Automotive Experiences

Vol. 8 No. 1 (2025) pp. 98-108

p-ISSN: 2615-6202 e-ISSN: 2615-6636

Research Paper

Computational and Experimental Study of the Dynamic Loading of the Tracked and Wheeled Vehicles Powertrain System in Harsh Climatic Conditions

Alexander Igorevich Taratorkin, **Igor Alexandrovich Taratorkin**, **Victor Borisovich Derzhanskii**, **Alexander Alexandrovich Volkov**

Institute of Engineering Science, Ural Branch of the Russian Academy of Sciences (IES UB RAS), 620049, Ekaterinburg, Russia

alexandr_tar@mail.ru

https://doi.org/10.31603/ae.11581

Published by Automotive Laboratory of Universitas Muhammadiyah Magelang

Article Info
Submitted: 13/07/2024
Revised: 27/03/2025
Accepted: 13/04/2025
Online first: 13/04/2025

Abstract

The article presents the results of a computational and experimental study of the dynamic loading of the powertrain system of the transport vehicles, which include a diesel engine and a hydromechanical transmission. The purpose of the study is a computational and experimental substantiation of the applicability of torsion bar spring between the engine and transmission as a torsional oscillation damper in the powertrain system with diesel engines of increased power and high-density arrangement of the block elements, which makes it difficult to apply traditional measurement methods. The novelty of the research consists in the development of a new method to determine experimentally the dynamic torque in the powertrain system, characterized by the fact that during processing, the signal of the engine shaft speed sensor is digitized and transmitted to the recording and processing device. Based on the direct Fourier transform, the amplitude-frequency response function of the torque is determined, including the main motor harmonics, harmonic components formed by the crankshaft and connecting rod and gas valve timing mechanisms of the engine, the generator drive, oscillations in the transmission, etc. It is established that the reason for the torsion bar springs durability limitation is their operation in off-design powertrain system modes, due to the occurrence of a phenomenon called «collision of tasks». Based on probabilistic calculation methods, it is shown that when such modes occur, the probability of failure of elastic torsion bar increases from 0.0001 to a shocking 0.29. According to the research results, it is concluded that torsion bar springs are applicable as torsional vibration dampers in high-duty powertrain systems.

Keywords: Dynamic loading; Powertrain system; Transmission and drivetrain; Torsion bar; Contactless method

1. Introduction

The current trends in the development of powertrain systems of prospective transport vehicles involve both the creation of hybrid and electromechanical structures. At the same time, the increased mobility of modern tracked vehicles operated in the harsh climatic conditions of the subpolar and polar regions of the Arctic is largely provided by traditional powertrain system (PS) with increased power, including an internal combustion engine and a hydrodynamic transmission (with torque converter). At the same time, the arrangement of the engine and transmission compartment (ETC) is still subordinated to the interests of achieving the maximum density of the arranging of the engine, transmission and their systems in order to minimize the volume of ETC to ensure ease of maintenance and assembling and dismantling work. The more perfect the design of the powertrain system, the smaller its volume, the denser the arrangement and, in general, the smaller the volume of the ETC.

The purpose of this study is a computational and experimental substantiation of the applicability of elastic torsion bar springs between the engine and transmission as a torsional vibration damper in powertrain system with diesel engines of increased power.

The objectives of the study are:

- Development of a contactless method for experimental determination of the dynamic torque using advanced hardware and software;
- Determination of the dynamic loading of the elastic bar, including in case of off-design loading modes;
- Probability estimation of elastic bar destructive failure in the calculated and off-design modes.

Diesel engine is the main source of dynamic loads in the transmission elements and, first, of the loads caused by torsional vibrations, which are most pronounced in transients - engine start and stall, acceleration and braking of the vehicle, gear shifting. The main danger is represented by resonant zones, when the frequency of disturbance effects from the engine coincides with one of the eigenfrequencies of the powertrain system [1], [2].

The cause of torsional vibrations is the periodic nature of the torque and force change on the crankshaft and connecting rod masses of the engine. The more cylinders the engine has and the more regularly the rotation angle of the engine crankshaft initiation of combustion occurs in the cylinders, the more the amplitude of the torque at the transmission input is smoothed out. However, the problem of protecting the transmission from torsional vibrations remains quite critical for mechanical and, especially, for hydromechanical transmissions. Torsional vibrations lead to high dynamic loading in the transmission elements and limit their lifetime [3].

Most companies that develop transport vehicles when creating ETC use special additional devices (dampers, anti-vibrators and other elastic dampers of torsional vibrations) that perform the function of protecting the transmission from dangerous torsional vibrations. This is often combined with the installation of large flywheels on the crankshaft, to which damping devices are attached. The use of highly efficient damping devices leads to a significant increase in the weight-size parameters of the ETC. The main of these parameters is the volume capacity of the ETC, which significantly affects the total weight of the vehicle, its specific power and, consequently, dynamic qualities. The best value of this parameter is achieved for vehicles with hybrid PS. At the same time, for the most common type of modern tracked vehicles under consideration (a classic powertrain system, including an internal combustion engine and a mechanical or hydromechanical transmission) of the leading global producers, this parameter ranges from 260 to 365 hp/m^3 [4].

The Experience in creation and operation of tracked and wheeled forestry vehicles of the world's leading manufacturers has shown that when using multi-cylinder engines alternative technical solutions can be used to protect powertrain from torsional vibrations that do not require an increase in weight and size indicators. Swedish and Finnish harvesters and forwarders (usually with wheeled undercarriages) use hydrostatic transmissions. In turn, the use of an elastic torsion shaft between the engine and transmission has become widespread when using hydrodynamic gears. This is most effectively implemented in the designs of wheeled vehicles of the Minsk Wheeled Tractors Plant, in the transmissions of tracked vehicles of various Russian manufacturers - snow and swamp vehicles TM-140, chokerless skidders ML-107, LP-18K, etc..

In these designs, the torsion shaft is located inside the central hole of the gear of the matching gearbox or in a long hollow shaft located inside the torque converter (**Figure 1**). This technical solution allowed for many years to solve the problem of torsional vibrations for hydromechanical transmissions, working, for example, with engines of the Yaroslavl and Barnaul Motor Plants. At the same time, the protective torsion shaft is not a damper in the direct sense of the word. Its role is to bring the main resonance of the main harmonic into the shift of non-working zone of the engine shaft speed of 350-500 rpm, which lies below the idle speed of the engine. In the design modes of operation, the excitation of resonant oscillations occurs only when the engine is started in a short period of "passage" of the resonant zone. A similar

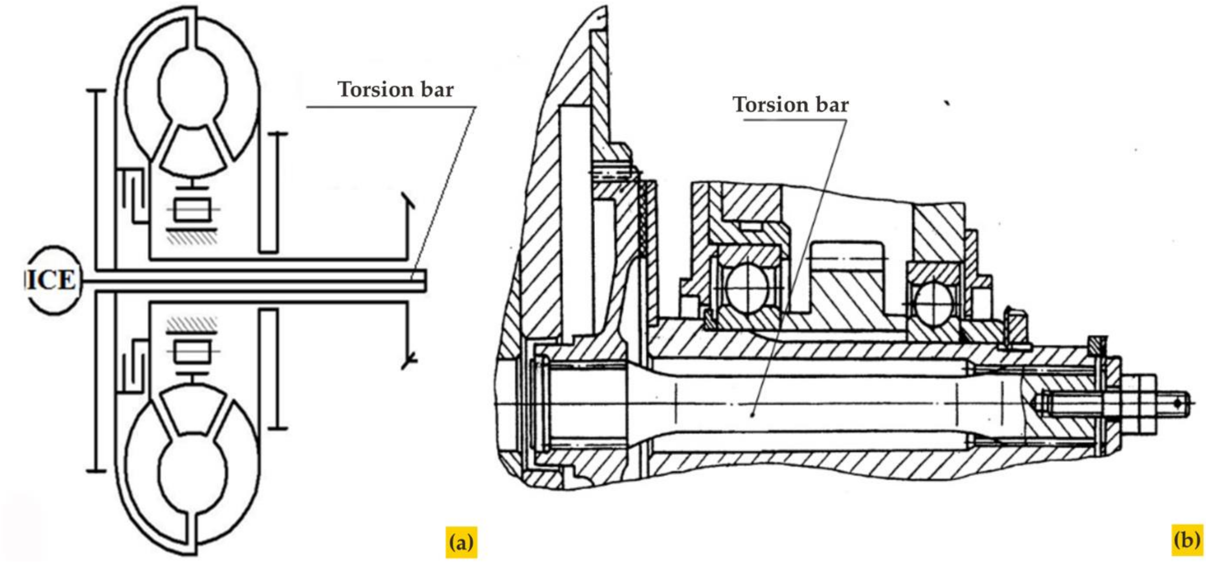

Figure 1. Options for using an elastic torsion shaft between the engine and transmission: (a) the torsion shaft is located inside the torque converter and (b) the torsion shaft is located inside the central hole of the matching gearbox gear

or close to this technical solution is currently used on most vehicles in Russia and Belarus. In [5], [6] the possibility of using torsion shafts as filter elements was studied with an increase in the power of a diesel engine, taking into account the acceleration of the shaft in the processes of starting and stalling, and the value of the maximum amplitudes of the dynamic moment was estimated.

At the same time, the experience of operating prototypes of the vehicles with increased engine power and equipped with new intelligent control systems has shown that in some cases there are repeated faults and failures of elastic bars (see Figure 2) connecting the transmission and various engine and transmission components mated with them.

Due to the failure of elastic bars in the above conditions, some experts voice doubt about a possibility of using such dampers in PS with high-power engines. At the same time, the findings of this research proves that the doubts are not justified, and the durability of torsion bars is determined by the peculiarities of the load modes appearance in the modern vehicles PS. The limited resource of elastic bars is determined by the peculiarities of their functioning, which is manifested in the occurrence of off-design resonant operating modes. In the existing methods of design and tests of transmissions, these modes are not taken into account. The above does not allow us to conclude about possibility or impossibility of using torsion bar springs in powertrain systems with increased power. The ambiguity of the conclusions is due to lack of reliable experimental data on the values of instantaneous torques and the conditions for resonant modes occurrence.

Figure 2. Low-cycle failure of elastic transmission bars when operating in off-design modes (in resonant modes causing plastic deformations)

For example, in the design of the experimental transmission of the TM-140 snow and swamp-going vehicle with a hydrodynamic torque converter (TC) and a hydrostatic steering unit (HSU), due to the limitation of its power, the required dynamics of vehicle turn is provided by a differential braking control system, which can create an additional yaw moment. This system is used when maneuvering (in the restricted size of the area), in severe dirt road conditions (deep track, etc.). The peculiarity of this system functioning is that when there is insufficient yaw moment, the parking brake of the lagging track is activated. The condition for activating this mode is provided by turning the steering wheel at an angle of more than 95 percent of the maximum possible value and simultaneously engaging the brake pedal. With such a short-term turning of the steering wheel to a position corresponding to an angle of more than 95%, the caterpillar brake of the lagging track is engaged. This is accompanied by an overload of the engine, a decrease in the rotational speed of its shaft to 300 rpm or less, and the engine “holdup” in the resonance zone. Complete engine stall does not occur in many cases, since the automatic PS protection system drains the working fluid from the transmission clutch boosters, which leads to a slight increase in engine speed and its return to the resonant mode or its passing through the resonant zone with a slight acceleration. Another set of conditions leading to destructive failure of elastic bar springs arises due to the inconsistency of the algorithms of the engine control unit equipped with the Common Rail fuel supply control system and the transmission controller unit (TCU), resulting in a phenomenon called «collision of tasks» [7].

The above-mentioned operating modes of the PS are «off-design» and are not taken into account when estimating the dynamic loading of the engine and transmission structural elements, at the same time this may cause the destruction of the torsion bar. To confirm this hypothesis, it is necessary to measure the dynamic torque with sufficient accuracy. However, reliable experimental studies of the real dynamic loading of torsion bars have not yet been carried out due to complexity of measuring torques in case of dense configuration of modern transmissions (especially, as mentioned above, with the hydrodynamic transmission, where the torsion bar is located inside the central opening of the gear and a long fluid coupling, as well as inside the TC). Nevertheless, recently, using the features of up-to-date software and hardware, it seems possible to measure the dynamic torque in a non-contact way, in gear systems with dense configuration as well.

2. Development of a Contactless Method for Experimental Determination of the Instantaneous Torque

One of the tasks for creating structures of prospective powertrain system of transport vehicles and upgrading the existing ones is an experimental estimation of dynamic loading during on-road tests to assess the correctness of assumptions made at modeling and to collect information to perform calculations for design integrity and durability.

There is a method for experimental determination of the dynamic torque [8], which consists in installing strain gauge transducers, a measuring system on the rotating parts of the transport vehicles PS with subsequent registration and statistical data processing. In this case, it is necessary to introduce devices into the measuring system for contact (current collectors) or contactless (telemetry) transmission of signals from the sensors mounted on the rotating parts and providing them with a supply voltage for subsequent processing. This method is characterized by the complexity of measurements and information processing, as well as by the limited lifetime of system elements [9], [10].

In modern and prospective designs of transport vehicles, onboard information and measurement control systems (OIMCS) are created, including sensors for measuring the angular velocity of PS elements and control shaping [11]. This makes it possible to determine the engine torque at a known external speed characteristic from the measured values of the angular rate of rotation of the engine shaft and the position of the fuel intake control. This method provides measurement of the current (static) engine torque, possibility to control the movement of the transport vehicle, diagnostics of the technical condition, solving other tasks, but does not allow to evaluate the high-frequency instantaneous torque, as well as to determine the sources of its appearance. The real dynamic torque differs significantly from the value determined by calculation algorithms embedded in the OIMCS. The dynamic torque contains high-frequency components formed by the engine mechanisms i.e. by the crank-shaft and connecting rod assembly (CCRA) and gas valve timing (GVT), as well as generated by dynamic processes in oil pumps, in the torque converter, electric generator, tracked vehicle propulsion system, etc. These components are not stable in time and are most accurately determined by the spectral density of the process. It is the dynamic torque (its periodic components characterized by amplitudes, frequencies and phases) that determines the durability of the PS structure elements.

In the proposed method, the instantaneous torque in the PS elements is determined by the results of processing the initial signal of the OIMCS angular velocity sensor measured with a high sample rate (over 30 kHz) – for example, in form of a meander wave (see **Figure 3**). For this purpose, a pulse sensor is installed in powertrain system, for example, an inductive sensor or a Hall sensor (see **Figure 4**). Such a sensor works in tandem with a master ring, which can be any convenient gear or specially installed toothed disc (impeller).

The essence and novelty of the method consists in determining the dynamic torque as the product of multiplying the moment of inertia J_i of the i_{th} element of the dynamic system by angular acceleration M_q=$J_i \cdot (\omega_i)^{\cdot}(t)$, where the angular acceleration function $(\omega_i)^{\cdot}(t)$ is determined by the results of measurement and differentiation of filtered signal of high-frequency periodic angular velocity deviations locked onto (and modulating) the carrier frequency of sensor pulses (meander frequency). The useful (desired) signal of the modulating frequency arises due to the functioning of various mechanisms that create disturbing effects. For example, at the flywheel of an engine, such disturbances may be ordinal frequencies of the engine harmonics of the internal combustion engine, disturbances from oscillations of the generator at its eigenfrequency, disturbances from the valve train and engine oil feed pump, disturbances from vibrations of the pre-convertor zone of the hydromechanical transmission, etc. At the same time, the hardware should, in accordance with the Kotelnikov-Shannon theorem [12], [13], [14], provide the required frequency of sensor request, and the processing program should support filtering functions of forward and reverse Fourier transform, determination functions for cyclic frequency, differentiation, etc. [15], [16], [17].

The flow-chart of the algorithm for determining the instantaneous torque in the powertrain system during the movement of the transport vehicle is shown in **Figure 5**. In block 1, the initial data is generated, and in block 2, the signal from the engine shaft speed sensor is digitized and transmitted to the registration and processing device. In block 3 and block 4, based on the spectral analysis of the signal (direct Fourier transform), the carrier frequency is filtered and the range of its change is determined. Applying cyclic functions (calculating the frequency of each cycle), the time function of the filtered signal is determined in block 5 (the circular frequency of the modulating signal induced onto the carrier frequency (being modulated). Differentiation and normalization of this function (block 6) determines the angular acceleration of the engine $\dot{\omega}_i(t)$ and, respctively, the function of the dynamic torque $M_{q_i} = J_i \cdot \dot{\omega}_i(t)$. On the basis of the direct Fourier transform (block 7), the amplitude-frequency response function of the torque is determined, including the fundamental

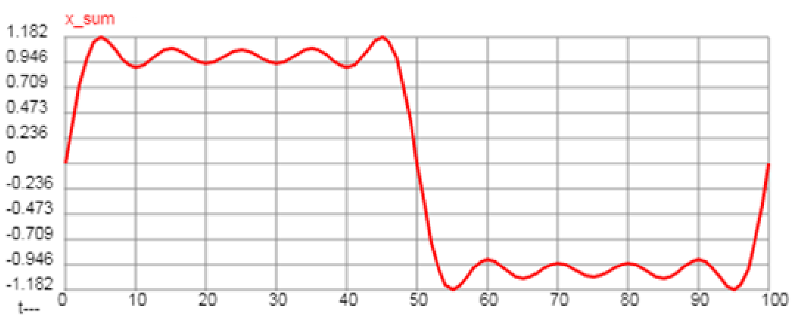

Figure 3. Example of a signal measured with a high frequency in the form of a meander wave taken from the analog output of the pulse sensor

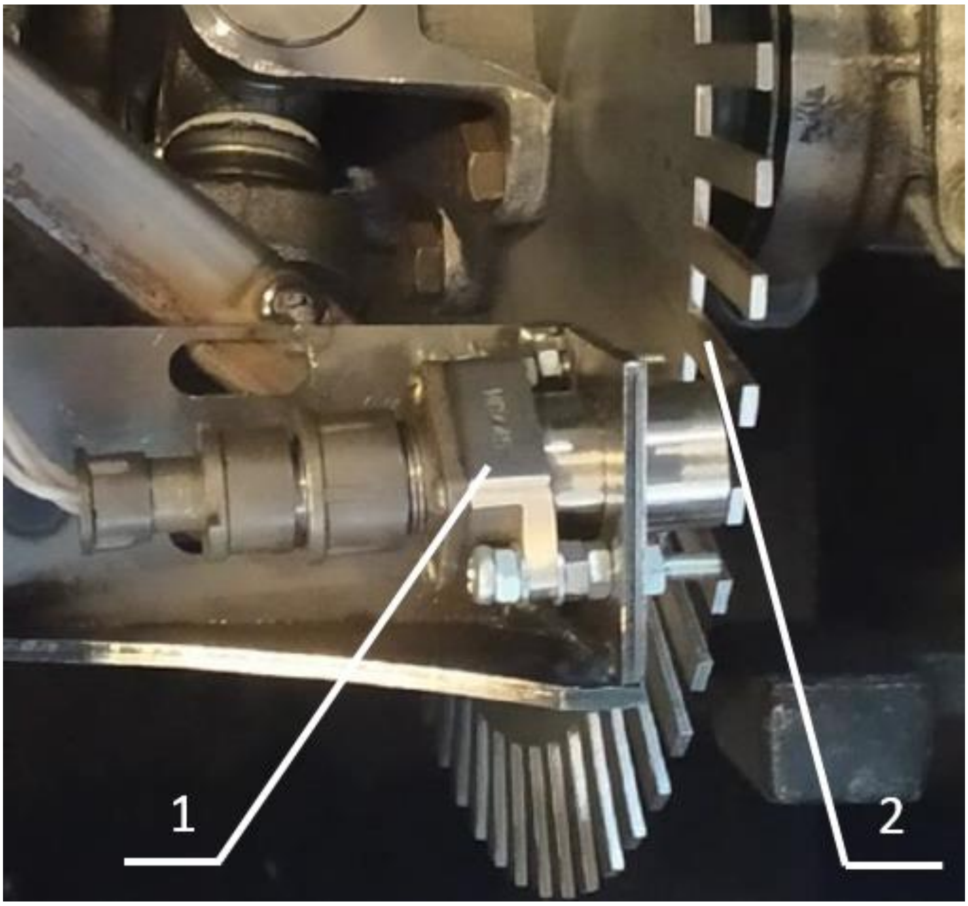

Figure 4. Example of pulse sensor (1) in powertrain system in tandem with impeller (2)

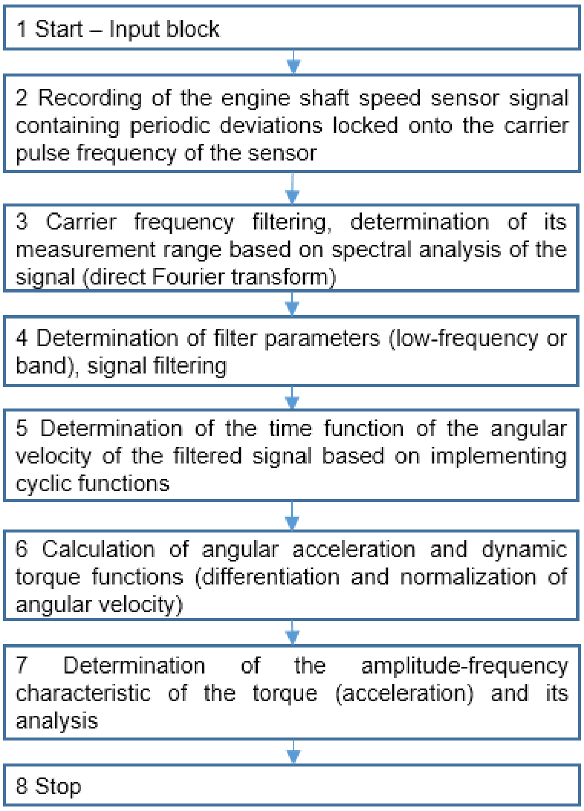

Figure 5. Flow-chart of the algorithm for determining the instantaneous torque in the powertrain system during the movement of the transport vehicle

harmonics of the engine, harmonic components shaped by the engine CCRA and valve train, generator drive, vibrations in the transmission, etc. Thus, the proposed method allows using the initial signal (initial meander wave) of the angular velocity sensor of the engine shaft to determine the magnitude of the high-frequency instantaneous torque and its structural components. Experimental validation of the proposed method with simultaneous strain measurement of the shafts showed the accuracy of determining the torque not worse than 3% and is determined by the accuracy of measuring and differentiating the angular velocity of the shafts and determining the moment of inertia of the part.

The obtained values make it possible to determine the dynamic loading and predict the probabilistic assessment of the durability of the transport vehicle powertrain system elements, as well as to determine the sources of high-frequency instantaneous variable components of torque.

3. Determination of Dynamic Loading of the Elastic Bar, Including Under Non-calculated Loading Conditions

The effectiveness of the proposed method for determining the dynamic torque is illustrated in Figure 6 to Figure 8. The object of the study is a prototype of a TM-140 tracked snow-and-swamp-going vehicle equipped with a differential hydrostatic steering unit. Figure 6 shows the measuring fragment of changes in the dynamic torque at the input shaft of the transmission. Figure 6 illustrates the dependence of the actual (static) torque of the engine on the angular velocity of rotation in two positions of the gas pedal $\alpha_{acc} = 0{,}8$ and $\alpha_{acc} = 0{,}9$. The diagrams are based on the results of measuring the angular velocity of the engine shaft ω_q and the position of the fuel feed control (determined by the OIMCS CAN bus [18]). The dependence corresponds to the static characteristic of the tested engine. Figure 8 illustrates a set of ith harmonic components of the dynamic torque determined by the proposed computational and experimental method, generally characterized by amplitude, frequency and phase A_i, ω_i,, and φ_i. For the case under consideration, the maximum value of the dynamic torque $M_q(\omega_q)$ corresponds to the 3rd harmonic of the engine, which is the main source of the instantaneous torque (of excitation). In other cases, the amplitude-frequency response characteristic may have variable components of the dynamic torque formed by the mechanisms of the diesel. This representation of the dynamic torque (the signal processing result of the engine angular velocity sensor) makes it possible to accurately determine and predict the dynamic loading of the elements of the PS structure, reasonably form loading blocks that display cumulative stress levels and the corresponding numbers of cycles that the part accumulates within the range of the unit of the durability parameter.

4. Probability Estimation of the Elastic Bar Destructive Failure in the Calculated and Non-calculated Modes

The statistical processing of experimental data was carried out for two modes (calculated and off-design) of operating the tracked vehicle prototype

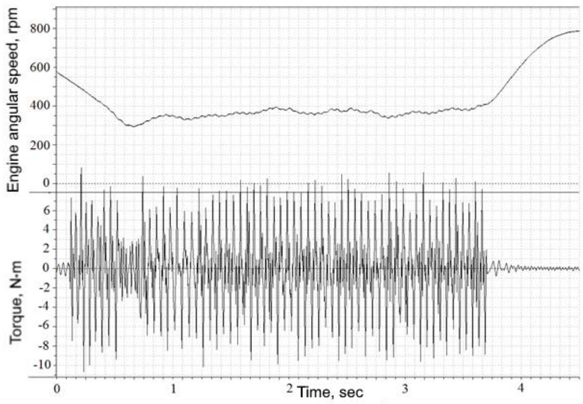

Figure 6. Measurement fragment of the change in the instantaneous variable component of the torque at the input shaft of the transmission

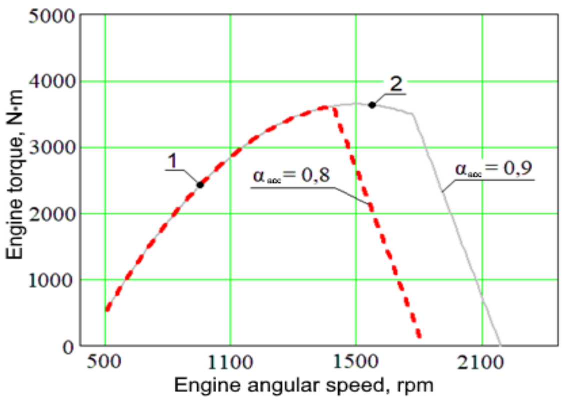

Figure 7. Dependence of the current (static) torque of the engine on the angular velocity of rotation at two positions of the fuel feed pedal $\alpha_{acc} = 0.8$ (1) and $\alpha_{acc} = 0.9$ (2).

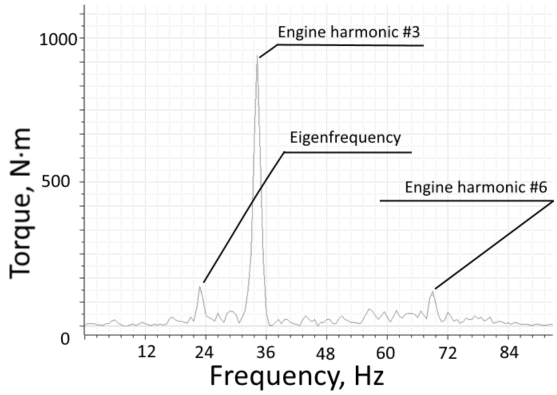

Figure 8. An example of the amplitude-frequency characteristic of the dynamic moment (i-th variable components of engine harmonics №3 and №6), determined by the proposed calculation-experimental method

- TM-140 snow-and-swamp-going vehicle with a hydromechanical transmission and a differential hydrostatic steering unit. These modes are characterized by statistical characteristics of the maximum stresses m_{σ_m} and σ_{σ_m} – mathematical expectation and root mean square deviation (see **Table 1**). The calculated mode takes into account the stresses being formed considering the probabilistic motion mode estimation in various road conditions, operation in the Neutral mode, at starting-stalling of the engine, etc. The off-design mode is characterized by operation at engine speeds corresponding to the resonance excited by separate braking control during the vehicle maneuvering. In both cases, the density of the distribution of maxima is determined by the Rice formula (Eq. (1)) [19]. The error function integral of stress maxima is determined by the expression (Eq. (2)) [20].

Based on Eq. (2), erf - Kramp function (error function, [21]), $\vartheta^2 = 1 - \beta^2$, $\beta = \frac{{\sigma_{\dot{\sigma}_m}}^2}{\sigma_{\sigma_m} \cdot \sigma_{\ddot{\sigma}_m}}$, β, β is the complexity factor of these process structure; σ_m is the stress maxima; σ_{σ_m}, $\sigma_{\dot{\sigma}_m}$, $\sigma_{\ddot{\sigma}_m}$ are the root-mean-square deviations of the stress maxima, its first and second derivatives.

The probability calculation of the occurrence of instantaneous faults from overloads is determined by A.R. Rzhanitsyn's theorem on probabilistic estimation of the sign of the magnitude S=R-Q, where R and Q are random values of strength and load characteristics [22]. The event S<0 corresponds to the case of destructive failure from overload effect, and the value S>0 does not correspond to the destruction of the structure. Therefore, the probability of non-destruction of the structure is determined by the Eq. (3).

Based on Eq. (3), $f(\sigma_m)$ is the maximum stress density (determined by the formula (1); $F_R(\sigma_\tau)$ is the cumulative distribution function of the material strength properties. For the case under consideration, R is determined by the mathematical expectation $m_{\sigma_\tau} = 760\ MPa$ and the root-mean-square deviation $\sigma_{\sigma_\tau} = 38\ MPa$ of the permissible tangential stress for 45KhN2MFA steel, from which the input torsion bar is made. The parameter Q is determined by the results of processing experimental data according to the above algorithm. The results of the statistical analysis of the maximum stress distribution process are summarized in **Table 1**.

As follows from the table, the calculated mode corresponds to the complexity factor of the process structure β, the values of which are in the range of 0.10 ... 0.15. In this case, Rice law transforms into a normal distribution law. In the off-design mode of operation, the complexity factor of the process structure β is close to 1, which indicates narrow-band, alternating sign and resonant loading described by the Rayleigh law. The functions of the distribution densities of the stress maxima for the calculated loading mode in **Figure 9a** and off-design one in **Figure 9b**, as well as the distribution density of the material strength characteristics.

$$(\sigma_m) = \frac{1}{\sigma_{\sigma_m}\sqrt{2\pi}}\left[\vartheta \cdot e^{-\frac{\left(\frac{\sigma_m - m_{\sigma_m}}{\sigma_{\sigma_m}}\right)^2}{2\cdot\vartheta^2}} + \sqrt{2\pi(1-\vartheta^2)} \cdot \left(\frac{\sigma_m - m_{\sigma_m}}{\sigma_{\sigma_m}}\right) \cdot e^{-\frac{\left(\frac{\sigma_m - m_{\sigma_m}}{\sigma_{\sigma_m}}\right)^2}{2}} \cdot F(\sigma_m)\right] \quad (1)$$

$$(\sigma_m) = \frac{1}{2} \cdot erf\left(\frac{\sqrt{2}\left(\frac{\sqrt{1-\vartheta^2}}{\vartheta} \cdot \left(\frac{\sigma_m - m_{\sigma_m}}{\sigma_{\sigma_m}}\right)\right)}{2}\right) + \frac{1}{2} \quad (2)$$

$$P(S < 0) = \int_{-\infty}^{\infty} f(\sigma_m) \cdot F_R(\sigma_\tau) \tag{3}$$

Table 1. The results of the statistical analysis of the maximum stress distribution process.

Loading mode	The complexity factor of the process structure β	Mathematical expectation of the maxima of the loading process, MPa	Root mean square deviation of the maxima of the loading process, MPa
Calculated mode	0.15	216.60	98.00
Off-design mode	0.90	339.50	260.00

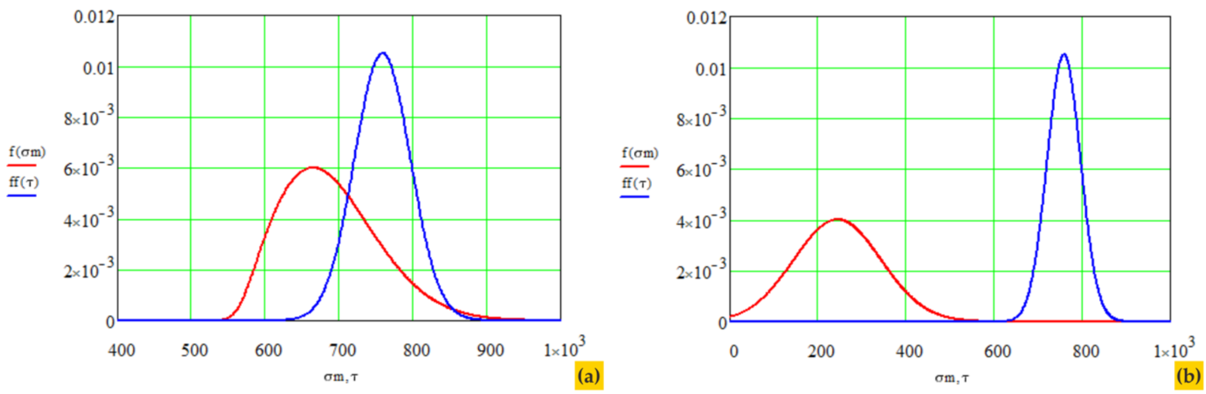

Figure 9. Distribution density functions of stress maxima for off-design (a) and calculated (b) loading modes

It follows from the above data that under the calculated loading conditions, the probability of instantaneous destruction from overloads does not exceed 0.0001, which confirms possibility of using torsion bars in the powertrain systems of prospective transport vehicles with increased engine power. At the same time, when off-design loading modes are excited, the probability of instantaneous destruction from overloads reaches 0.29. Given the fact that this result was obtained for a dynamic system which elastically and inertial parameters were selected in full accordance with the existing standard calculation methods, this result is shocking. This circumstance requires special attention to the issue of excluding off-design operating modes that arise as a result of the «collision of tasks» phenomenon in modern mechatronic systems, as for providing of durability, in particular, of torsion bars in high-duty PU, it requires development of engineering solutions that exclude off-design loading modes.

5. Conclusion

The following conclusions can be drawn from the above results:

a. The conducted studies revealed that the imperfection of the functioning algorithms of new intelligent systems used in modern transport vehicles leads to the appearance of "off-design" modes of operation, which are the result of the so-called «collision of tasks» phenomenon. In the framework of the study performed above, this phenomenon manifested itself in the process of starting the engine and in the process of maneuvering the machine on limited areas in the “separate braking” mode, which led to the excitation of resonant modes and high dynamic loading.
b. Based on the use of modern software and hardware, a calculation and experimental method for determining and analyzing the torque in powertrain of the vehicle, which consists in the separate calculation of the steady-state and dynamic components of the engine torque and their subsequent summation. The steady-state component is determined by the external and regulatory characteristics of the engine, depending on the position of the fuel supply control and the engine shaft speed. The dynamic component is defined as the vector sum of the periodic components allocated on the basis of the Fourier transform of the angular acceleration function of the engine shaft an the dynamic torque function accordingly. This

representation of the dynamic moment (the result of processing the signal of the engine angular velocity sensor) makes it possible to accurately determine and predict the dynamic loading of structural elements of the power plant, reasonably form loading blocks that display a set of stress levels and the corresponding numbers of cycles that the part develops within the unit of the durability parameter.

c. According to the results of the study, it was found that in the powertrain of promising machines with increased engine power, in the event of a «collision of tasks» phenomenon, dynamic loads may occur that are much higher than the calculated values. With regard to the object of study (snow and swamp vehicle TM-140), this phenomenon occurs during engine start-up and in the process of maneuvering the vehicle on limited areas in the "separate braking" mode. In this case, the probability of instantaneous failure is 0.29, which is unacceptable. It was established that if this phenomenon is excluded torsion shafts without additional dissipative devices can be used as an elastic element connecting the engine with the transmission.

Author's Declaration

Authors' contributions and responsibilities

The authors made substantial contributions to the conception and design of the study. The authors took responsibility for data analysis, interpretation and discussion of results. The authors read and approved the final manuscript.

Funding

No funding information from the authors.

Availability of data and materials

All data are available from the authors.

Competing interests

The authors declare no competing interest.

Additional information

No additional information from the authors.

References

[1] A. Milasinovic, I. Filipovi, Z. N. Milovanovic, and D. M. Knežević, "Determination of the engine torque of a four cylinder four stroke diesel engine on the basis of harmonic analysis of the crankshaft's angular velocity," *Transactions of FAMENA*, vol. 35, no. 4, pp. 55–65, 2011.

[2] X. Wang, "Effect of AVL-based time-domain analysis on torsional vibration of engine shafting," *Journal of Vibroengineering*, vol. 26, no. 6, pp. 1284–1300, Sep. 2024, doi: 10.21595/jve.2024.24143.

[3] M. A. Klesso, "Parameter Selection of Four-Cylinder Diesel Torsional Vibration Dampers," *Mechanics of Machines, Mechanisms and Materials*, vol. 4, no. 57, pp. 27–32, Dec. 2021, doi: 10.46864/1995-0470-2021-4-57-27-32.

[4] O. U. Usov, M. G. Gusev, A. L. Loyko, and A. M. Makarov, "Electromechanical transmission for military tracked vehicle with hybrid power plant," *St. Petersburg State Polytechnical University Journal*, vol. 219, no. 2, pp. 167–174, Jun. 2015, doi: 10.5862/JEST.219.18.

[5] G. S. Beloutov, "Mathematical model for calculating the dynamic loads in the input transmission elements during the passage of the resonance zone at engine starting," in *Aktualnye problem zashchity I bezopasnosti: Trudy XVII Vserossijskoj nauchno prakticheskoj konferencii RARAN (1-4 aprelya 2014 g.)*, 2014.

[6] I. Taratorkin, V. Derzhanskii, and A. Taratorkin, "Oscillation damping in the power unit when starting the engine equipped with the Common Rail system," *MATEC Web of Conferences*, vol. 224, p. 02037, Oct. 2018, doi: 10.1051/matecconf/201822402037.

[7] Bosch, *Diesel engine control systems [In Russian]*. Moscow: Za rulem Publ., 2004.

[8] V. A. Shushkevich, *Fundamentals of electrotensometry [In Russian]*. 1975.

[9] D.-C. Lee, M.-H. Song, and S.-H. Kim, "Measurement and Assessment on the Shaft Power Measurement of Diesel Engine using Strain Gauge in Marine Vessel," *Journal of the Korean Society of Marine Engineering*, vol. 33, no. 8, pp. 1152–1161, Nov. 2009, doi: 10.5916/jkosme.2009.33.8.1152.

[10] Y. P. Manshin, E. Yu Manshina, and M. Geue, "About the dynamic error of strain gauge torque measuring devices," *Journal of Physics: Conference Series*, vol. 2131, no. 5, p. 052041,

Dec. 2021, doi: 10.1088/1742-6596/2131/5/052041.

[11] A. M. Shcherbin, “Modern on-board information and control systems of automotive technology [In Russian],” *Zhurnal avtomobil’nykh inzhenerov*, vol. 3, 2015.

[12] V. A. Kotel’nikov, *Problems of noise-immune radio communication: radio engineering collection [In Russian]*. Gosenergoizdat, 1947.

[13] N. N. Andreyev, “Founder of domestic classified telephone communications [In Russian],” *Radiotekhnika - Radio engineering*, vol. 8, 1998.

[14] C. E. Shannon, “A Mathematical Theory of Communication,” *Bell System Technical Journal*, vol. T.27, no. C.379-423, p. 623—656, 1948.

[15] P. Gaydecki, *Foundations of Digital Signal Processing: Theory, Algorithms and Hardware Design*. University of Chicago Press, 2004.

[16] N. E. Huang *et al.*, “The empirical mode decomposition and the Hilbert spectrum for nonlinear and non-stationary time series analysis,” *Proceedings of the Royal Society of London. Series A: Mathematical, Physical and Engineering Sciences*, vol. 454, no. 1971, pp. 903–995, Mar. 1998, doi: 10.1098/rspa.1998.0193.

[17] PowerGraph, “User Manual [In Russian],” Power Graph. [Online]. Available: http://www.powergraph.ru

[18] Society of Automotive Engineers, *SAE J1939-71 — Vehicle Application Layer Issued 1994-08, Revised 2003-12*. 1994.

[19] S. O. Rice, “Mathematical Analysis of Random Noise,” *Bell System Technical Journal*, vol. 24, no. 1, pp. 46–156, Jan. 1945, doi: 10.1002/j.1538-7305.1945.tb00453.x.

[20] M. Stroyizdat, *Application of methods of probability theory and reliability theory in calculations of structures [In Russian]*. 1971.

[21] M. Abramowitz, I. A. Stegun, and R. H. Romer, “Handbook of Mathematical Functions with Formulas, Graphs, and Mathematical Tables,” *American Journal of Physics*, vol. 56, no. 10, pp. 958–958, Oct. 1988, doi: 10.1119/1.15378.

[22] M. Stroyizdat, *Theory of calculation of building structures for reliability [In Russian]*. 1978.