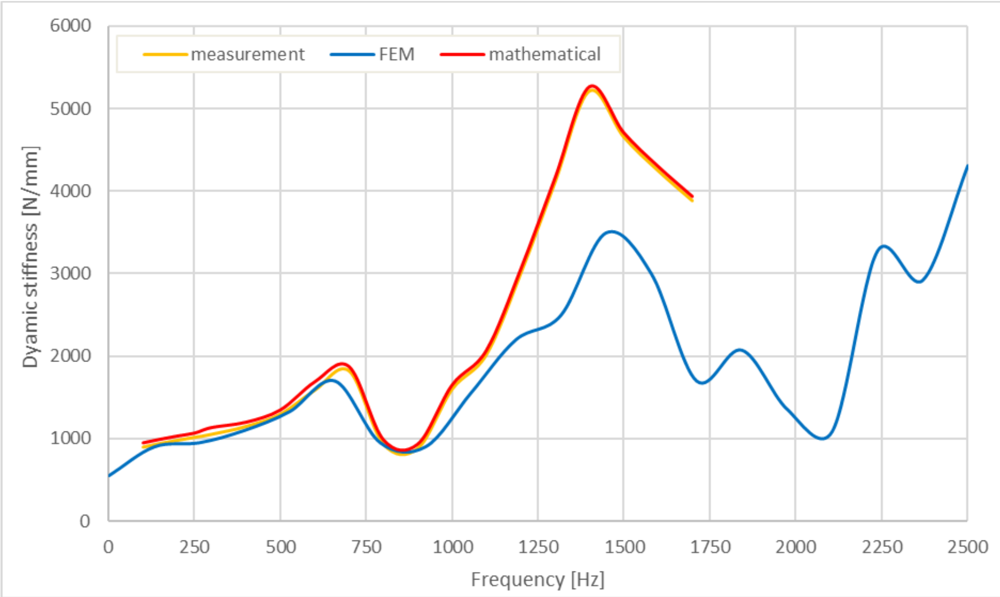

Fig. 9. Comparison of mathematical model and FEM model of rubber bearing

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