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A CONTRIBUTION TO THE STUDY OF THERMAL LOADS CAUSED BY VERTICAL OSCILLATORY EXCITATIONS IN PASSENGER VEHICLES WITH CONVENTIONAL AND ELECTRIC POWERTRAINS IN THE CONCEPTUAL DESIGN PHASE

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ORIGINAL SCIENTIFIC ARTICLE

ABSTRACT: This paper analyses the thermal loading of damping elements in passenger vehicles caused by vertical oscillatory excitations for both conventional internal combustion engine (ICE) powertrains and electric powertrains (EM). A simplified oscillatory model with three lumped masses is applied, comprising the unsprung mass, the sprung mass, and the powertrain mass. The excitations originate from road micro-irregularities and from inertial forces of the power unit, for which representative analytical models are adopted for both powertrain types.

By numerically solving a system of nonlinear differential equations, relative velocities and forces in the damping elements are obtained, on the basis of which the dissipated mechanical energy, i.e. thermal power, is calculated. For comparative analysis, RMS values of thermal power are used, together with frequency-domain analysis employing partial coherence functions.

The results show that thermal loads are highest in the tyres and lowest in the powertrain mounts. In the case of electric powertrains, lower thermal loads are observed in the suspension damping elements and powertrain mounts, whereas in vehicles with conventional powertrains lower loads occur in the tyres. It is concluded that the energy aspect of oscillatory processes represents an important design criterion in the conceptual vehicle design phase.

KEY WORDS: *passenger vehicle, powertrain type, oscillatory model, thermal loads*

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PRILOG ISRAŽIVANJU TOPLOTNIH OPTEREĆENJA IZAZVANIH VERTIKALNIM OSCILATORNIM POBUDAMA KOD KLASIČNOG I ELEKTROPOGONA PUTNIČKIH MOTORNIH VOZILA U FAZI IDEJNOG PROJEKTOVANJA

REZIME: U radu je analizirano toplotno opterećenje prigušnih elemenata putničkog motornog vozila izazvano vertikalnim oscilatornim pobudama kod klasičnog pogona sa motorom sa unutrašnjim sagorevanjem i kod elektropogona. Primenjen je pojednostavljen oscilatorni model sa tri koncentrisane mase, koji obuhvata neoslonjenu masu, oslonjenu masu i masu pogonske grupe. Pobude potiču od mikroneravnina puta i od inercijalnih sila pogonskog agregata, pri čemu su za oba tipa pogona usvojeni reprezentativni analitički modeli.

Numeričkim rešavanjem nelinearnih diferencijalnih jednačina određene su relativne brzine i sile u prigušnim elementima, na osnovu kojih je izračunata disipirana mehanička energija, odnosno toplotna snaga. Radi uporedne analize korišćene su RMS vrednosti toplotne snage, kao i analiza u frekventnoj oblasti primenom parcijalnih funkcija koherenci.

Rezultati pokazuju da su toplotna opterećenja najveća u pneumaticima, a najmanja u osloncima pogonske grupe. Kod elektropogona uočena su manja toplotna opterećenja prigušnih elemenata sistema za oslanjanje i oslonaca pogonske grupe, dok su kod klasičnog pogona manja opterećenja u pneumaticima. Zaključeno je da energetski aspekt oscilatornih procesa predstavlja značajan kriterijum u fazi idejnog projektovanja vozila.

KLJUČNE REČI: *Putničko motorno vozilo, vrsta pogona, oscilatorni model, toplotna opterećenja*

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INTRODUCTION

Oscillatory motions are an inevitable phenomenon in the dynamic systems of passenger vehicles and occur as a consequence of various internal and external excitations [1–6]. They have a pronounced negative impact on ride comfort and occupant health, increase dynamic loads on structural components, and accelerate material fatigue, ultimately leading to reduced system reliability and service life [4]. Therefore, the analysis of oscillatory processes represents one of the key tasks in the design and optimisation of vehicle suspension systems and powertrain assemblies [7].

In the existing literature, this problem has been extensively investigated from the standpoint of dynamic stability, vibration comfort, mass distribution, and transmission of oscillations from the powertrain and the road to the vehicle body [1–7]. Special attention has been devoted to modelling vehicles as multi-mass oscillatory systems, where, depending on the level of detail, models with two or three lumped masses are frequently used. However, relatively few studies systematically address the energy aspect of oscillatory processes, i.e. the transformation of mechanical work into thermal energy due to damping.

It is well known that the presence of damping elements in a system leads to energy dissipation, whereby the mechanical work generated by oscillatory motion is converted into heat. This phenomenon results in local and global thermal loads of damping elements and adjacent structural components. Elevated temperature affects the mechanical and rheological properties of materials, degrades damping characteristics, and may cause accelerated ageing and potential component failure.

For oscillatory processes of stochastic and nonlinear nature, the RMS value of dissipated thermal power represents a physically meaningful indicator of the effective energetic intensity of damping, and is therefore suitable for comparative assessment of different powertrain concepts.

It is particularly interesting to analyse these effects in passenger vehicles with different powertrain types, such as vehicles with conventional internal combustion engines and those with electric powertrains. These concepts differ significantly in the nature and spectral content of oscillatory excitations, the level of unbalanced masses, and the manner in which dynamic forces are transmitted to the suspension system. In this respect, the application of a three-mass oscillatory model enables the interaction between the powertrain, suspension, and vehicle body to be examined, as well as the quantification of the associated thermal loads in the conceptual design phase.

The objective of this study is to quantitatively assess the thermal loads of damping elements caused by vertical oscillatory excitations during the conceptual design phase of a passenger vehicle and to perform a comparative analysis of vehicles with conventional and electric powertrains. Particular emphasis is placed on the energy aspect of oscillatory processes, namely the transformation of mechanical work resulting from relative motion into thermal energy dissipated in the damping elements.

Unlike most existing studies that primarily focus on vibration comfort, stability, and oscillation transmission [1–7], this paper employs dissipated thermal power as the principal criterion for comparing different powertrain concepts. By applying a nonlinear oscillatory model and analysing the system response in both the time and frequency domains, it is possible to separate the contributions of road micro-irregularities and powertrain excitations to the overall energy response of the system.

The contribution of this work lies in introducing the thermal load of damping elements as an additional design criterion when selecting a powertrain concept and dimensioning suspension systems and powertrain mounts, particularly in the early stages of vehicle development when many design parameters are still unknown.

Unlike conventional analyses based on accelerations, forces, or RMS displacement values, this study uses the thermal power dissipated in damping elements as the primary comparison criterion, as it directly reflects the energetic intensity of oscillatory processes.

1 METHOD

The aim of this study is to investigate, based on the analysis of oscillatory processes in the adopted dynamic model, the thermal loads arising from damping in both conventional and electric powertrains, and to highlight their influence on the operational characteristics and reliability of the system.

For the analysis of thermal loads caused by oscillatory processes, a simplified dynamic model of a passenger vehicle with three lumped masses is employed [3,4]. The use of such a model is considered justified and useful, given that the subject of investigation is primarily vertical vibrations and that all initial structural and material parameters of the system are known for the selected model. The three-mass model represents an acceptable compromise between realistic representation of vehicle dynamics and numerical complexity.

The method is applied to a passenger vehicle with a total mass of approximately 2 t, considered in two powertrain configurations: an internal combustion engine (ICE) and an electric powertrain (EM). The following masses are included in the model:

- unsprung mass, comprising the wheel, tyre, part of the axle, and associated suspension components,
- sprung mass, representing the vehicle body together with payload and passengers,
- powertrain mass, which for ICE vehicles includes the engine, gearbox, clutch, and associated structural components, while for electric vehicles it includes the electric motor, inverter, reduction gear, and related elements.

Vertical oscillations resulting from road micro-irregularities and inertial vertical excitations originating from the powertrain are considered. These excitations differ in character and intensity between ICE and electric powertrains, as discussed later.

Using dynamic simulation based on the system of differential equations of motion, time histories of vertical displacements, velocities, and accelerations of the unsprung mass, sprung mass, and powertrain mass are obtained. Particular attention is paid to relative motions between the masses, which are directly relevant to the operation of damping elements.

Based on the obtained relative forces and relative velocities in the damping elements, the mechanical work dissipated by damping, i.e. the corresponding thermal power, is calculated using the first law of thermodynamics [8,9]. This enables quantification of the thermal loads resulting from oscillatory processes in the system.

Given that the oscillatory excitations are of random or polyharmonic nature, comparison of different powertrain configurations is most appropriately carried out using RMS (root mean square) values of the calculated thermal power. The results are presented in tabular form and serve as a basis for comparative analysis of conventional and electric powertrains.

In addition to time-domain analysis, frequency-domain analysis is also performed. This includes calculation of ordinary coherence functions between excitations arising from road micro-irregularities and inertial vertical forces of the powertrain, with the aim of assessing the degree of linear dependence between excitation signals in different frequency ranges.

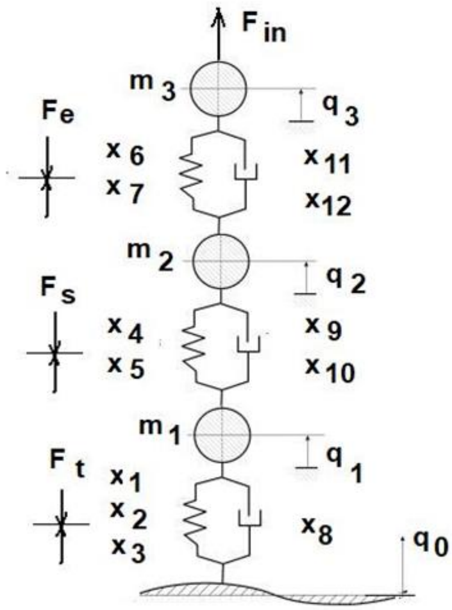

Figure 1. Oscillatory model of the passenger vehicle used

1.1 Mathematical model for the analysis of vertical vehicle oscillations

A large number of vehicle oscillatory models of different levels of complexity are available in the literature [1–7]. When selecting an appropriate model, it is necessary to use only the structure that can provide a reliable answer to the defined research problem. The application of models more complex than minimally required is not recommended, since such models require a large number of oscillatory parameters that are often not experimentally verified or are burdened with significant uncertainties [10].

Considering that in this study it is necessary to analyse the thermal power dissipated in damping elements for passenger vehicles equipped with an internal combustion engine (ICE) as well as with an electric motor, a three-degree-of-freedom model was assessed as an appropriate compromise between simplicity and physical representativeness. The adopted model is shown in Figure 1 and includes vertical oscillations of the sprung mass, the unsprung mass, and the powertrain mass.

Since models of this type have been described in detail in the available literature [4], only the basic relational expressions required for the formulation of the equations of motion are presented here. For clarity, the expressions are given in expanded rather than compact form.

With reference to Figure 1, the relative deformations and relative velocities of the elastic–damping elements are defined as:

$$\begin{aligned} &D_1 = q_1 - q_0 \quad \dot{D}_1 = \dot{q}_1 - \dot{q}_0 \\ &D_2 = q_2 - q_1 \quad \dot{D}_2 = \dot{q}_2 - \dot{q}_1 , \\ &D_3 = q_3 - q_2 \quad \dot{D}_3 = \dot{q}_3 - \dot{q}_2 \end{aligned} \tag{1}$$

where:

• q_i, $\dot{q}_i$ - denote the corresponding vertical displacements and velocities of the considered masses.

The forces in the elastic–damping elements are given by [4]:

$$\begin{aligned} &F_1 = x_1 D_1 + x_2 D_1^2 + x_3 D_1^2 + x_8 D_1 \\ &F_2 = x_4 D_2^3 + x_5 D_2^3 + x_9 \dot{D}_2 + x_{10} \dot{D}_2^2 Sign(\dot{D}_2) , \\ &F_3 = x_6 D_3 + x_7 D_3^3 + x_{11} \dot{D}_3 + x_{12} \dot{D}_3^2 Sign(\dot{D}_3) \end{aligned} \tag{2}$$

where:

•x_1........x_{12} are the corresponding oscillatory parameters, expressed in such units that multiplication by the relative deformation or relative velocity yields force in newtons (N).

Based on the adopted model (Figure 1) and applying Newton's laws, the differential equations of motion for each of the three masses can be written as:

$$\begin{aligned} &m_1 \ddot{q}_1 = F_2 - F_1 \\ &m_2 \ddot{q}_2 = F_3 - F_2 , \\ &m_3 \ddot{q}_3 = F_{in} - F_3 \end{aligned} \tag{3}$$

where F_{in} represents the vertical oscillatory excitation generated by the operation of the ICE or the electric motor, originating from unbalanced masses, electromagnetic forces, or periodic inertial effects of the powertrain.

Considering the stochastic nature of excitations caused by road micro-irregularities, the deterministic–periodic character of powertrain excitations, and the pronounced nonlinearity of forces in the elastic–damping elements, the equations of motion cannot be solved analytically. Therefore, the system of differential equations is solved numerically, using the fourth-order Runge–Kutta method.

For numerical integration, the system of second-order differential equations is transformed into an equivalent system of nonlinear first-order differential equations, which is a standard procedure and will not be discussed further here. Since the equations of motion include excitations from inertial forces and road micro-irregularities, these will be briefly described below.

1.2 Inertial forces of the internal combustion engine

In this study, a four-cylinder inline internal combustion engine is considered. In an ideal case, the primary inertial forces cancel each other, while the secondary components remain unbalanced and represent a significant source of vertical oscillatory excitations in the vehicle system, especially in the low- and mid-frequency ranges [11].

As is well known from the classical theory of slider–crank mechanism dynamics, such engines generate unbalanced inertial forces resulting from the reciprocating linear motion of the piston–connecting rod assembly [11]:

$$F_{in} = 4 m_k r \omega^2 \lambda \cos(2\omega t), \tag{4}$$

where:

- m_k - is the piston mass together with the associated portion of the connecting rod mass,
- r – is the crank radius,
- ω - is the angular speed of the crankshaft, and
- λ - is the ratio of the crank radius to the connecting rod length. ($\lambda = \frac{r}{l}$).

1.3 Excitations generated by the electric motor

Unlike internal combustion engines, electric motors do not include masses undergoing linear motion; however, oscillatory excitations occur due to [12]:

- unbalanced rotor masses (mechanical eccentricity),
- non-uniform air gap between rotor and stator,
- spatial and temporal harmonics of the electromagnetic field, and
- sideband components in the spectrum resulting from the interaction between eccentricity and the electromagnetic field.

The total excitation force of an electric motor is predominantly radial, and its vertical component acts directly on the vehicle structure through the motor mounts.

Based on available experimental and analytical results, the vertical excitation force of an electric motor can be approximated by [12]:

$$F_{in} = k_r \sum_{1}^{p\max} \varepsilon_p \frac{P_e}{\omega} \sin\left(p\omega t + \varphi_p\right), \tag{5}$$

- where:
- P_e – is the electric motor power (W),
- k_r – is the coefficient transforming tangential force into radial force (in this study, represents an effective energetic coefficient transforming electromagnetic torque into a radial excitation force),
- ω - is the rotor angular speed,
- p – is the harmonic order,
- p=1 – corresponds to the harmonic describing mechanical eccentricity,
- p=2,3, – correspond to harmonics originating from the electromagnetic field and sidebands,

- ε_p - are coefficients describing spatial and temporal harmonics of the electromagnetic field,
- φ_p – are phase angles of individual harmonics.

Expression (5) represents an energetic approximation, where the electromagnetic torque is transformed into an effective radial excitation force via the coefficient. It should be noted that k_r is not a universal constant but a combined coefficient that includes:

- electric motor geometry,
- pole–slot combination,
- air-gap non-uniformity, and
- motor mounting configuration.

Based on experimental data and [12], $k_r \approx 0.02–0.08$ can be adopted, while the most commonly used representative value in analytical models is $k_r=0.05$. It should be emphasized that the coefficient k_r is not intended to represent a direct physical constant, but rather an effective energetic transformation coefficient suitable for conceptual-level analysis.

The number of poles in electric motors used in passenger vehicles typically ranges from 6 to 8, which leads to dominant excitations in the mid- and high-frequency range (above 100 Hz). Compared to ICE engines, force amplitudes are usually smaller, but frequencies are significantly higher, which has important consequences for the vibro-acoustic behaviour of the vehicle.

Empirical data indicate that the maximum radial excitation of an electric motor is typically within:

$$F_{in\max} \approx (0,5 \div 2)\% G_{mot} \ ,$$

where G_{mot} is the weight of the electric motor.

In the following analysis, only the dominant harmonics p=1,2,3 are considered, with coefficients describing the intensity of mechanical and electromagnetic asymmetries.

Empirical data [12]: $n \leq 1200\text{min}^{-1}$, ε_1=0,01-0,03, ε_2=0,005-0,02. ε_3=0,002-0,01, are adopted data for analysis [12]: P_e=150000W, ε_1=0,02, ε_2=0,01, ε_3=0,005

1.4 Oscillatory excitations due to road micro-irregularities

Numerous data on road micro-irregularities are available in the literature, most of which are based on power spectral density definitions, which prevents the direct application of the inverse Fourier transform to determine the time-domain excitation function [4]. Therefore, this study adopts and briefly describes a procedure for defining the excitation time function according to [4].

The excitation function is assumed to be of the following form:

$$q_0(t) = \sum_{1}^{N} A(f) \sin\left[2\pi f t + \varepsilon(f)\right], \qquad (6)$$

In this study, a polyharmonic excitation function with 83 harmonics distributed over the frequency range from 0.05 to 500, Hz is used.

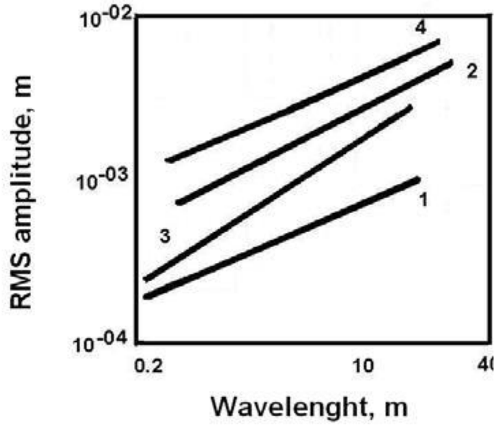

Figure 2. Limit values of road roughness (1 – modern road, 2 – road with poor concrete surface, 3 – repaired asphalt road, 4 – city street in good condition)

Based on Figure 2 [4], the following relation can be written:

$$A(f) = (A_o + B_o f)/v, \tag{7}$$

In the absence of more precise data, the phase angle is defined by:

$$\varepsilon(f) = 2\pi(rnd - 0.5), \tag{8}$$

where *rnd* denotes a set of random numbers uniformly distributed in the interval {0, 1}.

1.5 Thermal Loading of Damping Elements

The mechanical work generated in damping elements due to the action of relative velocities and damping forces is completely dissipated in the form of heat. The instantaneous dissipated power in a damping element is equal to the product of the damping force and the relative velocity between the connected masses. Since the damping force is a function of the relative velocity, the dissipated power exhibits a pronounced nonlinear character and directly depends on the oscillatory regime of the system.

The total thermal power generated in the damping elements represents the time integral of the instantaneous dissipated power and can be used as a quantitative measure of energy losses in the suspension system and powertrain mounts. This approach enables a direct connection between the dynamic behavior of the vehicle and the thermal loading of damping elements, which is of particular importance when comparing conventional and electric powertrains.

In the present case, the thermal power was calculated using the following expression:

$$P_t = F_{vrel} v_{rel}, \tag{9}$$

where:

- F_{vrel} - is the damping force in the corresponding damping element, and
- v_{rel} – is the corresponding relative velocity calculated using Eq. (1).

Considering the nature of the solutions of the differential equations (expressions 1-9), it is evident that the thermal power is also of a stochastic character, which will be discussed later.

Since the instantaneous thermal power exhibits a pronounced quasi-random character due to the stochastic nature of road excitations and nonlinear damping, RMS values were adopted as a representative measure of the effective thermal loading of individual damping elements.

2 DYNAMIC SIMULATION

The system of differential equations describing the oscillations of the observed passenger vehicle was solved numerically using the fourth-order Runge–Kutta method. For comparison purposes, both the internal combustion engine (ICE) and the electric motor (EM) were assumed to have a rated power of 150, kW and to operate at 1500, rpm under steady vehicle motion. It should be noted that this represents a simplified operating condition, particularly for electric vehicles, since power recuperation and similar effects were not considered.

During the conceptual design phase of a vehicle, many parameters are not precisely known-therefore, approximate relationships are commonly used. The mass of the powertrain can be related to the rated power of the propulsion unit. For ICE-driven vehicles, the ratio of maximum power to powertrain mass (engine, gearbox, clutch, and part of the driveline) typically ranges from 0.3 to 0.6, kW/kg, whereas for electric powertrains (electric motor, inverter, reducer) this ratio is significantly higher, ranging from 1.5 to 3, kW/kg. These relationships allow the estimation of the powertrain mass

In addition, empirical relations exist between the powertrain mass and the total vehicle mass. For ICE vehicles, the powertrain mass accounts for approximately 18 to 25,% of the total vehicle mass, while for electric vehicles this share is typically 8 to 12% (the battery is not included; in practice, the sprung mass of electric vehicles is usually 30 to 40% higher than in ICE vehicles).

In the present study, the inertial parameters listed in Table 1 were adopted.

Table 1. Inertial parameters of the analyzed vehicle

Powertrain type	m_1, kg	m_2, kg	m_3, kg
ICE	80	1600	375
EM	90	2150	75

For comparison purposes, it was considered appropriate to assume identical elasto-damping element properties for both powertrain types. These parameters are listed in Table 2.

Table 2. Characteristics of elasto-damping elements

Parameter	Value
x_1, N/m	$2*10^5$
x_2, N/m^2	$1*10^5$
x_3, N/m^3	$1*10^5$
x_4, N/m	$3*10^5$
x_5, N/m^3	$1*10^5$
x_6, N/m	$8*10^4$
x_7, N/m^3	$8*10^4$
x_8, Ns/m	$8*10^3$

x_9, Ns/m	$7*10^3$
x_{10}, Ns^2/m^2	$7*10^3$
x_{11}, Ns/m	$8*10^3$
x_{12}, Ns^2/m^2	$8*10^3$

In addition to these parameters, an equivalent piston mass of 1, kg was used for the ICE. An eight-pole electric motor was considered with the following parameters: ε_1=0,01; ε_2=0,01 ε_3=0,01, k_r=0,1.

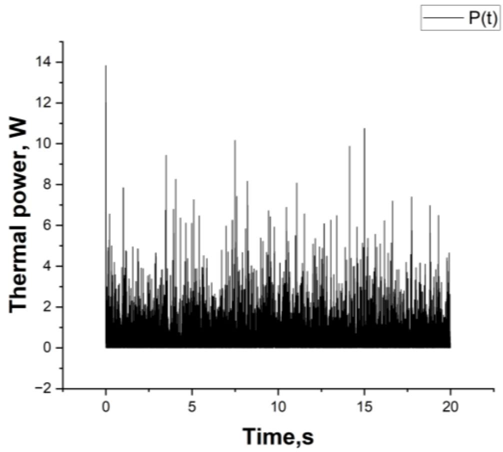

Figure 3. Time history of thermal power dissipated in the powertrain mounting damper of an electric vehicle

Numerical integration was performed using 20000 time steps with a time increment of 0.001, s, ensuring reliable analysis in the frequency range from 0.05 to 500, Hz [13]. This range covers the oscillations of the considered lumped masses, excitations due to road micro-roughness, and vertical excitations originating from both ICE and electric motors.

Based on the calculated relative velocities, the thermal power was determined using expression (1-9) for all damping elements of the vehicle and for both powertrain types. For illustration purposes, Figure 3 shows the time variation of the thermal power dissipated in the powertrain damper of the electric vehicle.

Analysis of all obtained time histories indicated that they are of a quasi-random nature. Therefore, for further analysis, RMS values were calculated and are presented in Table 3.

Table 3. RMS thermal power

	ICE - RMS, W	EM - RMS, W
Tire	3525741.6	3611390.5
Suspension system	42334.7	32103.5
Power train mounting	3.429	1.070

For more detailed analysis, it was considered appropriate to apply an approach commonly used in automatic control theory [14,15]. The oscillatory model was treated as a system with two inputs (road micro-roughness excitation and inertial force of the powertrain) and three outputs (thermal power in the tires, suspension system, and powertrain mounts), as illustrated in Figure 4a.

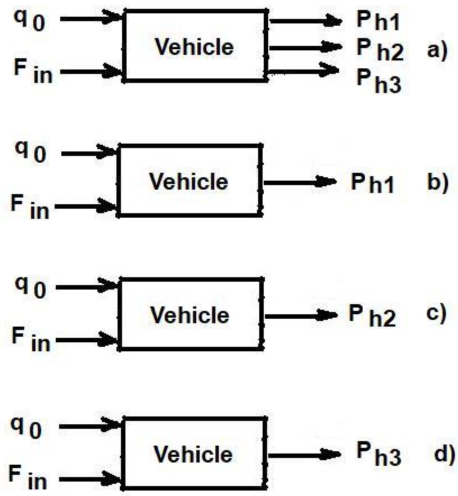

Figure 4. MIMO system (a) and its transformation into MISO systems (b, c, and d)

Such systems are known as multi-input multi-output (MIMO) systems [14,15]. They are analyzed by transforming them into multi-input single-output (MISO) systems (Figures 4b–4d), under the condition that the input signals are mutually independent. To verify this condition, the ordinary coherence function between the two analyzed inputs (F_{in} , q_0), was calculated. The obtained results are shown in Figure 5.

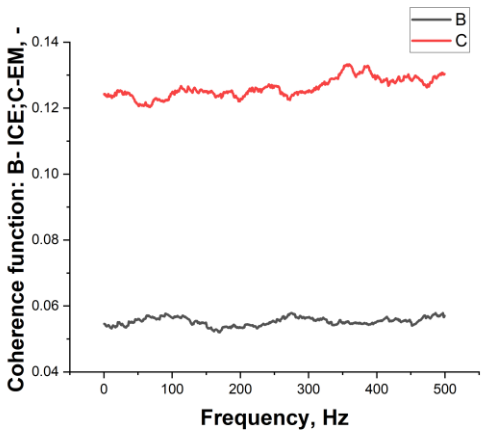

Figure 5. Ordinary coherence functions of the input signals: road micro-roughness excitation and vertical component of the powertrain inertial force; B – ICE, C – EM

Analysis of Figure 5, shows that the ordinary coherence function is approximately 0.06 for the ICE and about 0.13 for the electric motor. Both values are well below 0.3, which is commonly considered the lower threshold for significant frequency-dependent energetic coupling between input signals [13]. This indicates that signal decoupling is not required.

Based on the adopted MIMO approach, partial coherence functions were calculated, enabling a more reliable assessment of the influence of individual excitations on the system responses in the presence of multiple simultaneous inputs. This approach allows clearer separation of the contributions of road micro-roughness and powertrain inertial excitations to the overall dynamic and energetic response of the system.

Accordingly, partial coherence functions were calculated using a custom program developed in Dev Pascal. The results are shown in Figures 6–11.

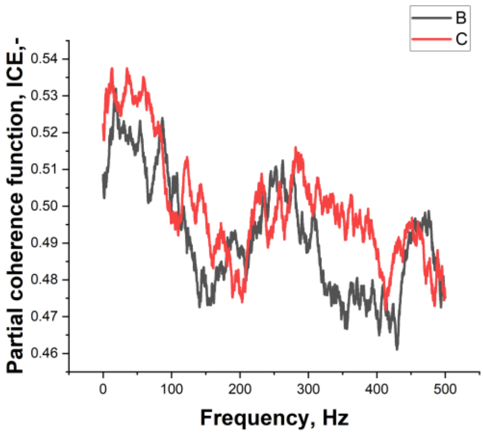

Figure 6. Partial coherence functions for the tire in ICE-driven vehicles: B – powertrain inertial forces, C – road micro-roughness

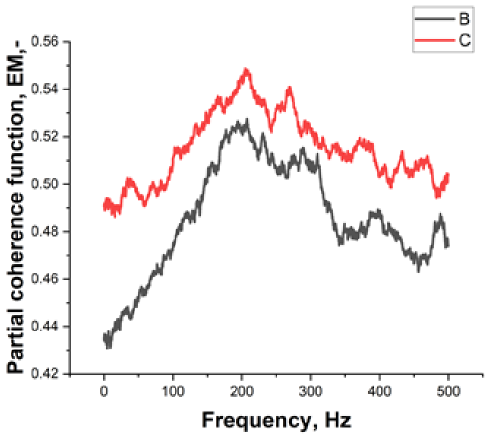

Figure 7. Partial coherence functions for the tire in EM-driven vehicles: B – powertrain inertial forces, C – road micro-roughness

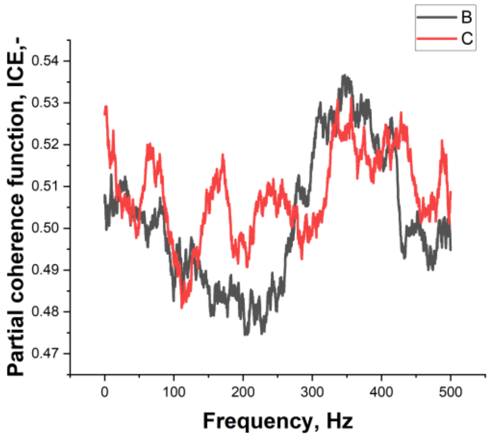

Figure 8. Partial coherence functions for the suspension system in ICE-driven vehicles: B – powertrain inertial forces, C – road micro-roughness

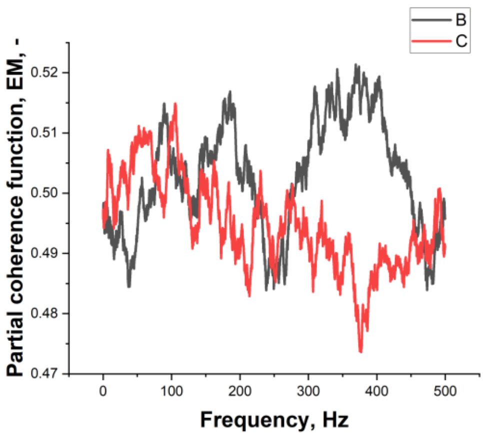

Figure 9. Partial coherence functions for the suspension system in EM-driven vehicles: B – powertrain inertial forces, C – road micro-roughness

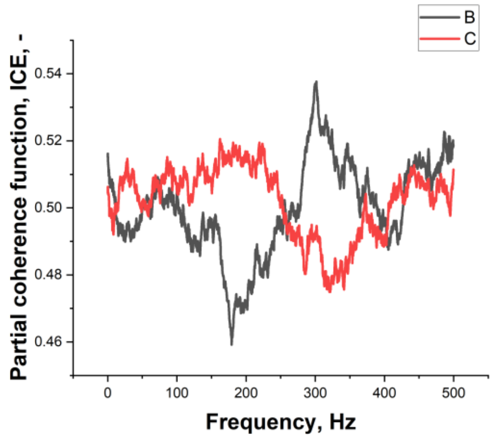

Figure 10. Partial coherence functions for powertrain mounts in ICE-driven vehicles: B – powertrain inertial forces, C – road micro-roughness

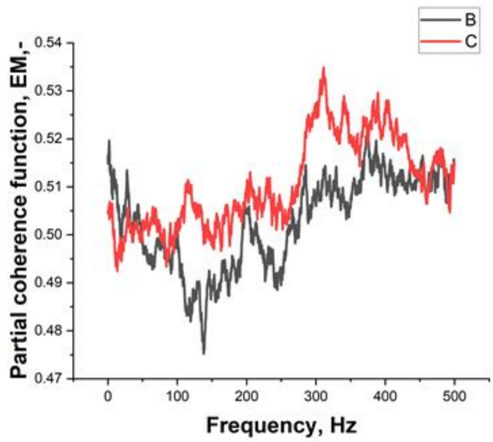

Figure 11. Partial coherence functions for powertrain mounts in EM-driven vehicles: B – powertrain inertial forces, C – road micro-roughness

3 DATA ANALYSIS

The analysis begins with Table 3. It is evident that, for both types of powertrain, the thermal power is highest in the tires and lowest in the powertrain mounts. When comparing the considered powertrain systems, the thermal power in the electric powertrain is higher in the tires than in the ICE-powered vehicle, whereas it is lower in the remaining damping elements. This can be explained by the fact that the sprung mass is significantly larger in the electric vehicle considered.

In practice, this issue is addressed by an appropriate selection of elasto-damping elements; however, in the present study, identical characteristics were assumed for both powertrain types in order to ensure comparability.

The obtained results reveal markedly different orders of magnitude of the thermal power dissipated in individual damping elements of the system. At first glance, it may appear unexpected that the RMS values of thermal power in the tires are of the order of megawatts, while those in the powertrain mounts are of the order of watts. However, this relationship is fully consistent with the physical nature of the processes involved and with the overall energy balance of the system.

The tire represents the primary element in the transmission chain of excitations originating from road micro-roughness and is subjected to the largest relative deformations and velocities, particularly over a wide frequency range. At the same time, damping forces in the tire are substantial, and energy dissipation occurs continuously along the tire–road contact patch, resulting in very high instantaneous and RMS thermal power values.

It should be emphasized that this energy is not accumulated within a single structural component but is distributed over a large tire volume and dissipated into the surrounding environment during vehicle motion. It is also noted that the influence of tire dimensions on the oscillatory excitation of the tire center was not included in the present model; therefore, the predicted excitations are more severe than in reality, leading to an overestimation of the thermal power dissipated in the tire.

In contrast, powertrain mounts are designed to operate with significantly smaller relative displacements and velocities, and their primary function is vibration isolation rather than the dissipation of large amounts of energy. Consequently, the thermal power dissipated in these elements is several orders of magnitude lower and remains within the watt range. The suspension system occupies an intermediate position, with thermal loads on the order of tens of kilowatts, which corresponds to its role in absorbing oscillatory energy between the unsprung and sprung masses.

The pronounced differences in the orders of magnitude of thermal power indicate that the energy balance of oscillatory processes cannot be properly assessed without clearly distinguishing the roles of individual damping elements. Furthermore, direct comparison of absolute values is meaningful only within the same component and for the same type of powertrain.

Analysis of Figure 6 shows that road micro-roughness has a greater influence on the thermal power dissipated in the tire, while the frequency-dependent energetic coupling decreases with increasing frequency. The highest values occur in the regions around 50, Hz and 300 Hz. The fact that the partial coherence function is less than unity indicates the presence of significant nonlinearities in the system. A similar behavior is observed for the electric powertrain (Figure 7), with the difference that the maximum coupling occurs at approximately 180 Hz.

Figure 8 shows that, over a wide frequency range, road micro-roughness dominates the thermal power in the suspension system for the ICE-powered vehicle, except around 300, Hz. The highest frequency-dependent energetic coupling occurs near 350 Hz. In the case of the electric powertrain, road micro-roughness also has a dominant influence over a broad frequency range, except around 50, Hz (Figure 9), with the strongest contribution observed near 350, Hz. These phenomena can be explained by the nonlinear dynamic nature of the vehicle suspension system.

From Figure 10, it can be observed that road micro-roughness has a greater influence up to approximately 250, Hz, while around 300, Hz the influence of engine inertial forces becomes dominant. Up to about 500, Hz, the frequency-dependent energetic coupling is relatively uniform. In the case of electric propulsion, road micro-roughness exhibits a stronger influence on the thermal power over a broader frequency range, as shown in Figure 11.

Based on the preceding analysis, it can be concluded that a frequency-dependent energetic coupling exists between the considered excitation sources in the vehicle and the thermal power dissipated in all damping elements. The influence of these excitations is not identical for the two powertrain types, which can be attributed to the nonlinear behavior of the system. Therefore, in subsequent stages of the project, further research should be conducted using more accurate input data and appropriately optimized elasto-damping elements for both propulsion concepts.

The applied oscillatory model is intended for use in the conceptual design phase and, accordingly, has certain limitations. The influence of road slope was not considered, which implies that additional static and dynamic load components arising from changes in normal reactions and mass redistribution during uphill or downhill driving were neglected.

Furthermore, the model does not account for the longitudinal dynamics of the vehicle, such as acceleration, deceleration, and variations in driving torque; therefore, only approximately steady-state operating conditions were analyzed. In addition, temperature feedback effects-i.e., the dependence of elasto-damping element properties on temperature-were not included, although it is well known that temperature rise can significantly affect stiffness and damping characteristics. Incorporation of these effects represents a logical direction for further model development and would enable a more realistic assessment of thermal loads under real operating conditions.

4 CONCLUSION

This paper investigated the influence of the powertrain type on the thermal loading of damping elements in a passenger motor vehicle subjected to vertical oscillatory excitations. By applying a multi-mass oscillatory model with three lumped masses and a nonlinear description of elasto-damping elements, a quantitative assessment of mechanical energy dissipation and its transformation into heat was achieved.

The results of numerical simulations show that, for both propulsion types, the highest thermal loads occur in the tires, while the lowest are observed in the powertrain mounts. Compared with conventional ICE propulsion, electric vehicles exhibit lower thermal loads in the damping elements of the suspension system and powertrain mounts, which can be attributed to lower excitation force amplitudes and a different mass distribution. On the other hand, the higher sprung mass associated with electric powertrains leads to increased thermal loads in the tires.

Frequency-domain analysis based on partial coherence functions demonstrated that the contributions of road micro-roughness and powertrain excitations depend on the frequency range and the type of propulsion, and that neither excitation source can be neglected. A pronounced nonlinear behavior of the system was observed, confirming that the energetic response cannot be reliably predicted using linear models.

The investigations conducted in this study confirm that the analysis of thermal loading in damping elements can serve as an important additional criterion during the conceptual vehicle design phase, particularly in the selection of the propulsion concept and the sizing of elasto-damping elements. Further research should focus on the application of more advanced vehicle models, inclusion of temperature-dependent damping characteristics, and consideration of realistic operating conditions, including acceleration, deceleration, and energy recuperation in electric vehicles.

RMS thermal power was shown to be a robust and physically meaningful criterion for comparing the energetic impact of different powertrain concepts under stochastic and nonlinear oscillatory conditions.

The obtained results indicate that the choice of propulsion concept affects not only the dynamic and vibro-acoustic response of the vehicle but also the energetic and thermal loading of key damping elements, which should be taken into account already in the early stages of vehicle design.

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